INDUSTRIAL STEAM TURBINES IMPROVE ENERGY UTILIZATION IN THERMAL POWER PLANTS

by

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ABSTRACT

Owing to high fuel costs, European Companies have always had to use special care in planning their process-energy requirements. Therefore, the industrial steam turbine has also played a major part in this respect.

The constant quest for improved fuel-economy and the development of new processes involving the use of turbines brought with them a great variety of heat-cycle arrangements, resulting in a considerable increase in the scope of design parameters.

In the fifties, it had become possible to build turbines individually matched to each particular process. But in the face of increased demand for turbines, the high cost of designing and building such machines, as well as the greater risk involved, no longer justified this practice.

This led in the sixties to the development of standardized turbines comprising several series of basic types. The type coming closest to the operating conditions encountered was normally selected for a given application.

The growing number of requirements imposed on steam turbines, and increasingly higher steam conditions, led to the development in the late sixties of a modular system of industrial steam turbines having the same reliable performance as a standard-type turbine.

Examples are presented to show how the most economical turbine can be composed from a building-block system. Design features allowing highly reliable performance of the turbine despite nearly individual adaptation to process requirements at maximum design parameters are also discussed.

INTRODUCTION

Germany covered its energy requirements from indigenous coal fields right into the 1950's. The difficult and costly mining of the coal made it imperative to ensure maximum efficiency in its utilization.

The progressive rise in the energy demand of the 1960's and the ever increasing cost of mining the coal, as well as the limited reserves in Western Europe justified the import of fuels, particularly the relatively cheap oil. Compared to the countries with their own, often easily accessible fuel resources, Europe was forced to exercise great care in the utilization of energy also during this phase.

The developments of the 1970's finally shed a new light on the energy and raw materials situation.

In consequence, processes which save energy and raw materials are being given priority in industry. Added to this are the cooling water and environmental problems which tend in the same direction.

The increase in population leads to the development of new processes. The unit ratings of industrial generating plants are also becoming larger. The heat flow arrangements must be planned with greater care.

Very soon after its invention, the steam turbine was introduced in several branches of industry to provide a better conversion of energy. New applications were constantly added; particularly where prime movers were required for generators, compressors and pumps of higher ratings.

The 1950's were a decisive period of development for the steam turbine in Germany. As a result of the energy situation described above the steam turbine was increasingly superimposed on thermal processes in industry.

To improve the thermal efficiency, high values were selected for the inlet steam conditions, particularly in the chemical industry. The turbine developments for inlet conditions of 300 bar/650°C (4350 psi/1202°F) (Fig. 1) date from this time. They are still in use for continuous duty today. The expected advantages, however, were not attained because of the high plant costs due to the necessary austenitic materials used.

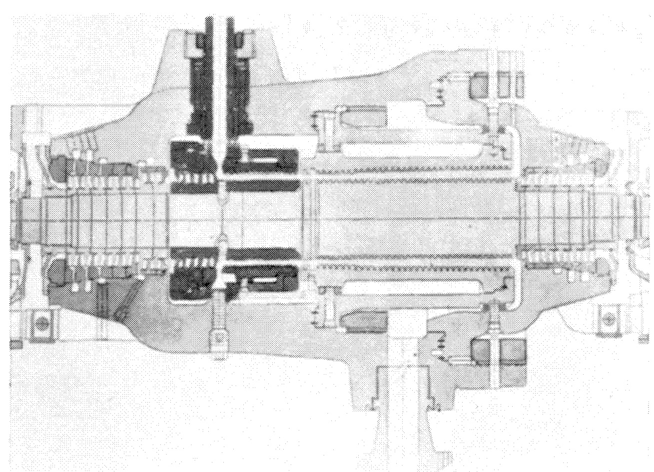

Figure 1. Longitudinal Section of a Barrel-Type High-Temperature Backpressure Turbine.

At that time, the turbines were custom designed and custom made for each application in order to obtain the desired turbine efficiency. The frequency of repetition of any particular type of these high duty turbines was still relatively low.

The 1960's were characterized by the more widespread advance of the steam turbine into new and other industrial fields. Besides generator drives, steam turbines were being used more frequently as turbo-compressor drives. In a number of processes, they were used for converting the exhaust heat of the process into mechanical energy, mainly in the form of the condensing turbine or the extraction condensing turbine. The greater number of units involved and the relatively small variation in the steam conditions led to the development of the standard turbine. For each type of turbine back pressure, condensing, extraction back pressure and extraction condensing turbines — a series of different sizes were designed. Their mechanical strength was so designed for the most frequent steam conditions of 45 bar/450°C (653 psi/842°F) that it was possible to use materials of lower quality. High-quality ferritic materials were then used for higher conditions. In order to be able to reach today's maximum values of 140 bar/540°C (2030 psi/1004°F), a new series of high-pressure back-pressure turbines were developed. Additionally they were designed as topping turbines in conjunction with the standard machines. The design with regard to thermodynamics and flow was prepared separately for each individual case.

The spectrum of design parameters, particularly the speeds, became ever wider. This finally led to the realization that with acceptable expenditure neither a custom design nor a system of rigid types would meet the requirements regarding efficiency and reliability. In consequence, at the end of the 1960's a beginning was made on the development of a modular system for industrial steam turbines.

MODULAR SYSTEM FOR INDUSTRIAL STEAM TURBINES

In order to be able to match the flow geometry of a turbine to the particular case, to assure a high degree of reliability and to keep the manufacturing costs and delivery periods within acceptable limits, the following concept was followed at Siemens in design and manufacture (1):

The turbine is divided into three main sections (Figs. 2 and 3)

- Admission section
  consisting of the steam chest with main stop valve and control valve, the wheel chamber part with nozzle casing, the glands, the front bearing with subsidiary drives, the associated part of the shaft with control stage and balancing piston.
- Exhaust section
  consisting of the exhaust hood, with condensing turbines including the last guide blade carrier, the shaft gland, the rear bearing, the associated shaft section, with condensing turbines including the l.p. blading.
- Centre section
  (consisting as required of reducing and/or extension pieces) which is required where it is necessary to extend the bladed section of the shaft or where small front sections have to be matched to large rear sections. It covers the centre portion of the casing and rotor with blading between the admission and exhaust sections. For machines with extraction the centre section also includes the extraction control valves and the diaphragm.

The admission section is manufactured in several sizes for two different working pressures and selected to suit the inlet or

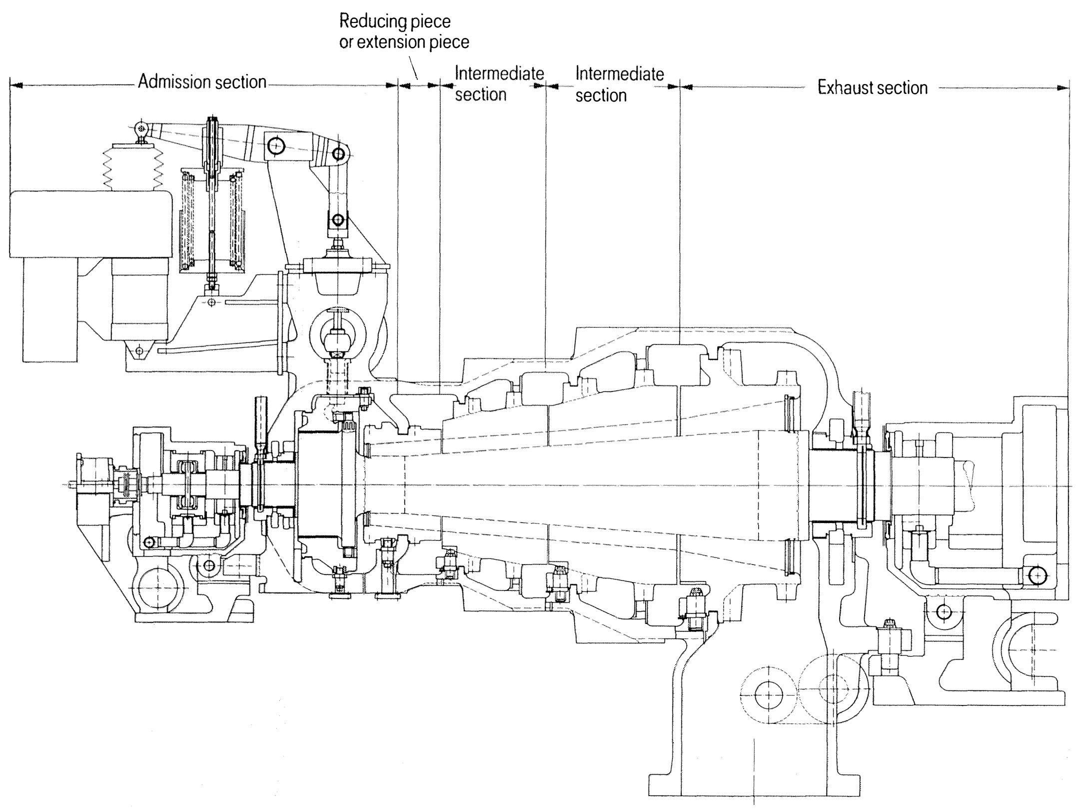

Figure 2. Longitudinal section of a Backpressure Turbine Showing the Division into Modular Sections.

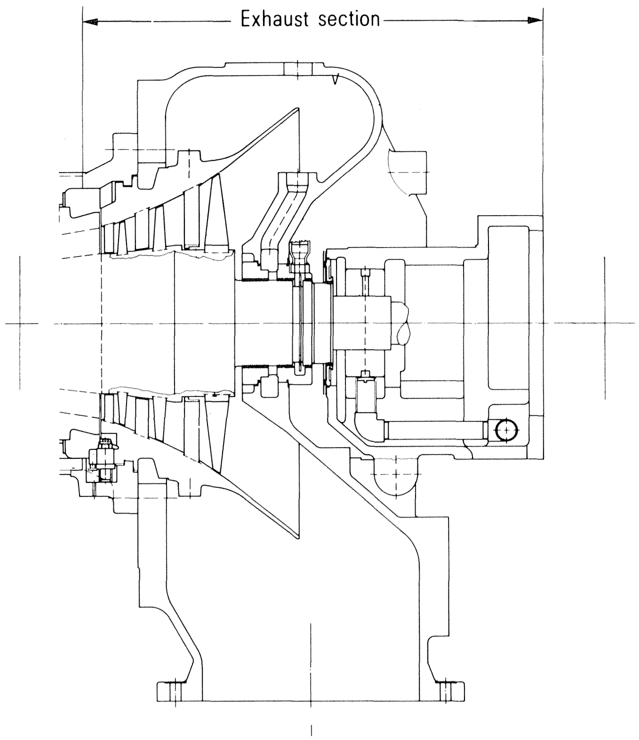

Figure 3. Exhaust Section of a Condensing Turbine.

wheel chamber conditions of the particular case irrespective of the centre or exhaust sections used. The advantages gained for the components of this section include larger manufacturing batches, greater operating experience and shorter design and manufacturing times.

The exhaust section is subject to the same design rules. In addition to two special stages for the backpressure version, it is necessary to have an exhaust section for condensing turbines. A low pressure stage set consisting of two or three stages with twisted blades is permanently associated with each exhaust section size. The values for the maximum speeds and the speed ranges are predetermined. The exhaust section can be combined with three different sizes of admission sections for both working pressures.

The centre section provides the connection, where required, between the admission and exhaust sections. An extension or a transition from a small admission section to a larger exhaust section also takes place here. In addition, both extension and transition are possible at the same time. The extraction control valves and the diaphragm also come within this section for turbines with controlled extraction. Particularly the relative blading length of the high pressure and low pressure parts brings advantages over the rigid types.

Any components within these sections which are mainly subject to static and/or thermal stresses can be used irrespective of the actual combination of the sections. The only criteria are the maximum stresses occurring in the particular case.

COMPONENTS TO BE MATCHED TO PARTICULAR CASES

The blading and the shaft must be considered individually also with the modular system, with the exception of the standardized low pressure stages. In the past, the blading was of two types:

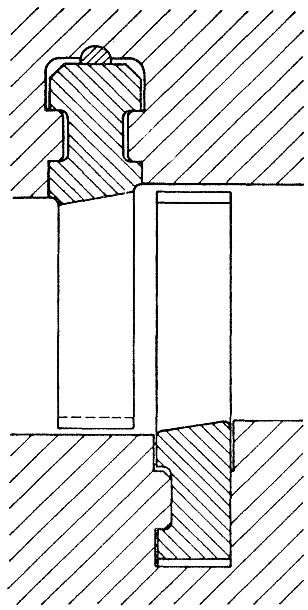

Figure 4. Previous Design of Reaction Blading.

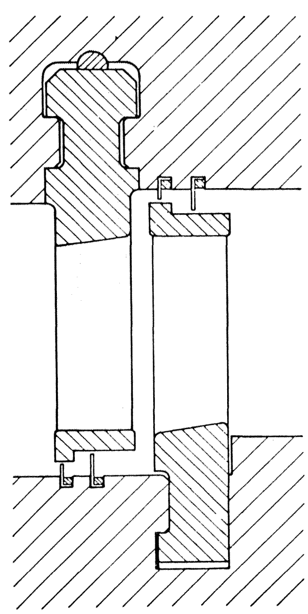

Figure 5. Present Design of Reaction Blading.

1. Free standing guide and moving blades held by the root (Fig. 4).

2. Moving blade with integral shroud and guide blade with riveted shroud, both held by the root (Fig. 5). This type of blading has smaller tip losses.

The blade profiles are taken from standardized families of profiles the flow characteristics of which are known.

In the case of compressor drives, the former type of blading has not proved successful. Irrespective of the type of driven machines, therefore, only the latter type is used today. Although the vibration characteristics are very difficult to determine in theory and to measure in practice the experience acquired with them has been very favourable, providing they have been accurately manufactured. It is not necessary to take any critical ranges into account; i.e., resonance is tolerated due to the outstanding damping by the shrouds.

The rotor, which is a single-piece forging, can be assigned to the individual sections only as far as the geometrical dimensions are concerned. With regard to its dynamic characteristics, an individual analysis is required for each unit. A characteristic feature of these drum-type rotors is the relatively large ratio of the diameter to the distance between bearings. This, however, also brings the important shaft stiffness, which is obtained from the ratio of the maximum deflection to the radial bearing clearance into the favorable range (2). New methods of calculation added to experience permit reliable calculation of the amplitudes and the preparation of balancing specifications. No critical ranges for the avoidance of large amplitudes are stipulated. No instabilities due to oil whip or gap excitation have yet been observed, although rigid four-and two-lobe bearings have been used throughout. Tilting-segment journal bearings have not found favour with rigid rotors due to the poorer damping.

For individual matching of the turbine to the particular requirements, the selection of the wheel chamber pressure is a decisive factor. Fig. 6 shows the influence of the ratio of admission pressure to wheel chamber pressure on the efficiency of the turbine. This investigation also applies to the example in Fig. 8. It follows that with high inlet steam conditions it is also necessary to have high wheel chamber conditions. In the past the resulting problem concerned with the "wheel chamber joint" was avoided by using the so-called barrel-type design for the outer casing. This design has the advantage of short start-up times. For industrial steam turbines, however, the axially-split casing is given preference for reasons of servicing. In order to obtain high wheel chamber pressures, nevertheless, the admission sections are available in several versions. For example, the outer casing section is made with two wall and flange thicknesses. In limiting cases an inner casing is fitted so that the wheel chamber pressure does not act directly on the joint flanges. The inner casing can either be split axially or be made of the barrel-type for the highest pressures. Wheel chamber conditions of 100 bar/510°C (1450 psi/950°F) can be handled without having to accept long start-up times.

The necessarily high speeds for compressor drives add the problem of the mechanical strength of the first stage. In addition to the conventional blade fixing methods the design of blades integral with the rotor was developed (Fig. 7). Turbines with this type of blading have been in operation since 1968, particularly for the prime movers of syngas compressors in ammonia plants. With the manufacturing process in use today, only free-standing blades can be produced. Of advantage, however, compared to the fitted blade is that even with compact blades (low ratio of radius of gyration to blade height) the natural frequencies have practically no scatter as a result of the perfect clamping and do not vary in operation. This absolutely avoids resonance in the operating range or during the starting phases. In view of the compactness of the blade, it is generally only necessary to take into account one to four times the nozzle excitation. This type of blade is able to handle the highest mass flows at the highest speeds.

APPLICATION OF MODULAR SYSTEM

The following two examples of completed drives are intended to demonstrate how a solution can be found for the particular case by the combination of modular sections. Fig. 8 shows the heat flow diagram of a paper mill having a power demand of approximately 20 MW. Four possible combinations from the modular system are compared in Fig. 9. The efficiencies attainable for the individual combinations at 20 MW are indicated. The variation of the actual section alone with the

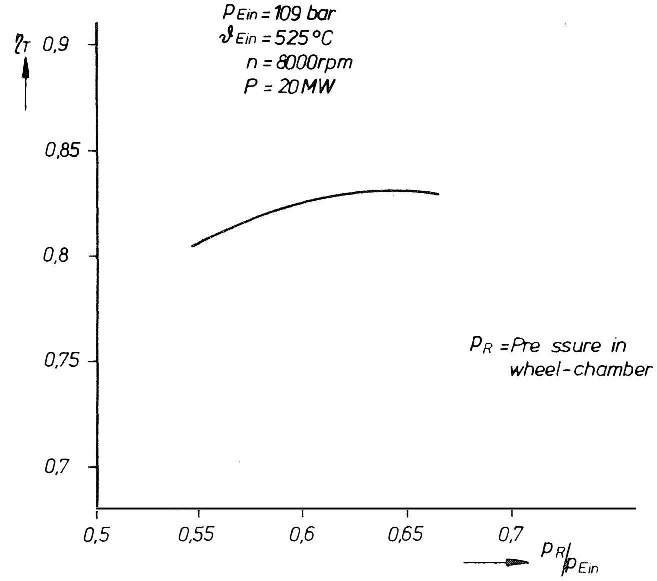

Figure 6. Influence of wheel chamber pressure on turbine efficiency.

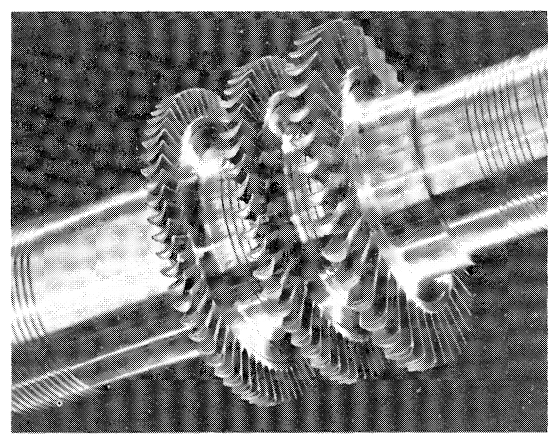

Figure 7. Electrochemically Machined (ECM) Blades for the Highest Speeds (Integral with Rotor).

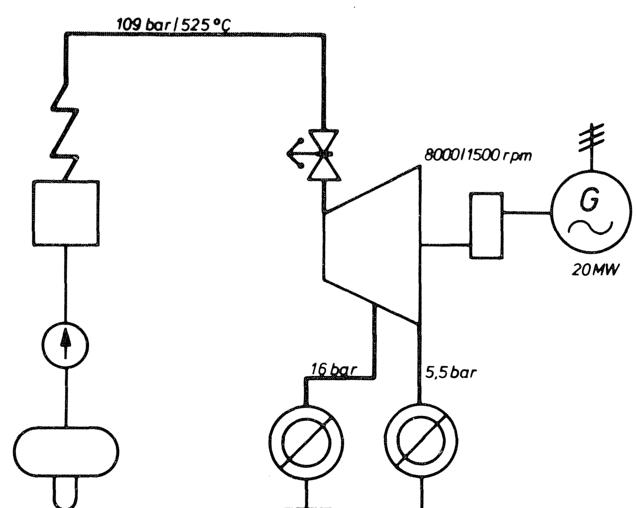

Figure 8. Heat Flow Diagram of a Paper Mill.

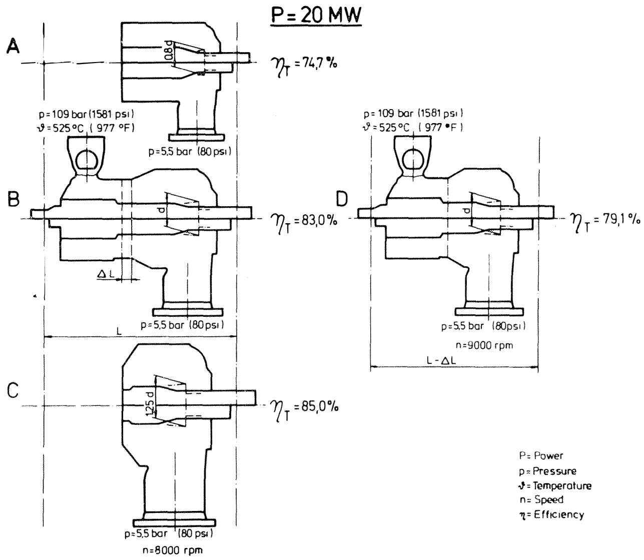

Figure 9. Schematic Diagrams of Section Combinations for Application Illustrated in Figure 8.

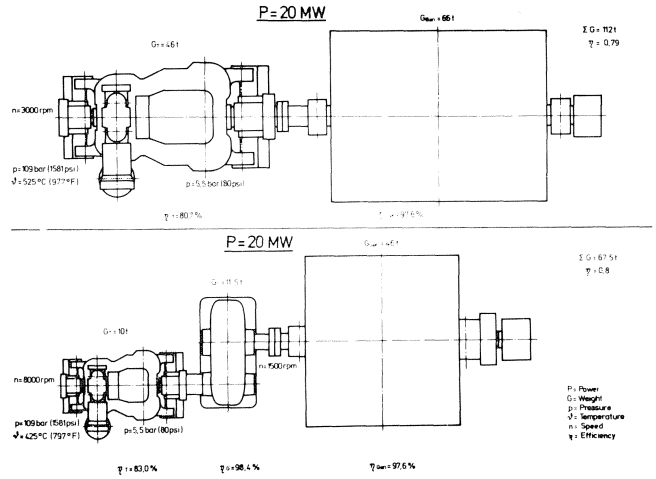

Figure 11. Comparison of a High-Speed Backpressure Turbine with a Direct-Drive Generator Turbine.

associated blading gives efficiency differences of 10%. In the case under consideration the combination B was used. The turbine was provided with an extension piece which brings 4% more efficiency than the non-extended combination D. A longitudinal section of the turbine supplied is shown in Fig. 10.

Compared to a direct-drive turbine running at 3000 rev/min, the two versions are shown for comparison in Fig. 11.

Despite the use of a reduction gear there is still a small lead in efficiency in favour of the high-speed version. The cost position can be estimated from the given weights. A number of high-speed geared turbines up to 20 MW are in operation in Europe. According to experience, there are no obstacles in the way of using high-speed turbines in the range up to 30 MW.

Figs. 12 and 13 show longitudinal sections of the turbines of a two-casing plant for driving a syngas compressor. The high pressure turbine is arranged as an extraction turbine. All parts, except the rotor, have been taken from the modular system.

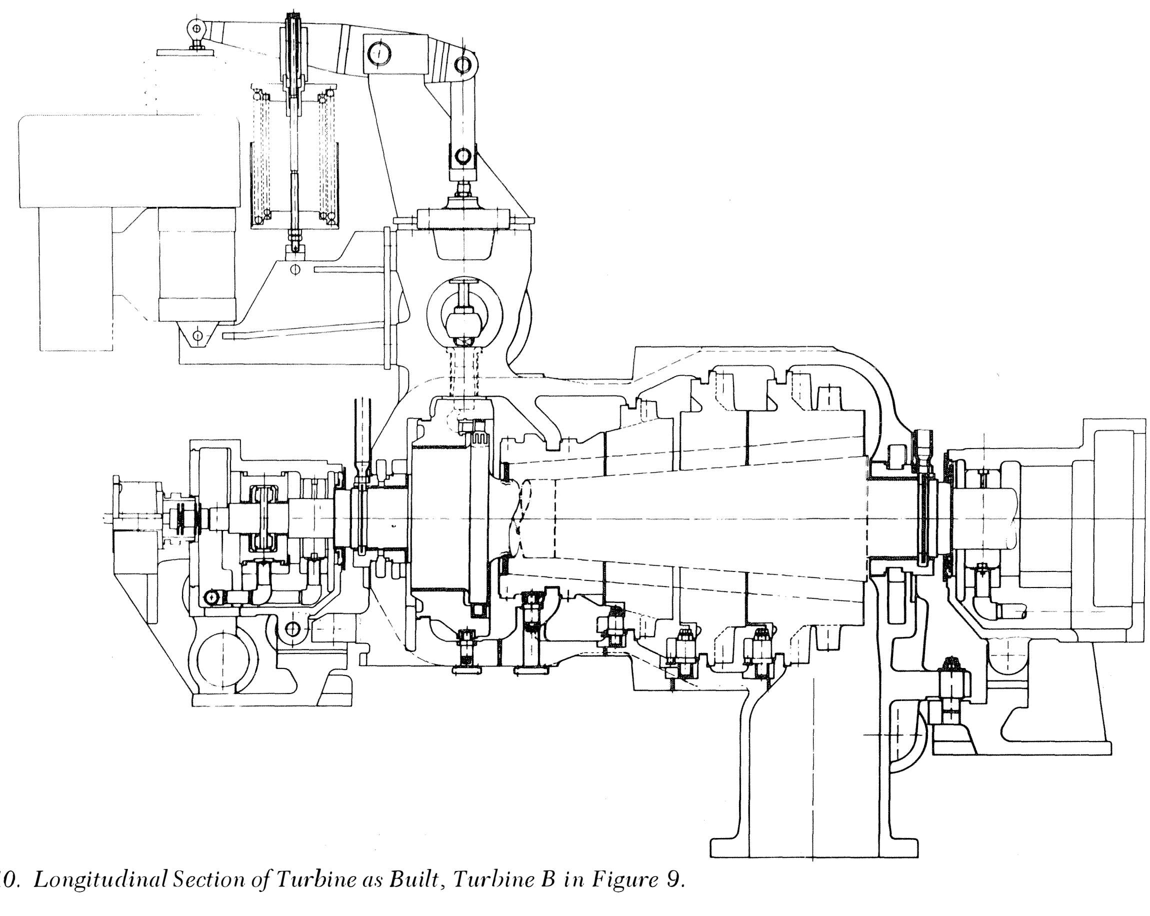

Figure 10. Longitudinal Section of Turbine as Built, Turbine B in Figure 9.

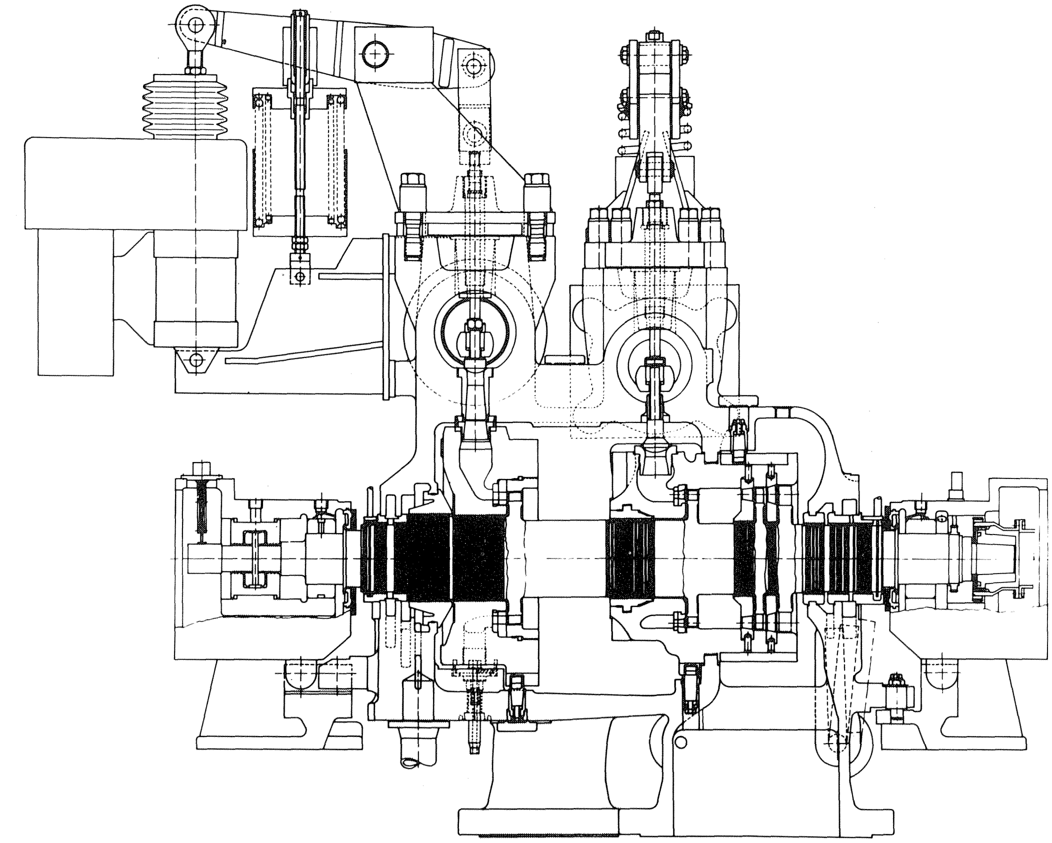

Figure 12. Longitudinal Section of Extraction Backpressure Turbine for NH_3 Syngas Compressor Drive (1st Casing).

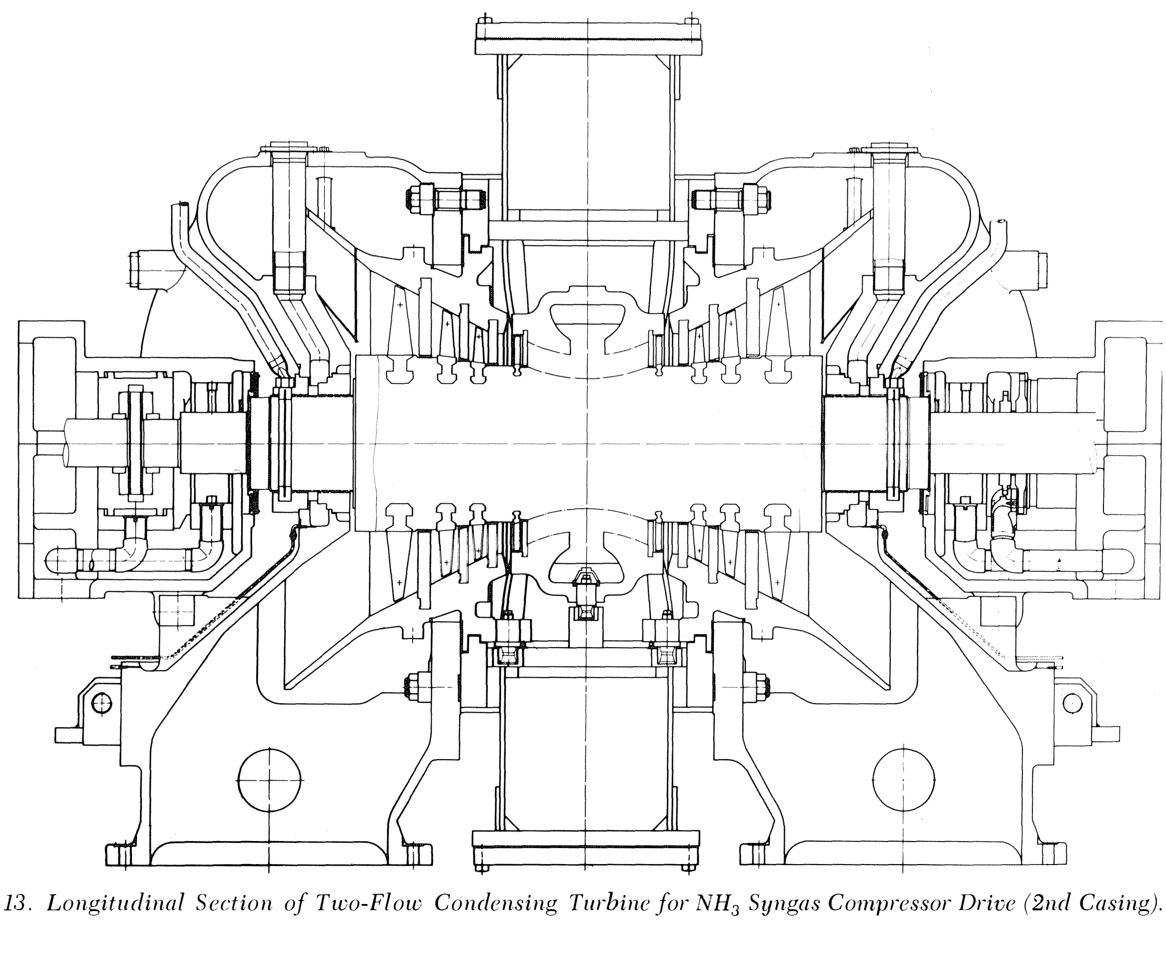

Figure 13. Longitudinal Section of Two-Flow Condensing Turbine for NH_3 Syngas Compressor Drive (2nd Casing).

This also applies for the two-flow condensing turbines. Two condensing exhaust sections have been connected in opposition. Only the central guide blade carrier and the central section of the outer casing form independent sections of the two-flow design. Here it was not so much a question of optimization of the efficiency but of a solution with proven components in view of the extreme ratings involved.

The output of the two-casing turbine is 33.5 MW (45,000 hp) at a speed of 11,300 rev/min. An amount of 470 t/h (1,040,000 lb/h) passes through the first stage with inlet conditions of 80 bar/480°C (1160 psi/896°F) and an extraction pressure of 42 bar (609 psi). In this case, the low pressure part is designed for a throughput of 86 t/h (190,000 lb/h) at an exhaust pressure of approximately 0.2 bar (2.9 psi). Turbines of this type are relatively small in number but form the heart of turbine drives of ammonia and methanol plants.

In view of the multitude of processes in this branch of industry, it is necessary to have a widely differing range of alternatives. For example, a number of high pressure turbines have been manufactured for reheating, the reheating temperature being approximately 500°C (932°F). The speeds, too, are frequently very much higher. The first Siemens turbine for a syngas compressor drive was built in 1968 with an output of 15 MW (20,000 hp) at a speed of 15,000 rev/min. The steam conditions with that plant were higher, being 111 bar/510°C (1,609 psi/950°F) at the inlet. An amount of 180 t/h (400,000 lb/h) passed through the three-stage hp part at an extraction pressure of 38 bar (551 psi).

It was intended to prove, by means of the examples described, that it is possible to handle even extreme design conditions with a modular system.

- In the case of the generator drive, the question of optimization with regard to efficiency and costs was the salient point.
- With the compressor drive, it was a question of a drive turbine for which there is relatively little demand but which is required to satisfy severe demands as to reliability.

EXPERIENCE GATHERED WITH THE MANUFACTURE OF TURBINES ACCORDING TO THE MODULAR SYSTEM

As mentioned at the beginning, it is expected of an industrial steam turbine, in the light of the present energy and raw materials situation, that it be matched to the requirements of a process not only technically but also economically, i.e., as regards efficiency, material and manufacturing costs.

An increase in thermal efficiency can be attained by the use of higher inlet steam temperatures. This method is also used for smaller plants so that it is necessary to build turbines ranging from 3 to 30 MW for the maximum conditions that are technically feasible today. It is expected, however, that the turbine efficiencies for this range are in the order of 70 to 85%.

- With generator drives, this efficiency level can be attained when using high-speed turbines. It is then necessary to interpose a reduction gear. As a result of the high turbine speed the diameters will be small and the blade lengths relatively large. In consequence, the clearance and tip losses are reduced. The gain in turbine efficiency is generally not absorbed by the reduction gear losses. The free selection of turbine speed facilitates the application of the modular system.
- With turbocompressor drives the direct drive is generally selected because of the speed level of this machine is already high. Nevertheless, efficiency and cost matching can be performed with the modular system by varying both the diameter and the blading length. In cases where the speeds limit the blading diameter the only possibility is an extension of the turbine, it being then necessary to take the rotor dynamics into account. At low speeds, large diameters are required. The variation possibilities in the blading length can be fully utilized. With a number of drives, the limit ratings for single-flow designs have already been reached. Higher ratings can only be achieved with two-flow machines. The modular system can be applied for such machines, too.

The experience so far gathered with the modular system has shown that, in the lower output range up to approximately 30 MW, it is possible to satisfy all technical requirements. Standard combinations can be taken from the modular system for the most frequent design data; individual combinations are required for extreme cases.

With the larger machines up to 100 MW, it has been found useful to manufacture the centre sections of the outer casings to suit the particular application. The admission and exhaust sections can be taken from the modular system. This applies particularly for machines with controlled extraction as are used in seawater desalination plants.

SUMMARY

A modular system for industrial steam turbines was developed with the aid of methodical design.

Generally, a number of module combinations can be offered for any individual case to permit economic assessment. This permits finely graded optimization.

Since interconnected operation, as is usual with public supply undertakings, is seldom possible with industrial plants, reliability takes absolute precedence. By the use of components proven in actual service, such a system satisfies this condition even with unusual specialized drives subject to extreme demands. Certain limitations in the application of this system are required for subordinate components if the machines are large and it is required to save costs.

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