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ADVANCED TECHNOLOGY OF STEAM TURBINE

Masami Nose
Yoshihiro Sakai
Takao Yamamoto

1 FOREWORD

Although the component ratio of thermal power plants in power source is decreasing year by year, they will continue to cover more than half of power generating capacity over ten years ahead. On the other hand, the growth of nuclear power generation above all has caused thermal power plants to turn over the role as base load to nuclear power plants. Thermal power plants, especially oil-fired power plants where the fuel cost is high, are required more intensively to perform their duties as power source for load adjustment. Therefore, besides the improvement of efficiency pursued up to now as one of the most important subject, strengthening and improvement of capability for cyclic operation such as DSS (Daily Start Stop) has become a significant topic of steam turbines for thermal power stations, in place of persuit of merits derived from enlargement of size, which is considered to have reached its limit.

The cost of primary energy appears to have stabilized temporarily, but is sure to rise in the long-range view and higher efficiency will remain the most important problem. Topics of steam turbines are roughly divided into:

(1) Development of construction and material to cope with cycle efficiency improving measures such as increase of steam pressure and temperature.
(2) Development to improve turbine internal efficiency, e.g. reduction of turbine internal losses mainly by improvment of blade profiles and enlargement of the last stage blades.

As for improvement of capability for cyclic operation, emphasis is placed upon technological development and perfection concerning:

(1) Suppression of life expenditure of equipment,
(2) Start-up circuit, operation, and control and monitoring which suppress the thermal stress of equipment,
(3) Stabilization of the shaft vibration characteristics, especially transient characteristic,

and

(4) Simplification and automatization of operation

at typical DSS operating conditions.

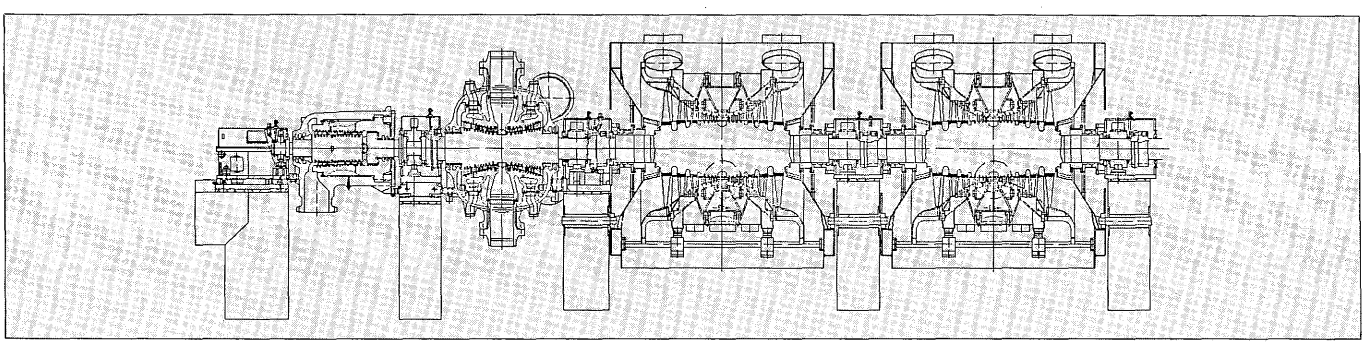
The design concept of Fuji Electric's steam turbines for power stations has been consistently directed toward "peak loads of tomorrow" from the beginning and has various features matched to DSS operation, such as three component design with a full-arc admission barrel type turbine at HP part and the separated IP turbine. Fuji Electric's newest technology for the topics mentioned above are outlined and example of its application are given below.

2 TECHNOLOGY FOR HIGHER PRESSURE AND HIGHER TEMPERATURE

2.1 Achievements of technology for higher pressure and higher temperature

The importance of higher steam pressure and temperature is increasing steadily as effective means of improving plant thermal efficiency. Fuji Electric has also played a leading role in this field. It is well known that Oi Thermal Power Station No. 3 Unit (350 MW) of the Tokyo Electric

Fig. 1 Sectional view of 375 MW 3600 rpm steam turbine



Power Co., Inc., which started commercial operation in 1973 as Japan's first supercritical sliding pressure unit, has exerted a great influence upon the technological development of domestic commercial thermal power plants thereafter. The sliding pressure system, which combines a spiral Bensen boiler and a full-arc admission barrel type turbine, used in the Oi Thermal Power Station No. 3 unit is essentially suitable to increasing the steam pressure and temperature and is passed on as one of the characteristic technology of Fuji Electric's steam turbines for power stations. The sectional view of the newest 375 MW, 3600 rpm unit is shown in *Fig. 1*.

The Ishikawa Coal-Fired Power Station No. 1 and 2 units (each 156 MW) of the Electric Power Development Co., Ltd., which started operation in 1986, uses 566 °C as main steam and reheating steam temperature. High temperature steam conditions without shaft cooling are realized with these units by using 12Cr steel for main parts such as rotors, inner casings and valves of HP and IP turbines. The 12Cr rotor was manufactured by the ESR (electroslag-remelting) method. The overlay welding of low alloy steel on the journal to prevent seizure was carried out for the first time in Japan.

2.2 Development steps for higher pressure and higher temperature

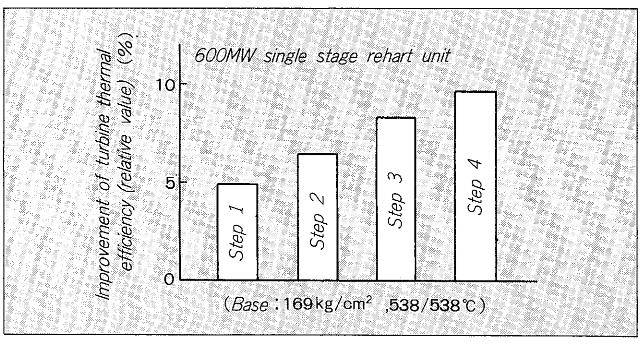
The following ultra super critical (USC) conditions are considered as steps to higher pressure and higher temperature in the future:

- Step 1 300 kg/cm^2, 570/570 °C
- Step 2 300 kg/cm^2, 600/600 °C
- Step 3 350 kg/cm^2, 650/600 °C
- Step 4 350 kg/cm^2, 650/650 °C

The improvement of thermal efficiency by these steam conditions is shown in *Fig. 2*.

Steps 1 to 3 can be realized with existing technology. The important technologies needed for these steps were developed in West Germany in the 1950s and 30 turbines with main steam temperature above 550 °C or main steam pressure above 280 kg/cm^2 went into operation in the 1960s. Of these, 14 turbines have main steam temperature between 600 and 640 °C. Except for disused plants, all these units are operating satisfactorily up to today, and three of them have achieved operating time over 200,000 hours.

Fig. 2 Improvement of thermal efficiency by use of ultra super critical condition



A barrel type turbine is used for HP section of these turbines. The barrel type turbine, which has no flange and is thermally stable owing to its rotationally symmetric form, is optimum for higher pressure and temperature and can be used without any problems up to main steam pressure 350 kg/cm^2. For the steam temperature of step 1, 12Cr steel is used for rotors, inner casings and other parts exposed to high temperatures so that the basic construction of the turbine need not to be changed. For the steam temperature of step 2, the inlet part of the HP and IP rotors must be cooled by external steam. Further, austenitic steel is used for the inner casings and inlet valves. At step 3, austenitic steel is also used for HP rotor. A one piece forged austenitic rotor with diameter of 830 mm and completed weight of 11 t (ingot weight 33 t) was already successfully manufactured on an experimental base. At step 4, IP rotor must be also made of austenitic steel and forging of a large austenitic steel is a topic of development.

3 TECHNOLOGY FOR IMPROVEMENT OF TURBINE INTERNAL EFFICIENCY

3.1 Results of development for efficiency improvement

Fuji Electric has promoted developments for improvement of turbine internal efficiency as the most important topic. The developed efficiency improving technology consists of various items. Only the most important items are introduced here.

(1) Improvement of reaction blade efficiency (Advanced reaction blade)

The aerodynamic characteristics of the new reaction blade were improved while maintaining its profile strength by using CAE (Computer Aided Engineering) techniques. This blade is used for HP and IP turbines and from the first stage to the intermediate stages of LP turbine. The new profile has succeeded to lower profile loss level while maintaining the advantage of the reaction blade that profile loss is low over wide range of inlet angle (*Fig. 3*). The blades are integrally shrouded; i.e. blade profile, shroud and root are milled solidly from one bar (*Fig. 4*). The blades

Fig. 3 Change in profile loss with inlet angle

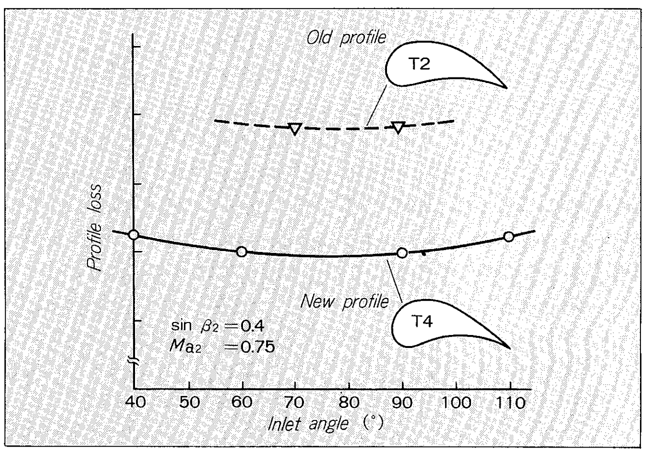


Fig. 4 Blade with integrated shroud

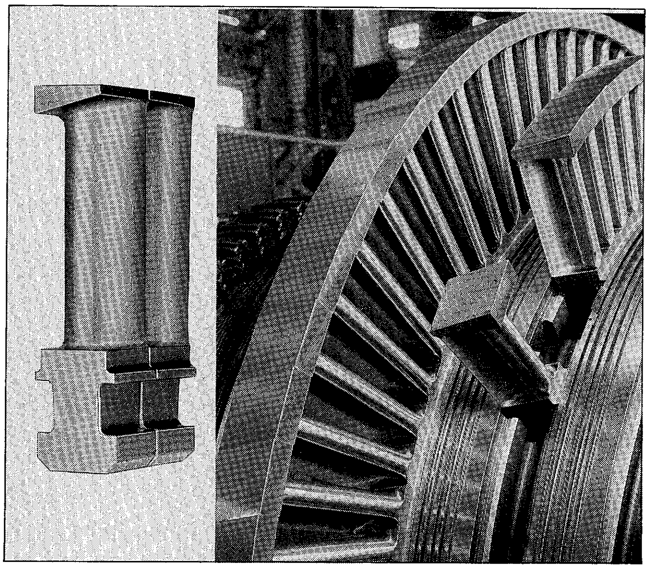


Fig. 5 Inlet section of IP turbine

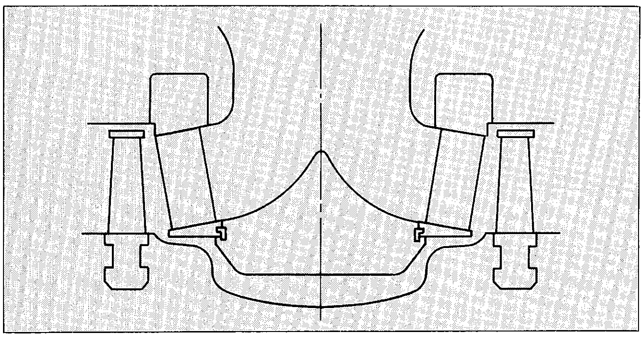


Fig. 6 Example of analysis of a tip section by three dimensional time marching method

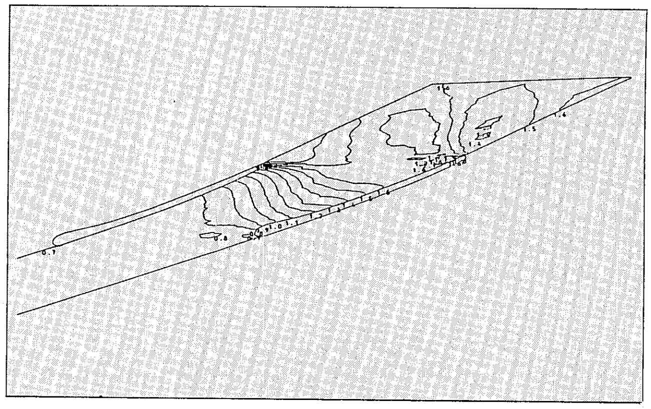
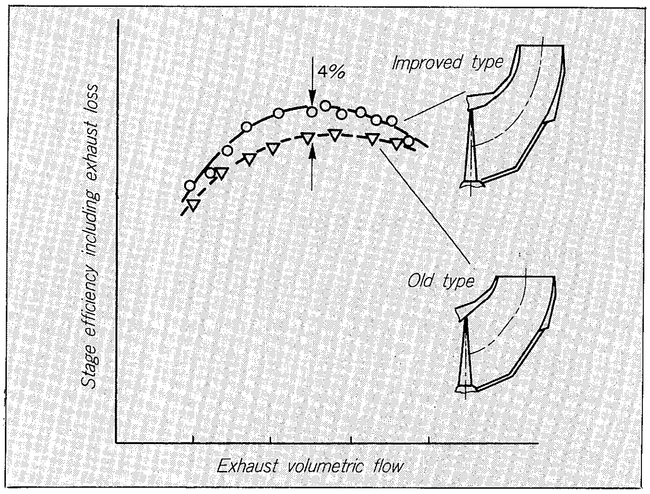


Fig. 7 Comparison of exhaust diffuser performance by model turbine test



have high reliability since caulking of tenon is not necessary and an excellent vibration damping effect can be obtained by applying a suitable prestress between the shrouds of adjacent blades during blade assembly.

(2) Application of three dimensional design to IP turbine blades

Since the reaction blade has a constant profile loss over a wide range of inlet angle, as mentioned in the preceding paragraph, a straight blade without any twist is usually used. However, depending on conditions, efficiency can be improved by using a twisted blade. *Fig. 5* is an example of application of three dimensional design applied to low pressure blades up to now to the first stage blade of IP turbine. The stationary blade is installed aslant and, together with the guide ring at inlet, enables to use velocity energy of flow effectively. The moving blade is a three dimensionally designed twisted blade. The surface temperature of the shaft is lowered and the leakage loss is minimized by selecting reaction degree lower than that of the other stages.

Efficiency of IP last stage can be also improved by using three dimensional design, when the blade is long.

(3) Development of high performance low pressure blade series using the transonic flow analysis method

Great progress has been made in low pressure blade analysis technology together with the increase in computer performance. In particular, a method of analyzing transonic flow by the time matching method has been made practicable recently so that the flow in the blade cascade can be accurately simulated. The effectiveness of this method is displayed in minimizing the profile losses. An example of analysis of a tip section by the three dimensional time marching method using a super computer is shown in *Fig. 6*.

The turbine exhaust loss can be reduced so that thermal efficiency is improved by using a long blade developed with the aid of such newest design techniques. For 50 Hz unit 41 inch blades are already in operation and longer 48 inch blade is being developed.

(4) Improvement of exhaust diffuser performance

Improving the diffuser is also important to reduce turbine exhaust loss. The new diffuser was developed to prevent flow separation at the turning part. Model tests and traverse test of an actual machine confirmed the performance improving effect. The measured results are shown in *Fig. 7*.

(5) Improvement of labyrinth seal construction

In order to seal blade end and gland, labyrinth type seal fins have been used for HP and IP turbines, while for

Fig. 8 Double fin sealing

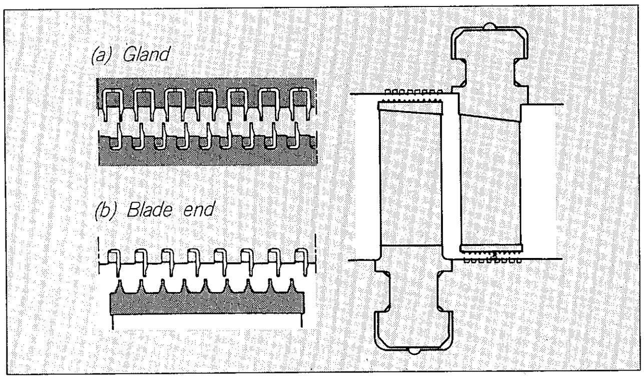


Table 1 Results of remodelling of aged turbines for efficiency improvement

Plant		A	B	C
Rated output	MW	175	175	350
First operation	–	1965	1968	1973
Turbine thermal efficiency (before remodelling)	%	43.59	43.80	44.29
Turbine thermal efficiency (after remodelling)	%	46.60	46.12	46.08
Improvement (relative value)	%	6.91	5.30	4.04

LP turbine, where differential elongation is large, straight type seal fins have been used. Results of a comparison study of seal fins of various forms by model tests showed that the leakage can be reduced by using double fin sealing shown in *Fig. 8*.

Another advantage of the double fin sealing is that even if rubbing should occur, it is limited to contact of fins and there is no metal contact with shaft or casing.

3.2 Remodelling of aged turbines for improving efficiency

The turbine efficiency improving measures described up to here can be applied not only to new units but also to aged units by remodelling so that considerable improvement of turbine efficiency is achieved. Results of remodelling of two 175 MW units and a 350 MW unit for efficiency improvement are shown in *Table 1*. The turbine thermal efficiencies were remarkably improved by 4.0 to 6.9% (relative values) compared with those before remodelling. In all cases, the turbine thermal efficiencies after remodelling exceeded even the valves of their initial performance tests by 1.6 to 2.5% (relative values), exhibiting satisfactory results of the development for improving efficiency.

4 IMPROVED TECHNOLOGY FOR CYCLIC OPERATION

Making the dimensions of each part as small as possible to minimize the temperature difference produced at each part of turbine is basic in securing ample reliability and life against quick and frequent start-up and load changes. The lighter and more compact a turbine, the easier it withstands sudden temperature changes. Newest examples of various technology used with Fuji Electric turbine facilities based on this basic concept are given below.

4.1 Independent HP- and IP-turbines (three component design)

The HP- and IP-turbines are separated and each turbine is designed to be light and compact using a barrel-type turbine at the high pressure part.

A 375 MW, 3600 rpm reheat turbine for cyclic operation under 250 kg/cm^2, 538/566 °C steam conditions is shown in *Fig. 1*.

4.2 Turbine shaft

When cyclic operation is considered, the diameter of the shaft is an important factor. Taking the blade design, critical speed and shaft deflection into account, ideally, the shaft diameter should be small and the span should be short. The HP-turbine shaft shown in *Fig. 9* has a 660 mm diameter and 3150 mm span. The thermal stress and axial and radial thermal expansion differences produced by the temperature change accompanying cyclic operations are very small.

As a rigid shaft, high temperature HP- and IP-turbine shafts prevent steam whirl and other unstable vibrations which are problems with high pressure, high output units. Since the critical speed is not passed at starting and stop-

Fig. 9 Sectional view of full-arc steam admission barrel-type HP-turbine

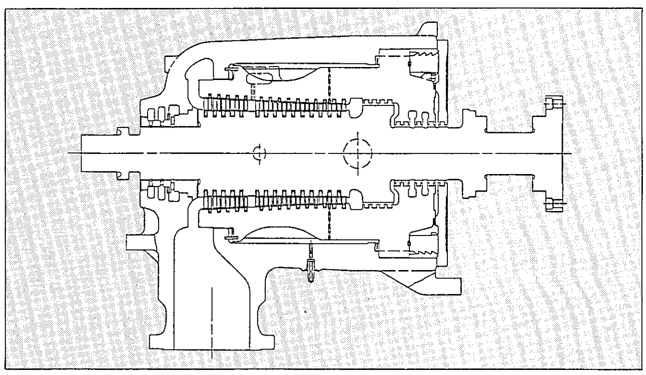
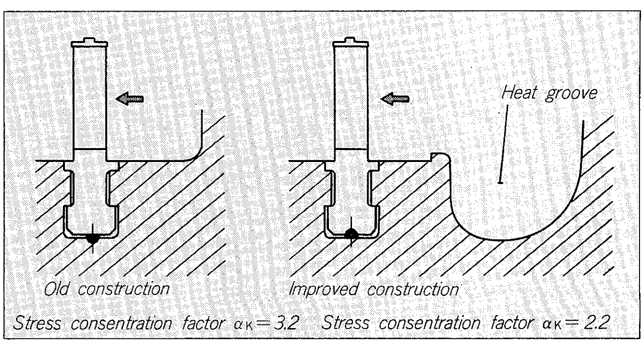


Fig. 10 Improvement of HP-turbine shaft shape



ping which receive sudden temperature changes, speed-up and load-up are easy without resonance peak vibration.

A heat groove is provided and bottom shape of blade groove is made flat at the inlet part of HP- and IP-shaft to reduce life expenditure of low cycle fatigue (*Fig. 10*).

4.3 Turbine casing

The HP-turbine casing is another important part from the point of view of stress and strength. The diameter of the shaft is small and the diameter of the casing can also be small. The HP-outer casing is the barrel type with roto-(symmetrical) cylinder and there is no large mass concentration without horizontal flange.

The inner casing has vertical flange but it can be designed also small by making the pressure distribution of inner- and outer-casing suitable. Therefore the thickness of the outer casing becomes much thinner and the diameter also becomes smaller. The outer casing diameter of *Fig. 9* is 1780 mm and the maximum thickness is 210 mm. The overall weight of the inner and outer casings and the shaft is 44.5 tons.

Because the sliding pressure operation has good partial load efficiency, small turbine internal temperature change accompanying load changes and other characteristics suitable for DSS operation, it is used with most new domestic units. Applying the full-arc steam admission barrel type turbine with excellent pressure resistance in sliding pressure operation, it can eliminate the governing stage and simplify the construction so that the thermal stress characteristic and inner efficiency are improved further.

4.4 Shaft support system

A stable shaft vibration characteristic is also an important factor in performing frequent starting and stopping smoothly. Besides making HP- and IP-shafts a rigid shaft design, the following constructions are used as standard to stabilize vibration characteristic and alignment of the shaft system:

(1) Single-point bearing support system

With this construction, only one bearing is provided between turbines. This improves the flexibility of the shaft system and receives less influence of alignment changes accompanying support system temperature changes. A comparison of the characteristics of the two-point bearing support system and single-point bearing support system is shown in *Fig. 11*.

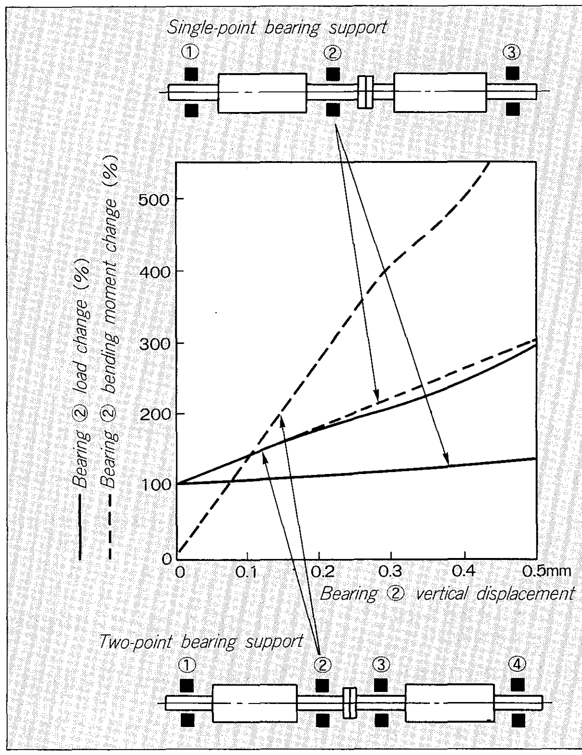
(2) Independent and separate bearing pedestal

With this construction all the bearing pedestals are independent and separate from the turbine casing and the bearing load point is on the foundation. With low pressure turbines, in particular, since the shaft end gland packing is also fastened to the bearing pedestal and the outer casing is connected by a bellows, the problem of vibration caused by outer casing temperature changes and deformation by vacuum force is essentially eliminated.

4.5 Improvement of material strength

(1) Shaft without axial through-bore

Fig. 11 Bearing support system and bearing load and bending moment at bearing



With large shafts which start and stop frequently, the shaft life (crack growth, brittle fracture) by the repetition of centrifugal force and thermal stress is an important topic. Recent improvement of shaft material forging technology and the accuracy of ultrasonic flaw detection from the outside of the shaft have made axial through boring for shaft center inspection accompanied by the risk of multiplying the shaft center stress unnecessary. According to fracture mechanics evaluation, when there is no axial through-bore, the critical crack size (a_{cr}) is approximately quintupled (*Fig. 12*).

Moreover, if the initial crack size is the same, the life (number of cycles) up to the generation of unstable fractures becomes longer by one or more digits. For these reasons, Fuji Electric has manufactured shafts without axial through-bore since the 1950s.

(2) Tougher heat treatment

Considering the life expenditure of the turbine main parts in high temperature region, material mechanical properties such as creep resistance, low cycle fatigue strength, fracture toughness, etc. are important, especially toughness properties are important for cyclic operated turbine material. Fuji Electric's shaft and casing material for high temperature use is so heat treated attaching importance to toughness properties, oil-quenched by making the austenitizing temperature lower (950 °C or lower for CrMoV) followed higher tempering temperature (680 °C or higher for CrMoV). Compared with material air-cooled from high temperature, it has excellent FATT, fracture toughness

Fig. 12 Rotating shaft stress and critical crack size

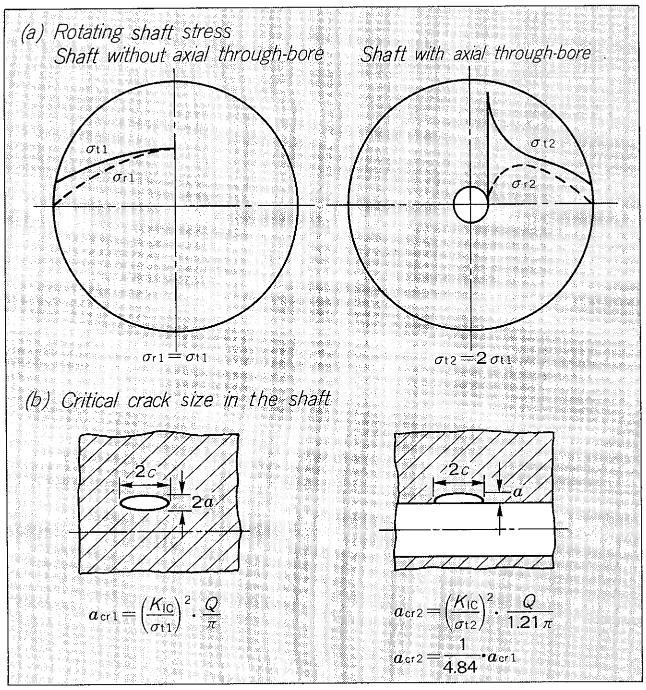
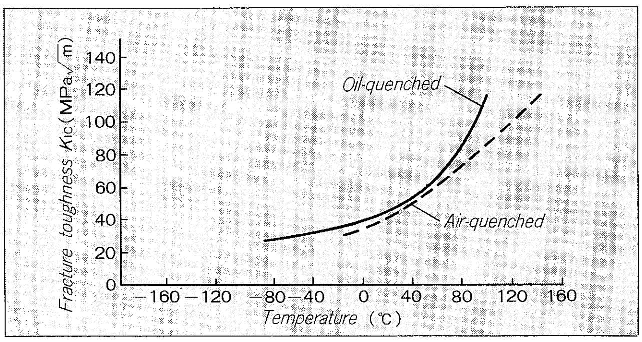


Fig. 13 1% CrMo(Ni)V shaft steel heat treatment and toughness



value (K_{IC}), and notch creep characteristics and is advantageous in suppressing long-term degradation and life expenditure (*Fig. 13*).

4.6 Automation and improvement of operability

(1) HP-LP turbine bypass system

With DSS operation units, boiler and turbine matching is more important. The boiler and turbine can be operated completely independently for the first time by building a complete turbine bypass system, from the boiler superheater outlet to the condenser through the reheater. This HP-LP turbine bypass system simplifies matching of the steam temperature and turbine metal temperature at start-up and shortens the start-up time. Shifting to isolated operation with only load of plant auxiliaries is possible even at coal-fired plants.

The bypass amount uses the entire boiler capacity more than ease of handling. Recently, however, simplification of the condensing facility equipment has been taken into account and it has become equivalent to the maximum condensate amount of the turbine (approximately 2/3 of the maximum boiler steam amount).

(2) Turbine stress evaluator (Wall temperature monitoring system)

A turbine must be operated while keeping the temperature difference of the main parts within the prescribed limits so that main parts of the turbine do not wear out during the planned operation period. The wall temperature monitoring system that performs this automatically simultaneously monitors the four points: (1) main steam valve casing, (2) HP-casing, (3) HP-shaft, (4) IP-shaft. Turbine speed up and load up and boiler steam temperature are controlled automatically so that the limit temperature difference set for the temperature state of each of the four

Fig. 14 Monitoring thermal stress with turbine stress evaluator

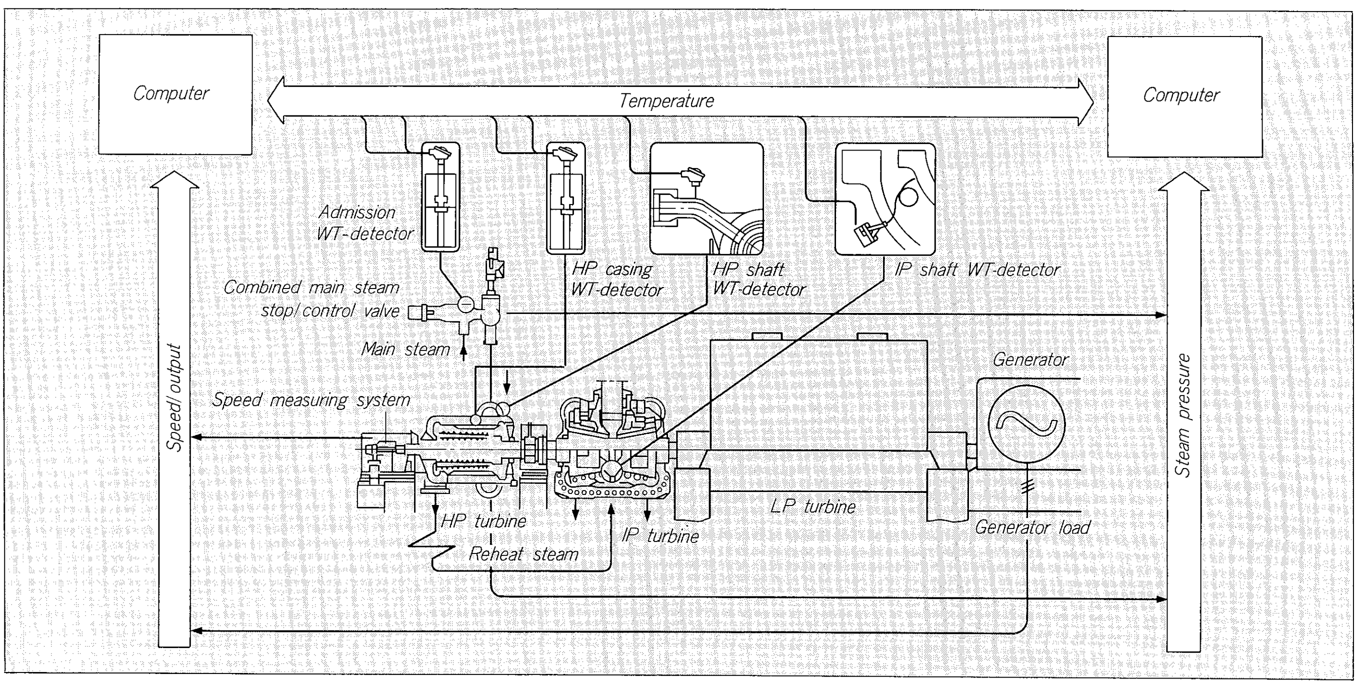
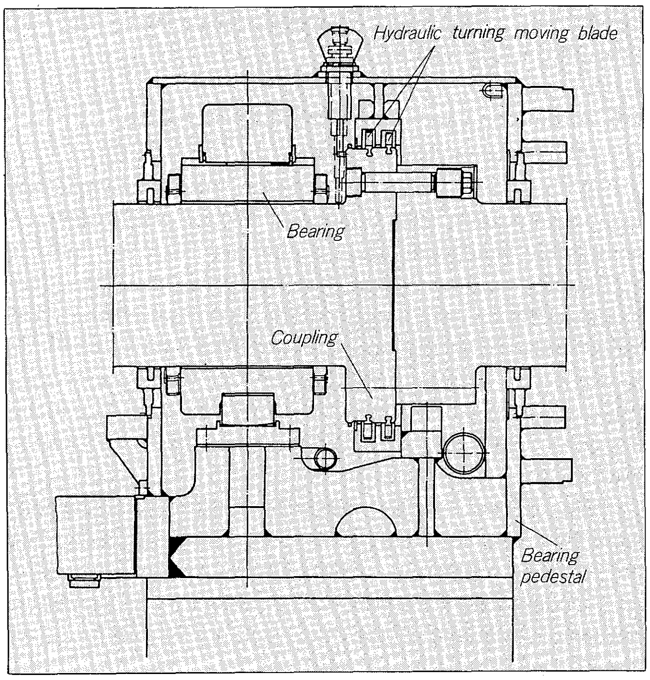


Fig. 15 Sectional view of blade wheel of hydraulic turning gear



points corresponding to the three modes of cold, warm, and hot is not exceeded (*Fig. 14*).

(3) High speed oil hydraulic turning gear

A construction which performs turing at a high speed of approximately 100 rpm by oil hydraulic turbine mounted directly to the coupling flange of the turbine shaft is used. Since turning is turned on and off only by opening and closing a motor oil supply valve, operation is simple and automation is easy. Since the bearing is fluid lubricated, the oil film is thick and complex oil temperature operation at turbine start-up is unnecessary. With the newest large units, the bearing is protected more reliably by continuously jacking up the shaft by an oil pressure of approximately 200 kg/cm^2 during the turning operation.

5 CONCLUSION

Generally, if turbine design is good, the start-up and load change characteristic of the unit are not restricted by the turbine. The latest state of HP- and IP-turbine independent construction with a barrel-type turbine at the high pressure section matched to the demands for higher pressure, higher temperature, and cyclic operation and other technologies used consistently by Fuji Electric from the past were outlined above.