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IMPROVEMENT OF STEAM TURBINE EFFICIENCY

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I. INTRODUCTION

The demand for electric power in Japan is expected to grow at an annual rate of 6%. In 1980, about 434.9 billion kWh of electric power was generated. This is expected to reach 589.1 billion kWh in 1985 and 779.5 billion kWh in 1990.(1) (Central Electric Power Council 1981 Long-Term Power Expansion Plant).

The gross thermal efficiency of central power stations has been improved substantially from about 20% during the past 30 years to about 40% at present. However, most of this improvement has been made at the cycle side, mainly by higher temperature and higher pressure initial steam conditions, the adoption of reheating cycle units, etc. Improvements made by raising the turbine internal efficiency have been no more than several percent.

Improvement of the turbine internal efficiency is an important subject, because it contributes directly to improve the plant efficiency. The internal efficiency of the steam turbine has been steadily improved during its history of 100 years, and its practical limit is being approached, even though further improvement may be possible at least theoretically.

Any future improvement in the turbine internal efficiency will be achieved through steady accumulation of research and evaluation of long-term operating experience.

In the power generation field, which consumes vast amount of energy, the trend is toward the adoption of still higher initial steam condition, the introduction of the combined cycle, and other measures to raise thermal efficiency from the cycle side for the purpose of saving resources and energy.

At the same time, more effort must be made in improving the internal efficiency of the steam turbine itself.

Theory of the steam turbine is not yet completed in some aspects, and it can be said that there is room for further improvement of the turbine internal efficiency from this viewpoint. Efficiency improving measures that have no merits in the past from the viewpoint of cost performance must be re-evaluated in the future. Progress in the technology of reliability may also make it possible to realize efficiency improving measures which have been regarded as impossible. According to the above mentioned forecast for power supply in Japan, more than 80% of the future power demand, remaining approximately 10% of water power plants, will be supplied by atomic power plants, oil, coal, and LNG-fired power plants, and geothermal plants. All these plants use steam turbines as the prime mover, and the mission of improving their efficiency is vital.

This report describes the measures Fuji Electric has taken to improve the turbine internal efficiency, the results of these measures, and future topics of study.

II. INTERNAL EFFICIENCY OF STEAM TURBINE

Fig. 1 shows the energy flow of a general 600 MW fossil Fuel power plant of the fuel energy entering the plant, 9.7% is lost in the boiler, about 44.3% is wasted from the condenser into the environment, even if there were no

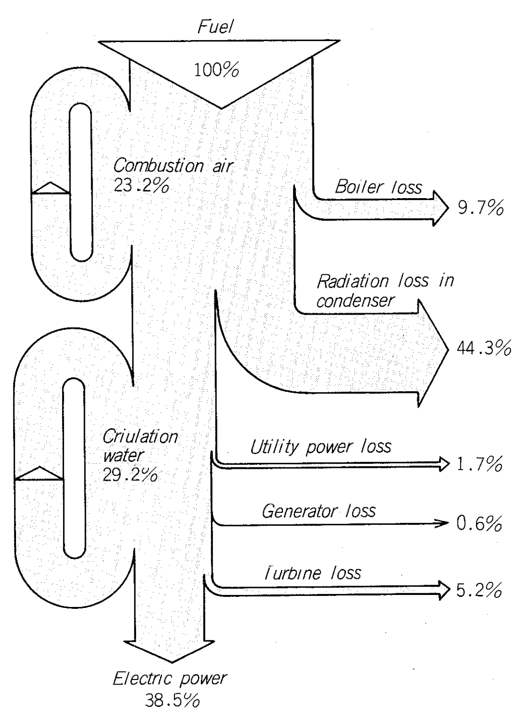


Fig. 1 Energy flow diagram for 600 MW thermal power plant

loss in the turbine, and about 5.2% is lost from the condenser by the loss generated in the turbine. Excluding the utility power loss and the generator loss, no more than about 38.5% of the energy entering the plant can be used as electric power. Improving of the turbine internal efficiency is nothing else than reducing this 5.2% loss.(2)

Fig. 2 shows the ratio of the internal losses generated in the high, intermediate, and low pressure turbines against the isentropic heat drops according to the main causes of these losses. The contents of these losses are described as follows.

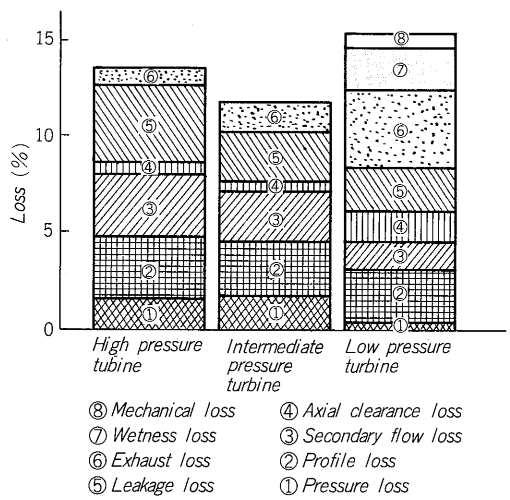


Fig. 2 **Turbine loss ratio against isentropic enthalpy drop**

1) *Pressure loss.* Pressure loss caused by inlet steam pipes and connecting pipes between the casings. For HP and IP turbines, losses in the inlet valves of main steam and reheat steam are included here.

2) *Profile loss.* Kinetic energy loss in the blade row with infinite height; that is, supposing the influence of the secondary flow at the blade ends (hub and shroud) to be negligible.

3) *Secondary flow loss.* This loss is caused by the secondary flow at the blade ends.

4) *Axial clearance loss.* This loss is caused by the axial blade clearance. It is an additional loss due to deviation of the axial clearance from the optimum design point to meet the differential thermal expansion of stationary part and rotating part.

5) *Leakage loss.* This loss is caused by the steam leaking from the radial clearance between the stationary and rotating parts.

6) *Exhaust loss.* This is the exhaust loss at the last stage of each turbine. The exhaust loss of the LP last stage, where the steam velocity is high, is especially large.

7) *Wetness loss.* This loss is caused by the wet steam in the LP turbine.

8) *Mechanical loss.* This is the bearing loss of each

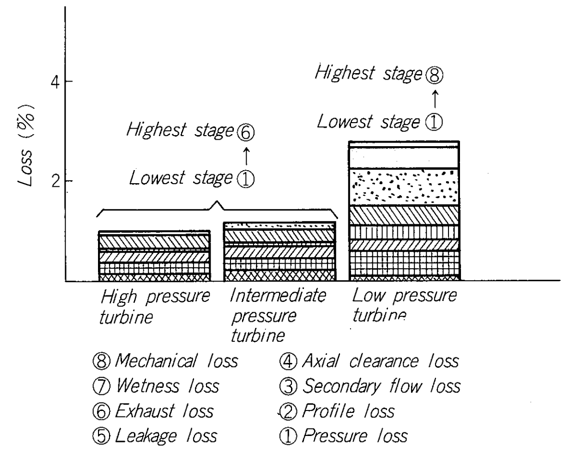


Fig. 3 **Turbine loss ratio against primary energy**

turbine and the power loss produced by driving oil pumps, and is included in the LP turbine loss for convenience' sake.

Fig. 3 shows these internal losses produced by each turbine as the ratio of loss to input fuel energy; that is, primary energy. This presents a quite different aspect. It is evident from the figure that the loss having the greatest effect on thermal efficiency is that produced by the LP turbine. This loss forms about 50% of the total turbine losses.

Therefore, it can be said that for improvement of turbine internal efficiency, emphasis must be placed on the LP turbine.

However, improving the efficiency of the LP turbine is a difficult and complex problem because of the large effect of the Mach number on the cascade losses, expansion of the flow path with rapid increase in the volume flow, the complexity of calculation required by three-dimensionality of the flow caused by this, or the effects of wet steam and so forth.

The main ratio of losses against primary energy shown in *Fig. 3* are:

(1) Pressure loss. Approximately 0.3% of the entire turbine.

(2) Profile loss. Approximately 1% for the entire turbine.

(3) Secondary loss. Approximately 0.7% for the entire turbine.

(4) Leakage loss. Approximately 1% for the entire turbine.

The following describes the possibility of reducing these losses further.

Let us begin with the pressure loss. The throttle loss of the valves, which forms a large part of this loss, can be reduced by lowering the flow velocity by using larger valves. However, considering the higher cost of the larger valves themselves and the increased building costs caused by the modification of the layout demanded by the increased size, there is an optimum limit for increasing the size of the valves. Although it may not be feasible from the cost/effect standpoint at the present time, increasing the size of the valves should become a more effective measure as the cost of primary energy rises in the future.

Next we take up the blade profile loss that forms about 1% of the total loss. As will be described in Section IV, the development of a new profile with lower loss and larger strength for HP and IP reaction stages is contributing to higher thermal efficiency. In the LP stages the complex transonic flow causes an additional loss due to the high Mach number, and the profile loss is large especially at the root and the tip. Furthermore, since the stress in the LP blade is extremely high, the ideal aerodynamic blade profile may not be obtainable because of strength limitation. These leave room for future improvement.

Next, the secondary flow loss will be described. To reduce this loss, the aspect ratio of the blade must be made large. That is, the chord length versus the necessary blade height must be made as small as possible. However, reducing the chord length increases the bending stress in the blade. Calculations show that the stage efficiency can be improved by 1% if a bending stress of double the present value is allowed. This results in the improvement of the thermal efficiency by 0.2%. (See *Fig. 4*). However, to raise the bending stress level, safety must be sufficiently proved. To do this, research and development are necessary. The measured vibration stress of the moving blades of an actually operating turbine is shown as an example of research to prove safety in *Fig. 5*. This figure shows the comparison of the measured vibration stress loaded on the blade with integrated shroud which has been used as the standard construction in Fuji Electric steam turbines, and the blade without shroud during operation. These results show that the vibration stress on the blade with integrated shroud is essentially small, remaining room for decreasing the blade chord and to reduce the secondary loss.

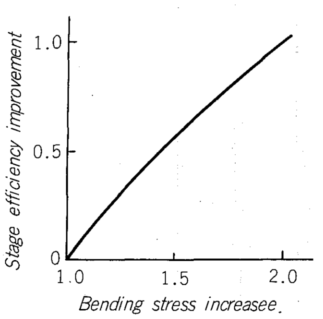


Fig. 4 Improvement of stage efficiency by increase of blade bending stress

The leakage loss, which forms about 1% of the primary energy, is described next. The leakage loss is the loss caused by the wasted steam leaking from the radial clearance between the stationary and rotating parts. To reduce this loss, the radial clearance must be made small and the number of sealing fins must be increased. However, the axial expansion difference limits an increase in the number of fin. Moreover, the radial clearance is determined by considering the

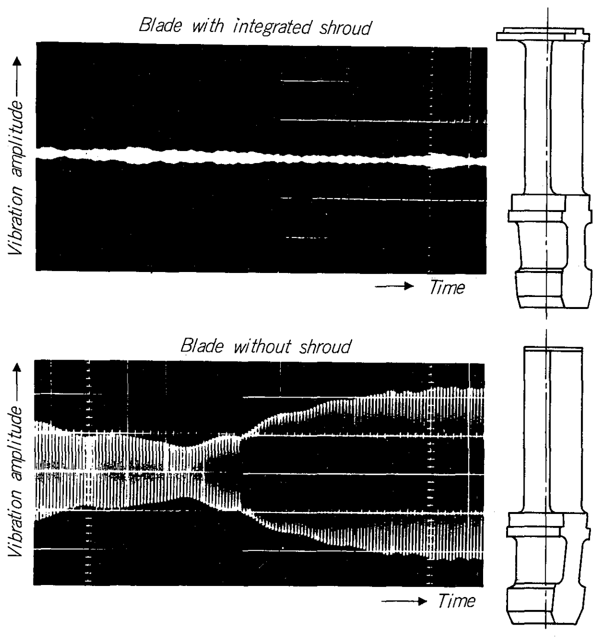


Fig. 5 Measured vibration stress of moving blade of actually running turbine

elongation caused by the centrifugal force of the rotating part, the thermal expansion due to the temperature difference between stationary and rotating parts in the steady and nonsteady state, deformation of the inner casing and outer casing, displacement of the bearing pedestals and casing supports, amplitude caused by rotor vibration, error during assembling, etc. However, if it can be proved that factors which reduce the radial clearance during operation do not occur simultaneously, it may be possible to reduce the radial clearance in the future.

To do this, data based on a wide range of records on reduction in the clearance under all operating conditions must be collected. The possibility of reducing losses for several loss factors were described above. Theoretical studies on the possibility of reducing losses are fairly well advanced; that is, it seems that theoretical possibility for loss reduction have already been exhausted, and conscientious judgement and costly research and development is necessary to make any further advances in loss reduction. However, trial calculations indicate that, in the future, accumulated research and development should permit a 1 to 1.5% improvement in thermal efficiency for the 600 MW turbine shown as an example. That is a 0.4 to 0.6% improvement in efficiency as a proportion of primary energy.

III. APPROACH TO EFFICIENCY IMPROVEMENT

1. Efficiency improvement measures

Almost a century has passed since de Laval and Parsons built the first steam turbines in the 1880's. Continuous technological development since then has led to the highly refined steam turbines featuring superior performance and reliability.[(3)]

With the advances made in analysis techniques accompanying the appearance of the large electronic computer, the internal efficiency of the steam turbine has been improved remarkably. Today, the internal efficiency of the HP turbine of large capacity units has reached nearly 90% and that of IP and LP turbines exceeds 90%.

As a result, to improve the internal efficiency of the turbine still further, attention must be focussed on all the loss factors and the possibility of all-around improvement connected to the reduction of each loss must be studied. The steep rise in fuel cost in recent years has increased the merit of the accumulation of improvements with comparatively small effect, to say nothing of those with large effect.

Against this background, systematic research and development were performed on improving the turbine internal efficiency as described below.[4]

First of all we picked up the possibility of reducing the internal losses as much as possible, based on the results of detailed analysis of internal losses. Concerning the proposed loss reduction measures of various kinds, the following were studied.

1) *Studies on the efficiency improving effect.* The effect on thermal efficiency achieved by the efficiency improving measures was estimated from the results of turbine internal loss analysis.

2) *Reliability studies.* Technology for improvement in turbine internal efficiency could not attain the final object to reduce power generating cost, if it caused deterioration in operating reliability. Therefore, the effects on reliability which adoption of efficiency improving measures may exert were conscientiously studied from all aspects.

3) *Economical studies.* The increase in facility cost and the reduction in fuel cost resulted from the implementation of efficiency improving measures was studied and economically evaluated.

An overall evaluation was made of the results of these studies and the feasibility of each improvement item was judged.

2. Main efficiency improvement measures

The efficiency improving measures finally adopted can be classified into the following two major groups.

1) *Measures based on existing technology and its extension.* Measures, that have not been adopted because of their lower improving effect, reduction in design margin that can be realized through advances in manufacturing technology and good operating results, scaling up of the turbine components and improvement of design techniques, are included here. The main improvement items are:

(1) Increase in the number of blade tip seal fins (reduction of leakage loss. (See *Fig. 6*).
(2) Reduction of radial clearance (reduction of leakage loss).
(3) Optimum design of blade row (reduction of cascade loss).
(4) Equal pitch construction of entire blade row (reduction of cascade loss).

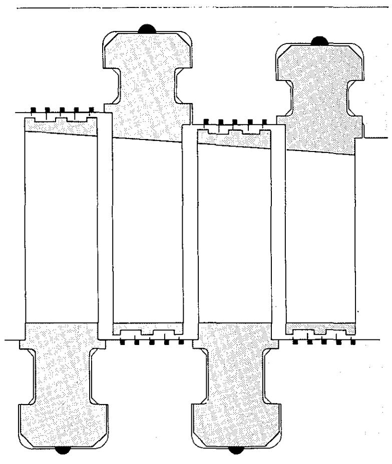


Fig. 6 Blade tip sealing

(5) Adoption of larger turbine modules (reduction of cascade loss).
(6) Increase in valve diameter (reduction of pressure loss).
(7) Improved design of steam inlet piping (reduction of pressure loss).
(8) Improved design of extraction slit (reduction of pressure loss).

2) *Measures requiring research and development.* Of the efficiency improving measures, development of a new blade profile, development of long LP blades, improvement of LP blade performance, improvement of the LP exhaust diffuser, etc. are especially important because of their great improving effect. Since these cannot be carried out by computational analysis alone, experimental researches in laboratory and at actual turbines are indispensable. Therefore, cascade tests in a wind tunnel, cascade tests in a multi-stage test turbine, model turbine tests, traverse tests in an actual turbines, etc. were performed, and the accumulated improvements based on the test results were applied to actual machines.

The main improvement items are:

(1) Development of a high efficiency reaction blade profile (reduction of cascade loss).
(2) Improvement of LP blade performance (reduction of cascade loss).
(3) Development of a large LP blade (reduction of exhaust loss).
(4) Improvement of LP exhaust diffuser (reduction of exhaust loss).
(5) Improvement of IP exhaust diffuser (reduction of exhaust loss).
(6) Development of high performance diffuser type valve (reduction of pressure loss).

3. Application to machine

The efficiency improving measures finally adopted after above-mentioned researches were first applied experimentally to a few machines. The improvement effect and reliability were fully proved there, and then the measures were applied to all machines.

IV. RESULTS OF TESTS AND RESEARCH FOR EFFICIENCY IMPROVEMENT

Main items of the turbine internal efficiency improving measures described in Section III, are treated below in outline.

1. Development of high efficiency reaction blade profile

In developing a turbine blade profile not only the aerodynamic performance, but also the blade strength must be considered. If the blade strength is low, a larger chord length would be required for the same stage load and, as a result, the number of stages which are possible to be mounted in the span limited by the rotor strength and critical speed would be reduced, and thus the effect of the improved blade profile would be canceled. For that reason, a new reaction blade profile (T4 profile) was developed on the precondition that the same strength of the conventional reaction blade profile (T2 profile) would be secured.

Compared to the conventional blade profile, the cascade performance of the new reaction blade profile is improved due to the following features (*Fig. 7*, *Fig. 8*).

(1) The growth of the boundary layer is suppressed and the profile loss is reduced by moderate velocity change on the profile suction side.

(2) Since the maximum flow velocity on the profile surface is reduced, the profile loss for the same blade load and surface roughness is small. Moreover, the loss is small even in the larger Mach range.

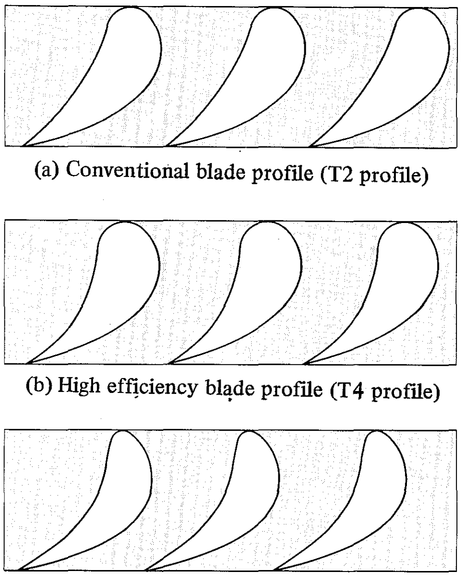
(3) The trailing edge loss is reduced by making the suction side (convex side) and pressure side (concave side) crossing angle at the trailing edge small.

To confirm the performances and cascade characteristics of the new blade profile, cascade tests in a wind tunnel, and single stage and multi-stage rotating cascade tests were performed.

Fig. 8 and *Fig. 9* show the results of the cascade tests in a wind tunnel. These figures show that the new blade profile improves the velocity distribution on the blade surface and reduces the profile loss over a wide range of outlet Mach number.

The results of the multi-stage rotating cascade tests proved that the new blade profile improved the cascade efficiency by 1 to 2% over a wide load range. *Fig. 10* shows the rotor for multi-stage rotating test and *Fig. 11* shows the test results.

A new reaction blade profile called H1 profile was also developed in parallel with the development of the T4 profile (*Fig. 7*). The performance of the H1 profile obtained by rotating cascade tests are shown in *Fig. 11*. The cascade efficiency at the best point is almost the same as that of the T4 profile. The efficiency in the high load range is better than that of the T4 profile. However, since the H1 profile has less strength than the T4 profile, its usage is restricted.



(a) Conventional blade profile (T2 profile)

(b) High efficiency blade profile (T4 profile)

(c) High efficiency blade profile (H1 profile)

Fig. 7 High efficiency reaction blade profile

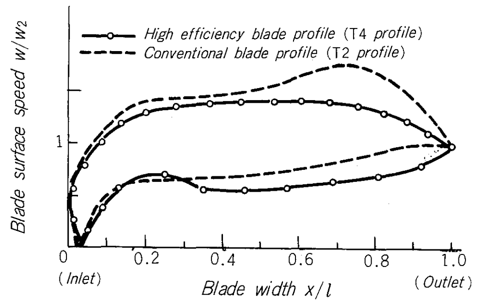


Fig. 8 Velocity distribution on blade profile surface

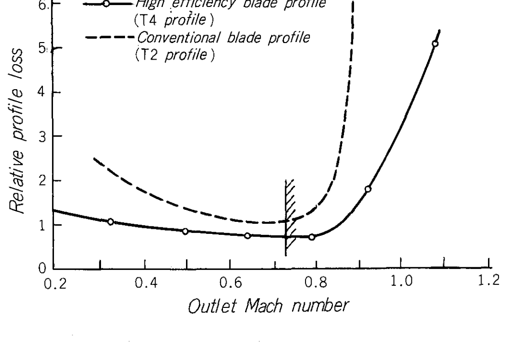


Fig. 9 Relation between outlet Mach number and profile loss

2. Development of long LP blades

The exhaust loss forms a larger part of turbine losses. In order to reduce this loss, a long LP blade which meets the larger turbine output must be developed. For developing a long LP blade, the aerodynamic performance, static and dynamic strength, natural frequency, erosion resistance

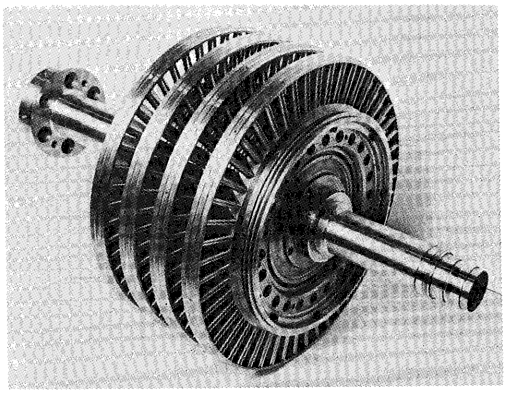


Fig. 10 Rotor for multi-stage rotating test

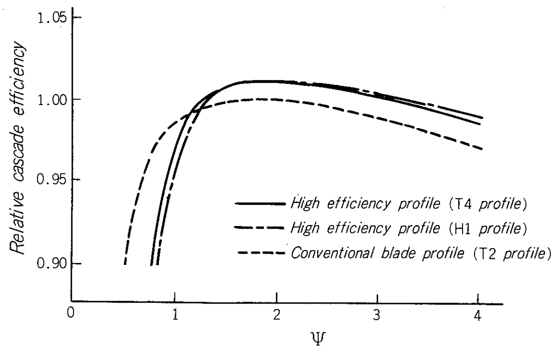


Fig. 11 Test results of stage efficiency

and many other problems must be solved. A 50 Hz 1050 mm (41.3 inches) blade and a 60 Hz 875 mm (34.4 inches) blade were developed after long-term research and testing, including transonic wind tunnel tests and model turbine tests. High reliability and excellent aerodynamic performance of these blades were confirmed by actual machine tests and long-term operation.[5]

The first machine using the 50 Hz 1050 mm blades was placed into operation in 1976. Since then the 1050 mm blades have been successfully operated in 16 flows of four machines, and proved to be the most established long blades of this class.

Fig. 13 shows the 1050 mm and 875 mm blades. *Fig. 14* shows the low pressure rotor using 1050 mm blades.

3. Improvement of LP blade performance

Development of the three-dimensional flow calculation method and high performance blade profiles suitable for transonic flow has improved the performances of the LP stages noticeably, and thus contributed to improvement of turbine efficiency. However, the results of LP model turbine tests and actual turbine traverse tests made it clear that there was room for improvement in the flow patterns around the tip of the last stage moving blades. The reasons for this are:

(1) Near the tip of the last stage moving blades, the high Mach number leaked steam through the blade tip wall

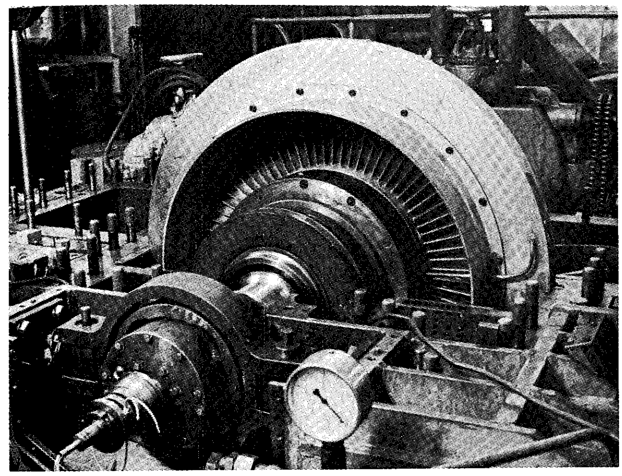


Fig. 12 Low pressure model turbine

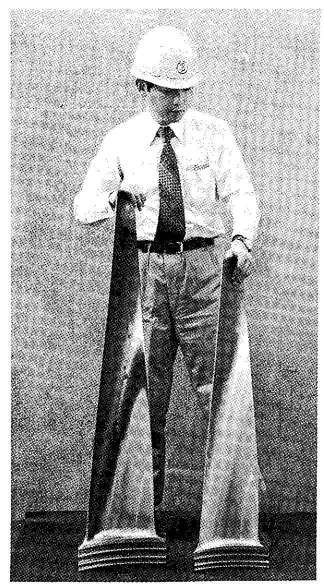


Fig. 13 50 Hz 1050 mm (41.3") and 60 Hz 875 mm (34.4") low pressure blades

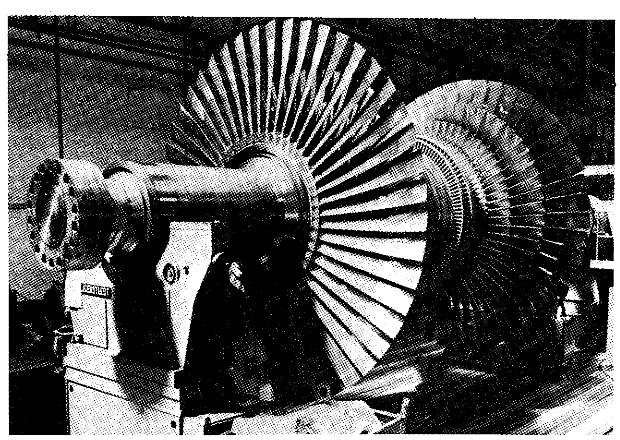


Fig. 14 Low pressure rotor with 1050 mm blades

clearance affects on the flow patterns inside the blade rows and lowers efficiency.[6] The effect of the blade tip leakage covered a fairly wide range, and the flow

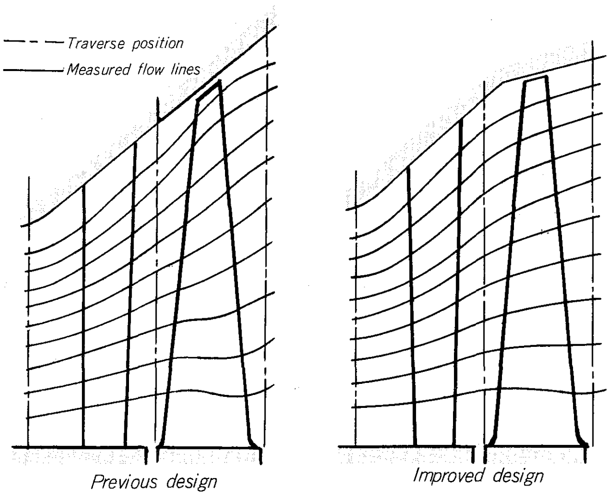


Fig. 15 Comparison of flow pattern of low pressure end stage

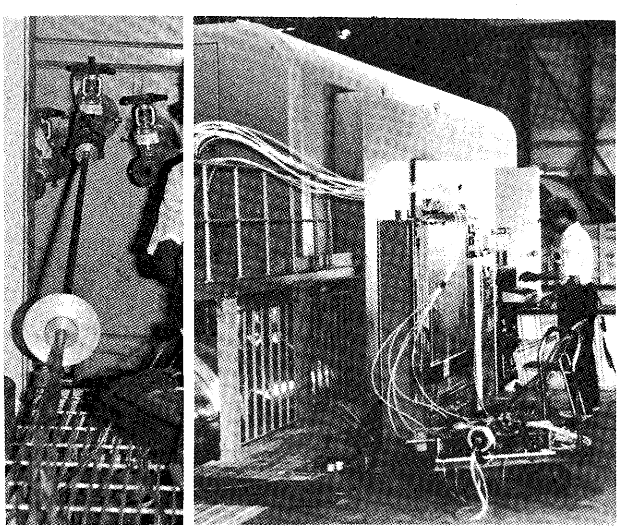


Fig. 16 Traverse test of low pressure stage at power station

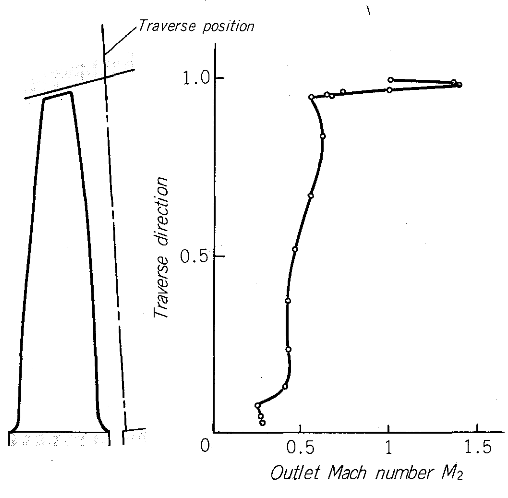


Fig. 17 Outlet velocity distribution of the last stage

trended to be deflected toward the outer wall. (*Fig. 15*).

(2) Near the tip of the moving blades, the radial velocity component of the flow was large and increased the exhaust loss.

To improve these, the shape of the last stage outer wall was changed and the flow patterns were modified to reduce the effects of the blade tip leakage.

Comparison tests with the LP model turbine confirmed that flow patterns of the last stage were improved considerably. (*Fig. 15*).

In order to examine the flow in the last stage under actual operating conditions, a traverse tests in an actual turbine was performed with a five hole yaw meter. *Fig. 16* shows the traverse test setup. Measurements were made at three points; before the last stage stationary blades, between the stationary and moving blades, and after the moving blades, and the pressure and flow velocity distribution under various operating conditions were studied.

Fig. 17 shows the measured outlet velocity distribution. It was confirmed that the range of the blade tip leakage effect was substantially reduced, and thus the exhaust loss was also decreased. The improvement of the LP turbine efficiency was as well proved by turbine performance measurements made at the same time.

4. Improvement of LP exhaust diffuser

The LP exhaust diffuser recovers a part of the kinetic energy of the steam discharged from the last stage moving blades as pressure, and thus increases the effective turbine heat drop and the power output. The following improvements were made to increase the pressure recovery effect of the exhaust diffuser:

(1) The diffuser channel was made longer than previous design.

(2) The diffuser was separated into two functional parts, namely the flow turning part and the pressure recovering part. At the flow turning part, the flow is accelerated by reducing the flow path to prevent flow separation at the suction side wall.

Figure 18 shows the improved shape of the flow path of the LP exhaust diffuser. The performance of the LP exhaust diffuser was checked by diffuser model tests in a wind tunnel and LP model turbine tests.

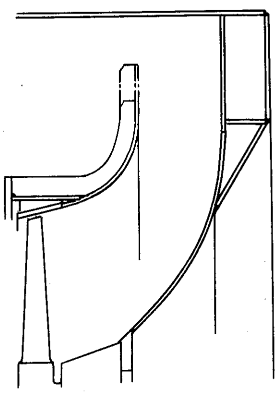
Figure 19 shows the exhaust diffuser test model.

The results of the LP model turbine test confirmed that the efficiency of the last stage was improved by 4 to 6%, owing to the reformed exhaust diffuser together with improved last stage described in Section III.

Fig. 20 shows the measured results.

5. Development of high performance diffuser valve

The pressure loss can be reduced by using diffuser type combined valves for HP inlet (main stop valve + control valve) and IP inlet (rehear stop valve + intercept valve). The optimum shape of these diffuser type combined valves was determined by theoretical analysis and full scale model performance tests. The adoption of the diffuser type valves



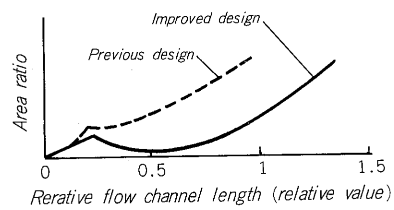


Fig. 18 Flow channel of low pressure exhaust diffuser

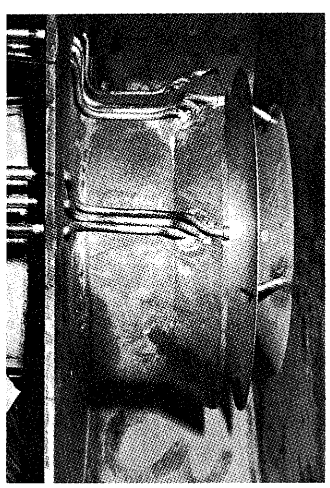


Fig. 19 Test model of exhaust diffuser

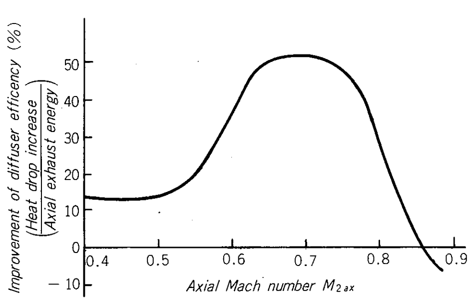


Fig. 20 Improvement of exhaust diffuser efficiency

improved the turbine internal efficiency by 0.1 to 0.4%.

V. ADVANCE IN EFFICIENCY

During this decade, continuous research and development has improved turbine internal efficiency by about 2% and more in terms of turbine heat rate. *Fig. 21* shows the advance made in heat rate by improvements in turbine efficiency.

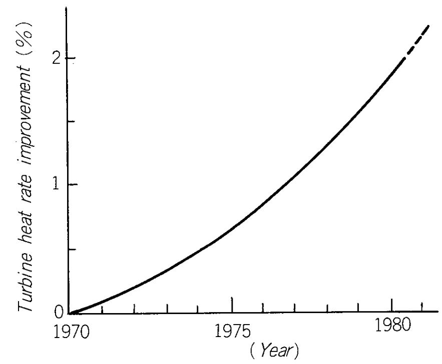


Fig. 21 Advance in heat rate due to improvement of turbine efficiency

VI. FUTURE TOPICS

1. Theoretical analysis and research and development

The use of theoretical analysis methods, as well as experiments, are indispensable in further improving of turbine internal efficiency. The blade-to-blade flow computational method, design method of three-dimensional flow, etc. have already made large contributions to improve performance. Advances in electronic computer technology in recent years have been amazing and made it possible to calculate the problems unsolvable in the past. For example, in the past, the only positive approach to transonic flow from the standpoint of design was experimental analysis using a shallow water table and a high speed cascade wind tunnel. However, the appearance of the Time-Marching method by which steady-state problems are calculated as a asynptotic solution of nonsteady-state problems has made practical calculation possible.

This trend is expected to grow in the future, and there is not a few fields in which further development of the fluid dynamic analysis techniques for the steam turbine is expected. Several of these problems are introduced here.

1) **Blade profile design.** The profile loss still occupies larger part of all turbine losses. Therefore, reducing the profile loss will continue to be one of the main problems in improving the efficiency of the steam turbine, so that the analysis method currently in use must be restudied or an analysis method that permits more accurate prediction of the cascade performances must be developed. Under

present conditions, the profile loss is finally found by cascade wind tunnel experiments. In these tests, various cascade models designed on the same cascade boundary conditions must be manufactured and measurements of pressure distribution along blade surface and pitot-traverse have to be carried out.

The development of a blade profile with minimum profile loss under the given conditions is a more acute problem for moving blades that must satisfy the opposing conditions; aerodynamic performance and strength.

To minimize the growth of the boundary-layer and to prevent separation near the trailing edge, a maximum flow velocity at the suction side that does not exceed the discharge velocity is ideal. However, this condition cannot be satisfied many times because of strength restrictions, as the maximum velocity at the suction side becomes larger, the profile is generally made thicker to maintain the strength. The optimum velocity distribution that minimizes the growth of the boundary layer at this time is difficult to evaluate. When the boundary layer becomes comparatively thick, a computational method[7] which considers the interference between the main flow and boundary-layer must be studied.

2) **Secondary flow.** At the hub and shroud ends of the stationary blades, the effect of the boundary layer unbalances the centrifugal force and pressure difference to produce flow from the pressure side to the suction side. Since the blade row is arranged cylindrically, a pressure difference balanced with the centrifugal force is also produced in the radial direction. However, as the pressure difference is larger at the boundary-layer on the suction surface, low speed fluid flow is concentrated at the hub, as shown in *Fig. 22*. Here, the total pressure is low; that is, the loss is large. Further, deflection of the exit flow angle against the main flow also forms the flow patterns different from the design.

There are two main methods of calculating the secondary flow. One distributes the vorticies in the inviscid flow field and the other calculates the three-dimensional boundary-layer on the hab and shroud ends. The former is yet in the laboratorial phase. On the other hand, in the case that the aspect ratio is small, as in the HP and IP stages, and the secondary flow effects the main flow strongly, the precision of the latter method decreases.

3) **Wet steam.** In the fossil fuel steam turbines, the steam crosses the saturation line, and expands to become wet at the last stages. From the standpoint of efficiency, problems of the wet steam are; 1) the behaviour of actual flow differs from the analytical results of such as three-dimensional flow which assumed one-phase fluid flow, and inter-blade flow, because of water and air two-phase flow with relative motion between condensed water droplets and steam, 2) decreasing of the efficiency at the LP turbines caused by wetness loss. The sources of wetness loss are; 1) thermodynamic loss during condensation, and 2) dynamic loss caused by the water droplets produced by condensation. Since the expansion of the steam in a steam turbine is fast, expansion continues without condensation immediately after crossing the saturation line, and supercooling proceeds untill ultimately steam pressure, velocity, density and other parameters abruptly change with the condensation shock. The diameter of the water droplets produced at this time depends on degree of supercooling. Therefore, theoretical prediction of the position in the i-s diagram and in the turbine at which condensation shock occurs is extremely important.

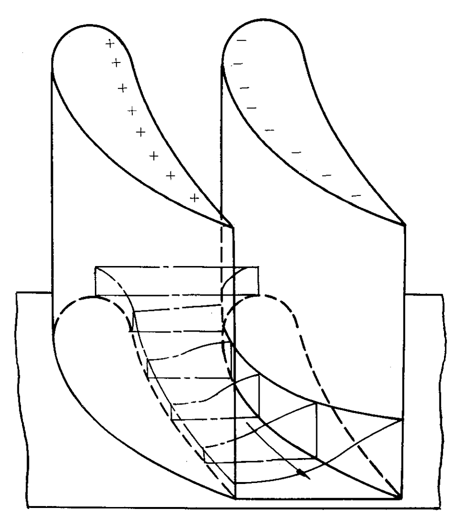


Fig. 22 Boundary-layer thickness distribution generated by secondary flow[2]

4) **Three-dimensional design.** In the three-dimensional design method, the blade-to-blade flow and hub-to-tip flow are calculated. In the hab-to-tip flow calculation, the stream line curvature method is widely used because of its good calculation efficiency. In this method, the flow density distribution is given and the position of the streamline is calculated. The radial momentum equation is then solved from the static pressure on the mean stream line, and correcting the stream line location this procedure is continued until the static pressure reaches to a constant value.

The blade-to-blade flow is calculated from the inlet and outlet flow angles, Mach number, Reynolds number, etc. found from the hub-to-tip flow calculation. This method assumes an axisymmetric flow and as is not three-dimensional in the strict sense. For three-dimensional flow computation corresponding to the actual flow, calculation planes shown in *Figure 23* for example, must be used, and calculation must proceed, so that the solution of surface S1 and the solution for surface S2 coincide at the intersecting point. However, since genrally the S1 plane does not exist on a circumference at the trailing edge, the periodicity condition cannot be used and, furthermore, a vortex sheet sheds from the discontinuous plane. Recently, a number of methods have been used to analyse this problem. However, the wake models used in the methods are not general enough to be applied to every case of practical use. Therefore, it is thought that rather simple improvements considering the circumferential gradient term in the radial-equilibrium equation, for instance, should be studied as the present subjects.

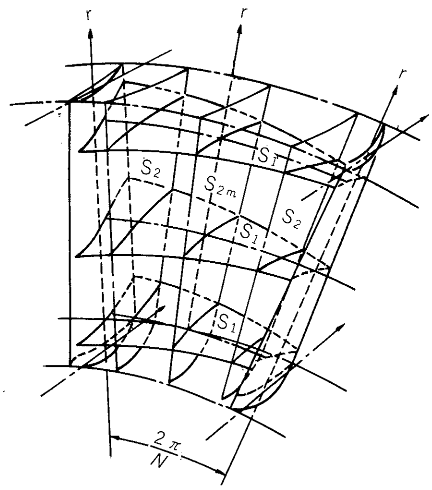


Fig. 23 S1 and S2 surface for computing three-dimensional flow(8)

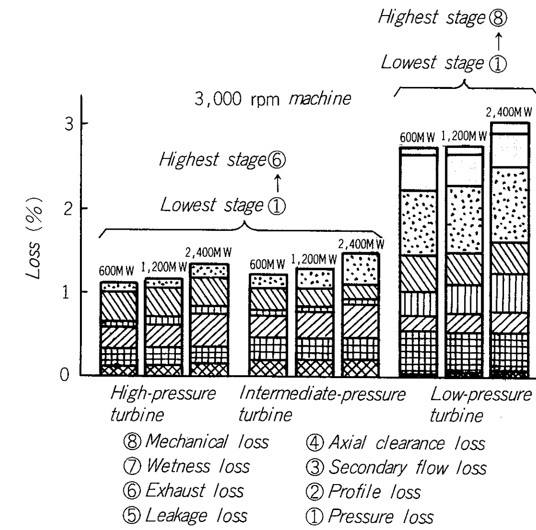


Fig. 24 Turbine loss ratio against primary energy

2. Introduction of large capacity 4-pole machines

In the case of a conventional 2-pole machine, according to the increase in the unit output turbine internal efficiency gradually rises because the volumetric flow increases. However, if the unit output increases further, the turbine internal efficiency begins to drop because of the size limitation imposed on the steam turbines by materieal strength and rotor vibration restriction. This drop of internal efficiency is caused mainly by; (1) increase in the secondary flow loss of the HP and IP turbine blade rows, and (2) increase in the IP and LP turbine exhaust loss. The LP turbine exhust loss, which has the greatest effect of these, can be reduced by increasing the number of exhausts. However, the increase in the number of exhausts is limited from the standpoints of safety and reliability, because it brings increase in the number of casings.

Fig. 24 shows the calculated internal loss of an IP and a LP turbine for an unit capacity of 600 MW, 1200 MW and 2400 MW designed as a 2-pole machine. The gradual increase in the LP turbine axial clearance loss is caused by the increase in the axial clearance between stationary and moving blade rows required by increased number of casings.

One method of suppressing an increase in the exhaust loss without increasing the number of exhausts, that is, the number of exhaust casings, is the adoption of a cross-compound unit with a 4-pole generator for the LP turbine only. This is becoming a standard model for 3000 rpm, 1000 MW class units. However, since the HP turbine remains as a 2-pole machine, it limits the loss reduction for the reasons described above. On the other hand, making all HP, IP, and LP turbines as 4-pole machines have the following advantages:

(1) Since the peripheral velocity and steam velocity in the HP and IP turbines can be suppressed to a low value (for a 1000 MW class machine, the increase in the mean blade diameter for a 1500 rpm machine does not exceed 40 to 50% that of a 3000 rpm machine), the aspect ratio of the

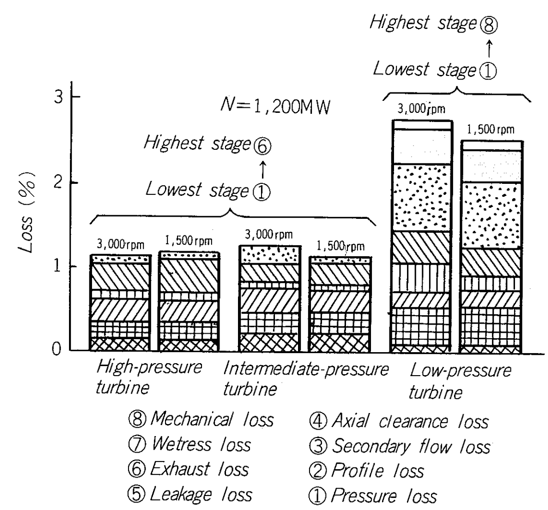


Fig. 25 Turbine loss ratio against primary energy for 2 pole machine and 4 pole machine

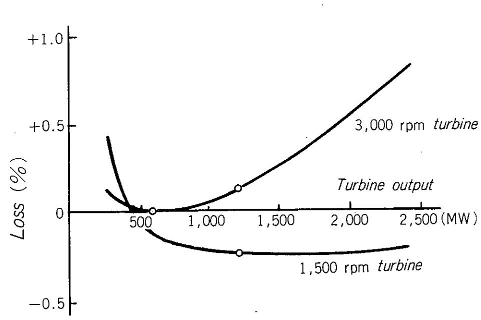


Fig. 26 Variation of turbine loss for 3000 rpm and 1500 rpm turbines against primary energy

blades increases and the secondary loss of the blade row decreases. The increase in the annulus area also reduces the exhaust loss substantially.

(2) The increase in the exhaust area results in reduction of the number of LP casings and the axial clearance loss is

substantially reduced.

Fig. 25 shows the turbine loss ratio against primary energy for a 1200 MW machine. The loss of the 1500 rpm machine is about 0.4% smaller than that of a 3000 rpm machine. In other words, 0.4% of the primary energy is obtained as an increase in output.

Fig. 26 compares the change in the turbine internal losses of a 2-pole and a 4-pole machine (3000 rpm and 1500 rpm) in the 300 to 2400 MW output range. The comparison base is a 3000 rpm, 600 MW machine.

For a 3000 rpm machine, the loss begins to increase gradually at 600 MW and increases sharply when 1000 MW is exeeded. For a 1500 rpm machine, the loss decreases as the capacity increases up to about 1200 MW, then becomes flat up to 2400 MW. At 1200 MW, the difference of the loss between both machines is about 0.4%, as previously described. Although details are omitted here because of space limitations, the 4-pole HP and IP turbines also bring smaller centrifugal force at the center of the rotor than that of a 2-pole machine. Therefore, the margin for creep at the center of the rotor in the high temperature range, where the thermal stress is high is enlarged, and the reliability is improved substantially. As described above, making the HP, IP, and LP turbines as 4-pole machines can reduce the internal losses compared with the conventional 2-pole machines and contributes substantially to improve thermal efficiency, if the unit capacity continues to increase in the future.

VII. CONCLUSION

This report describes the importance of improving the internal efficiency of the steam turbine, R and D made for this purpose, some of the results, and future topics. Improvement of the turbine internal efficiency has steadily advanced along with the history of growth of the steam turbine. However, it seems that the first oil crisis in 1973 accelerated efforts toward importance of improving the internal efficiency as well as the reliability should continue to increase in the future.

On the other hand, the possibility of improving turbine internal efficiency is gradually approaching to the limit, and further improvements will be extremely difficult. However, as primary energy costs rise in the future, new turns will be made in improving efficiency. That is, re-evaluation of improvement measures overlooked in the past because of cost considerations may become necessary. Therefore, further research and development must be promoted and, at the same time, the systematic collection of a wide range of test results based on operation experience is indispensable. The manufacturer must, of course, makes efforts to achieve this, but understanding and cooperation of users is also necessary. Although not touched on in this report because it is outside its objective, improvement of the efficiency of other components of plants is also a large topic.

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