PERFORMANCE IMPROVEMENT OF MECHANICAL DRIVE STEAM TURBINES

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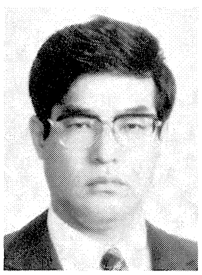

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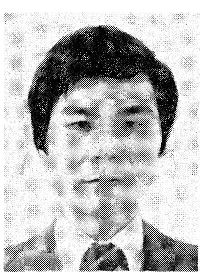

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ABSTRACT

Recently, the importance of mechanical drive steam turbine efficiency has been much highlighted in addition to its reliability with drastic increase in fuel cost. In this paper the typical losses in steam turbines are reviewed first, and then the technology to improve turbine performance by decreasing (1) exhaust loss and (2) highly distributed stage losses is discussed.

Also, possible efficiency gains are described briefly. Some examples of modification of existing turbines for energy conservation are given and, finally, shop performance tests are introduced.

INTRODUCTION

In keeping with the expansion of petro-chemical industry in recent years, driving turbines for various types of compressors have been developed. High reliability is required of mechanical drive turbines because they are important machines in plants, operated at variable speeds. Additionally because verification of their performance is difficult unlike electric generator drive turbines, only reliability has been stressed up to this time, and performance has not been valued highly.

With the increase in oil prices, however, energy-saving has become an important theme in petro-chemical industry, too. Especially in non-oil producing countries like Japan, energy costs have risen remarkably as shown in Figure 1, and a drastic improvement in turbine efficiency has been demanded. Figure 2 shows a comparison, before and after the oil crisis, of the plant cost which is the sum of the turbine initial cost and running cost. It is seen that the optimum economical value for turbine efficiency has become remarkably high.

In power plants, high temperature, high pressure steam conditions and high vacuum for condensers have been used as a means to improve thermal efficiency. Regenerative cycle and reheat cycle have also been introduced. The same means have naturally been studied for chemical plants, too, and high temperature, high pressure steam conditions and regenerative cycles have been contemplated for driving turbine cycles in recent LNG plants.

However, for chemical plants which are planned based on a proven process, much improvement in thermal efficiency by improved steam condition, reheating and regeneration cannot be expected. Therefore, improvement in thermal efficiency is to be made in a large measure by an improvement in turbine efficiency.

This paper introduces an outline of energy loss in the turbine and explains the transition of turbine efficiency, technologies for turbine efficiency improvement, verification of

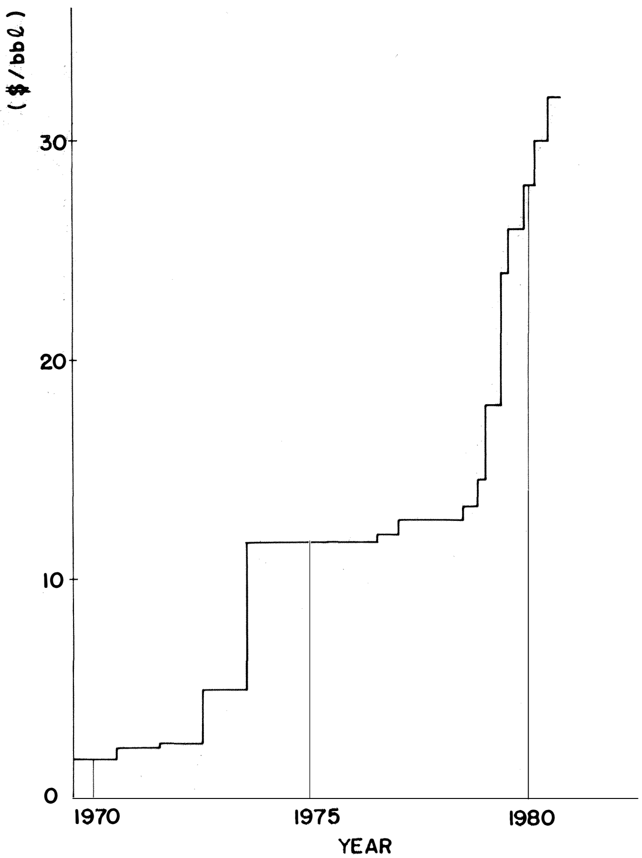

Figure 1. Price of Crude Oil.

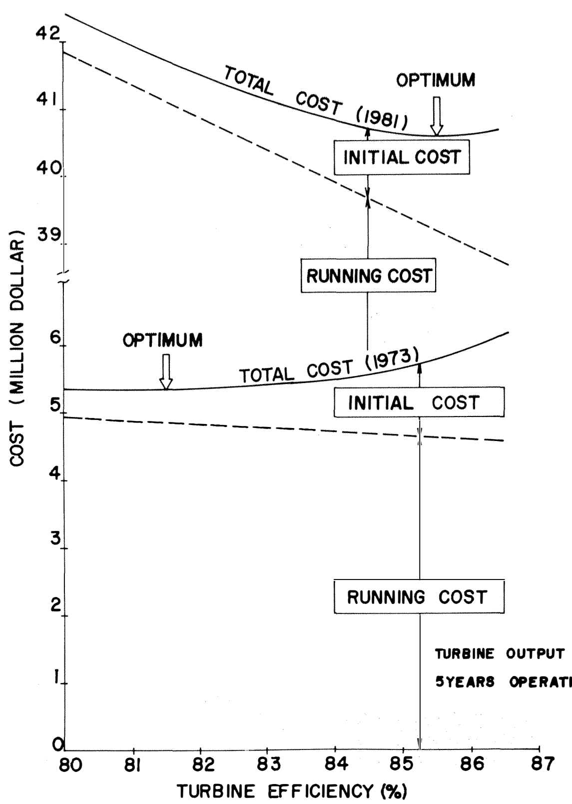

Figure 2. Effect of Turbine Efficiency on Plant Cost (Comparison Between 1973 and 1981).

turbine reliability and examples of energy-saving modifications of installed turbines.

TREND OF TURBINE EFFICIENCY

Steam condition, speed and output of mechanical drive steam turbines vary widely. So it is necessary to introduce a parameter which governs the efficiency if the transition of turbine efficiency over the years is to be recognized.

The efficiency parameter defined by the formula shown in Figure 3 is used.

Turbine efficiency must show a linear relation to the efficiency parameter, assuming that isentropic velocity ratio, profile loss, moisture loss and exhaust loss are constant. Experienced turbine efficiency is plotted against the efficiency parameter in Figure 3.

From Figure 3, it is determined that turbine efficiency saturates at a low value of the efficiency parameter before 1975 because of the larger volumetric exhaust flow and increased exhaust losses. After 1975, turbine efficiency was improved as shown in Figure 3 by application of longer blades and a decrease in stage losses. The technologies applied to improve turbine efficiency will be presented in detail later.

LOSSES IN STEAM TURBINES

To find the effective means for improving the turbine efficiency, it is necessary to know the loss distribution inside the turbine. Figure 4 shows an example of loss analysis of a condensing turbine. It is seen from this figure that stage losses and exhaust loss account for a greater part of the total loss. Exhaust loss is the sum of the leaving loss in the last stage and hood loss. Stage losses can be divided into; (1) profile loss, (2) secondary flow loss, (3) blade tip leakage loss, (4) diaphragm leakage loss, (5) partial admission loss, (6) disc and shroud friction loss, (7) moisture loss and (8) leaving loss. Main losses in stage losses are outlined below.

Profile loss

Profile loss is the sum of friction loss in the development of the blade surface boundary layer and mixing loss due to blade exit edge thickness. Since the thickness of boundary layer is influenced by pressure distribution in the blade path, it is an important point to design the velocity distribution on the blade suction side so as to have a uniform acceleration in order to reduce profile loss. Further, the incidence loss, which occurs when the flow mismatches the blade, is included in the profile loss. Generally, profile loss can be obtained experimentally by two-dimensional cascade tests.

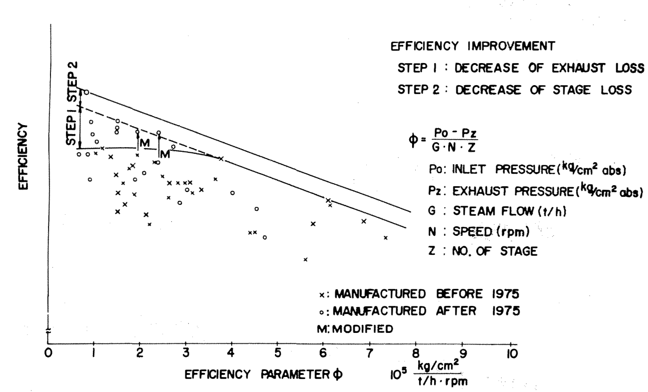

Figure 3. Trend of Turbine Efficiency.

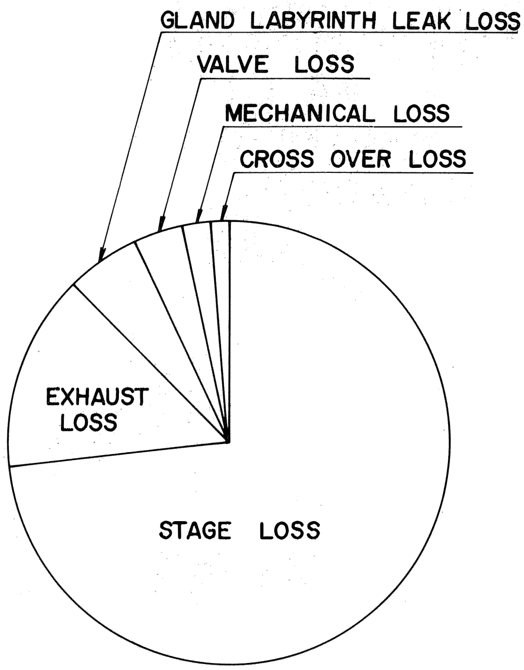

Figure 4. Loss Analysis of a Condensing Turbine.

Secondary flow loss

Secondary flow loss is produced by an unbalance between the pressure gradient, due to the turning of the main stream in the blade passage and the centrifugal force inside the end wall boundary layer. This relates to blade profile, pitch, wall boundary shape, aspect ratio (blade height/chord) and turning angle.

It is obtained by cascade tests or turbine traverse tests, and is greatly influenced by aspect ratio and turning angle. To reduce secondary flow loss, reducing the turning angle of steam flow is effective.

Leakage loss

Leakage loss is caused by leakage from the diaphragm labyrinth and blade tip shroud, and the amount of leakage steam flow is determined by leakage area, labyrinth shape and pressure difference before and after the seal.

Moisture loss

Moisture loss is produced when the stage operates in wet steam, and can be classified into supersaturation loss, condensation shock loss, water droplet acceleration loss, braking loss due to water droplets impinging upon blade suction side. Moisture loss is generally expressed experimentally with moisture degree as parameter. As a means to reduce moisture loss in low pressure stages, it is fundamental to reduce moisture degree in each stage by drain removal. So discharging of harmful coarse droplets by optimized design of the drain catcher is effective. Figure 5 shows an example of stage loss analysis in a typical condensing turbine.

The abscissa shows the theoretical output ratio of each stage, and the ordinate shows each loss ratio. In this figure, therefore, the area of each loss corresponds to the absolute value of each loss. The following points have been made clear by the loss analysis.

1. Loss is largest at the first stage (control stage) and the last stage, and is smallest in intermediate stages.

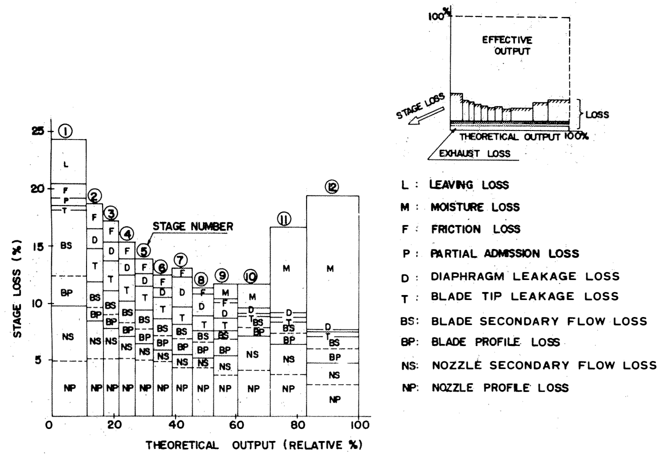

Figure 5. Stage Loss Distribution of a Condensing Turbine.

2. Nozzle profile loss is relatively large.
3. In high pressure stages, secondary flow loss and blade tip leakage loss are large.
4. In low pressure stage, moisture loss is relatively large.

Therefore, it is important to reduce exhaust loss and stage loss,which accounts for a large portion of the losses, in order to improve the efficiency of condensing turbines.

IMPROVEMENT OF TURBINE EFFICIENCY

As a technology to improve steam turbine efficiency, a method to decrease the last stage leaving loss and exhaust hood loss is explained first, and then a method to improve the performance of high pressure and low pressure stages is introduced. Finally, typical overall turbine efficiency gains obtained by these methods are shown.

Decrease in the exhaust loss

It is effective to increase the exhaust annulus area at the last stage blade exit to decrease the last stage leaving loss. However, the increase in the exhaust annulus area is restricted by the strength of the moving blade when single flow construction is applied. Figure 6 shows the relationship between the exhaust annulus area and maximum allowable speed of existing long blades. In this figure, the solid line shows the limit for the conventional stainless steel blade material. The new blade material such as titaninum alloy and maraging steel, has already been developed and applied in order to increase the exhaust annulus area especially for high speed machines.

Additionally, endless grouped blading has been applied since 1976 in order to extinguish the resonant vibration stress, which is being operated with good success.

The principle of endless grouped blading is shown in Figure 7. Many kinds of tests have been carried out to establish the design of the blading. Figure 8 shows the twist back test facility used to confirm the relation between the turbine speed and the twist back angle due to centrifugal force. Due to the application of the new technology, the longer blades can be applied for higher speed with higher reliability.

Single flow construction is preferable because of its simple construction and easy maintenance.

However, double flow construction is also applied when its efficiency is superior to single flow construction, after the precise evaluation of leaving loss, crossover loss and exhaust

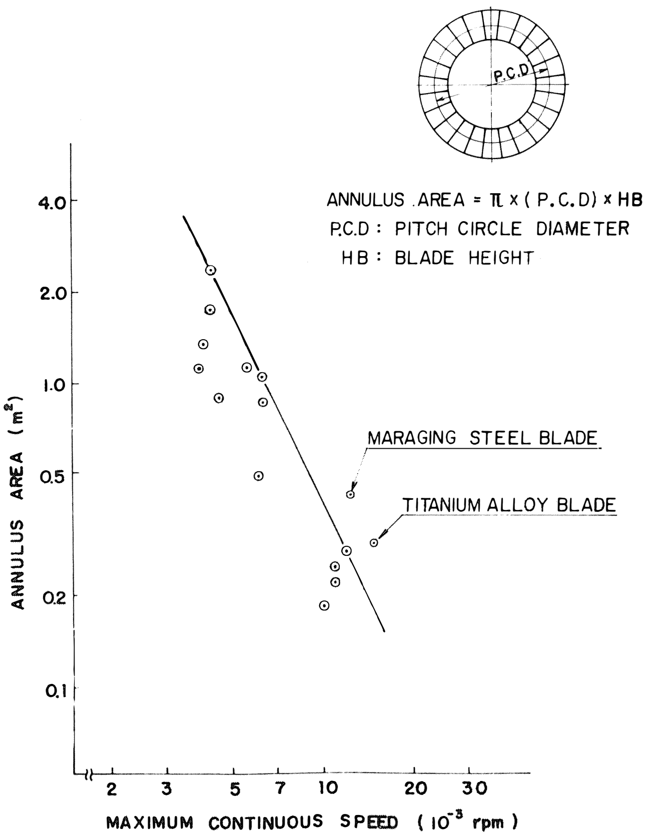

Figure 6. Applicable Long Blade (Relation Between Allowable Maximum Speed and Annulus Area).

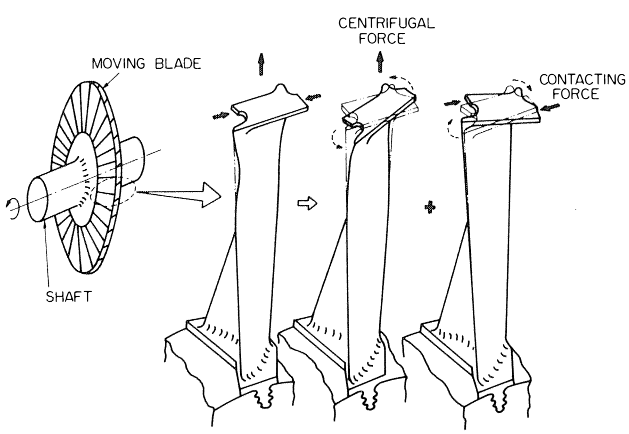

Figure 7. Principle of Endless Grouped Blading.

loss. Figure 9 shows an example of double flow construction applied to the ammonia synthesis gas compressor driver. For this high speed machine, the highest performance consistent with reliability has been achieved.

Namely, the flow path and internal struts are designed to minimize crossover loss and exhaust hood loss within the limit of bearing span because those losses are apt to increase in shorter axial space.

In order to design an optimum exhaust hood, model tests have been performed to investigate the effect of various geometrical parameters such as height, axial length, width and shapes and number of internal struts.

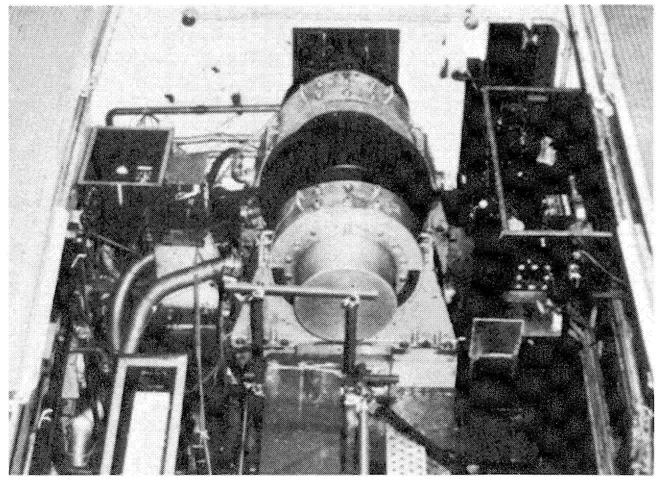

Figure 8. Twist Back Test Facility.

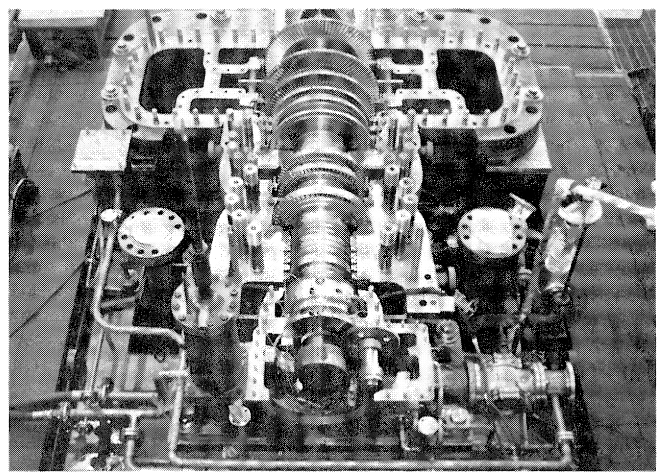

Figure 9. The Example of Double Flow Construction Applied for Synthesis Gas Compressor Driver.

The test results have been used as design data for calculations of turbine performance.

In order to reduce blade profile loss and secondary flow loss in selecting the nozzle and blade, the profile with a slightly large exit flow angle is adopted.

Performance Improvement of High Pressure Stages

Improvement of nozzle and blade profile and the reduction of leakage losses are effective for improving high pressure stage performance.

Improvement of nozzle and blade profile

Nozzle and blade performance are improved by conducting cascade flow analyses and cascade tests in low speed and high speed wind tunnels.

Figure 10 shows comparison of new Rateau nozzle and blade profiles and conventional ones.

Figure 11 shows the velocity distribution in the nozzle passage. As seen from the figure, steam flow acceleration on the suction side of the new nozzle is achieved more smoothly than on the conventional nozzle; development of boundary layer is controlled, and as a result, profile loss is smaller.

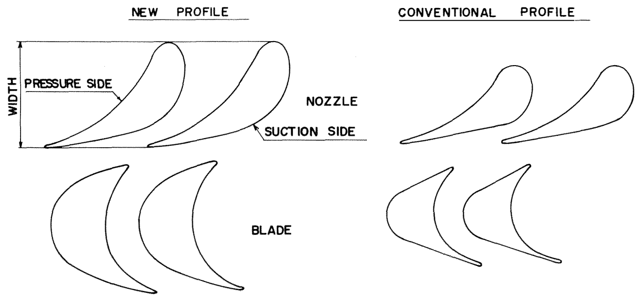

Figure 10. Comparison of Rateau Nozzle and Blade Profiles.

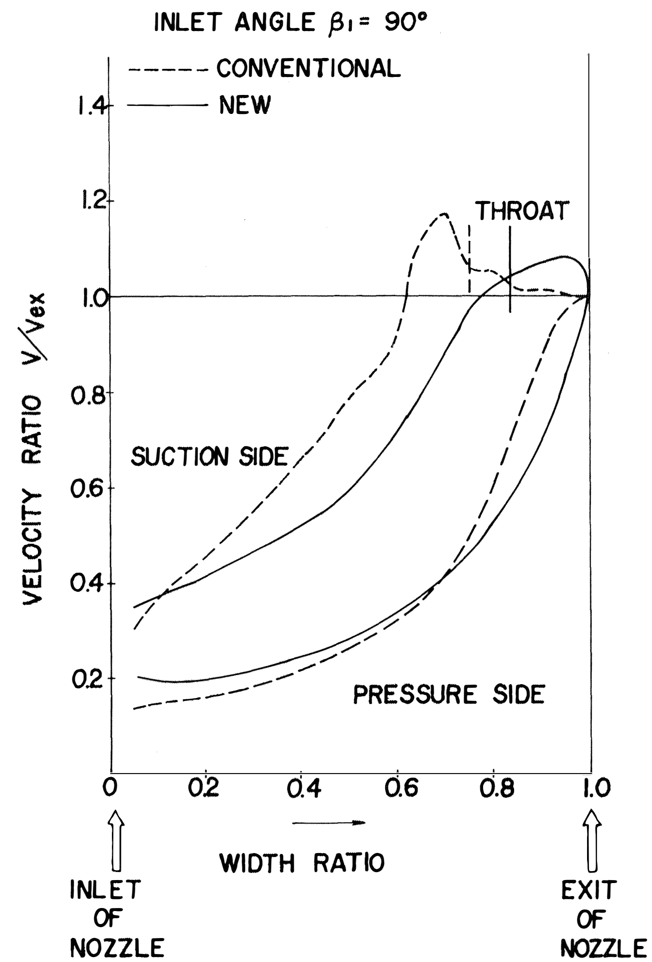

Figure 11. Comparison of Rateau Nozzle Velocity Distribution.

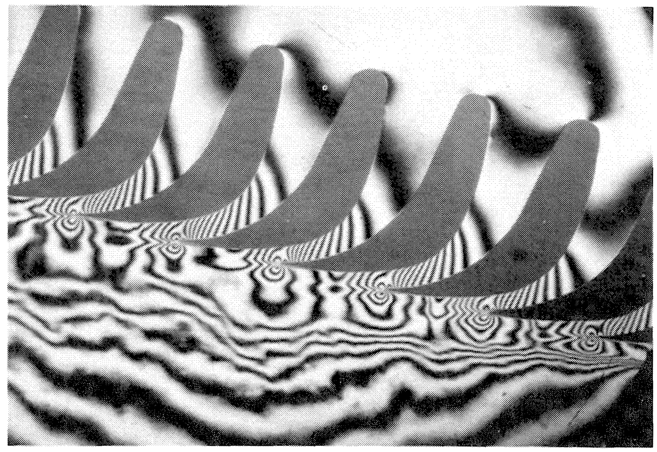

Figure 12. Interferogram of Rateau Nozzle.

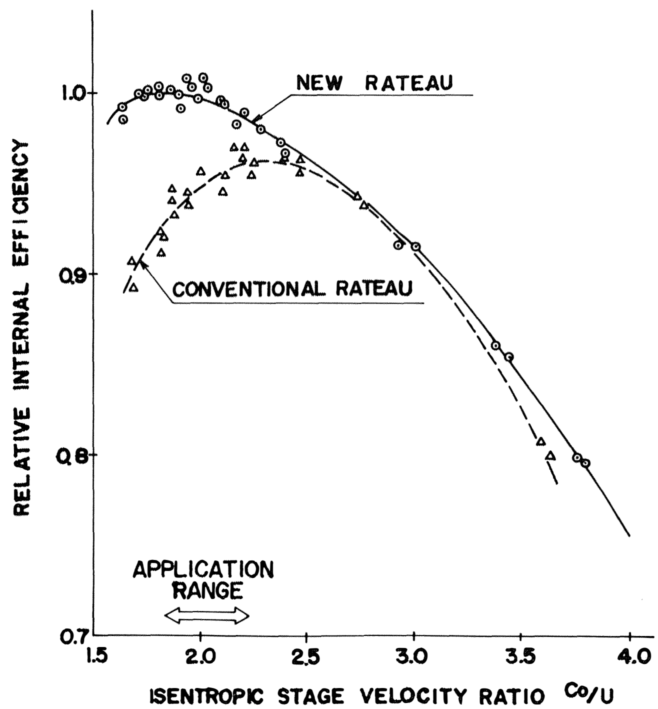

Figure 13. Comparison of Internal Efficiencies Between New and Conventional Rateau Stages.

The typical optical test result obtained in the high speed wind tunnel is shown in Figure 12. Smooth flow acceleration can be observed in this figure.

To verify the effectiveness of the new nozzle and blade profiles, overall performance tests were conducted using a high pressure steam test turbine.

As a result, it was found that in the range of ordinary velocity ratio, the new Rateau stage was one level higher in efficiency than the conventional Rateau stage as shown in Figure 13. Using this test turbine, the performance of Rateau stage is experimentally examined; that is, the effects of velocity ratio, pressure ratio, blade tip clearance and blade height upon its efficiency is made clear.

Reduction of leakage loss

Leakage loss on blade tip side is affected by reaction degree, shroud fin shape, leakage area and step-up (difference of blade and nozzle heights). In order to minimize leakage loss on the Rateau stage blade tip side, the relationship of leakage flow coefficient versus step-up and nozzle oblique angle is studied. Figure 14 shows the relation between the step-up and leakage flow coefficient which has been obtained in wind tunnel tests. This figure has made it possible to select the optimum step-up. It has been confirmed, by a turbine performance test, that leakage loss is lower and internal efficiency is higher in the Rateau stage with the optimized step-up and oblique nozzle than in the Rateau stage with insufficient step-up. The test results obtained with test steam turbine are shown in Figure 15.

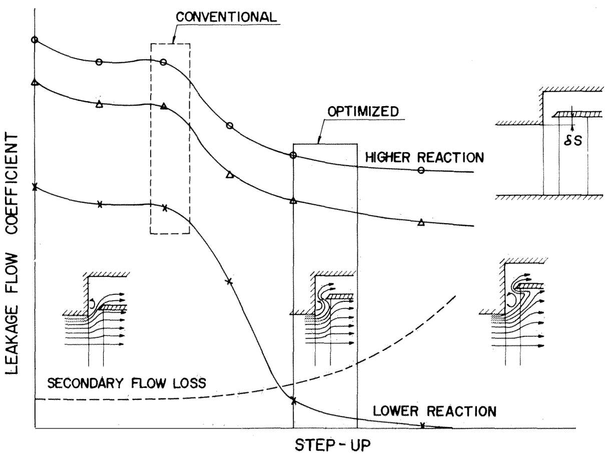

Figure 14. Effect of Step-up on Blade Tip Leakage Flow (Flow Coefficient vs. Step-up).

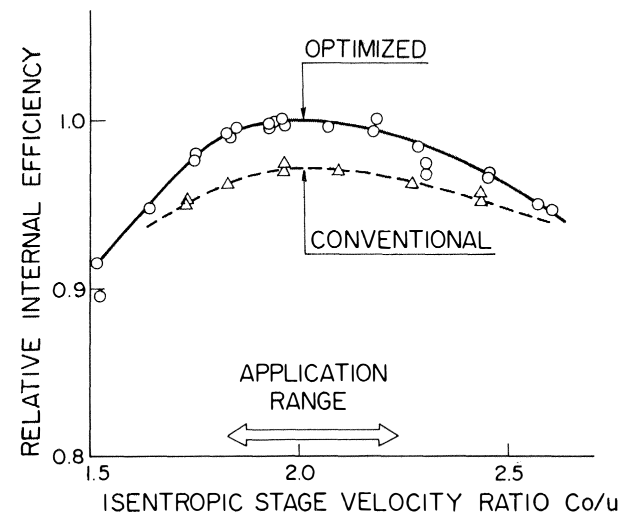

Figure 15. Comparison of Rateau Stage Performance (Effect of Step-up).

Performance improvement of low pressure stages

Low pressure stages have features that sharply expand with the passage in the meridional plane. Pressure ratio per stage is great, blade length is long, and the blades operate in wet steam.

Effective means for improving the low pressure stage performance are: reduction in profile loss due to improvement of profile, reduction in incidence loss and secondary flow loss due to improvement of flow pattern, and reduction in moisture loss due to optimized design of the drain catcher.

Improvement of nozzle profile

Since long blades in low pressure stages operate in a wide range from transonic to supersonic speed according to variation of turbine load and vacuum, it is necessary, in designing nozzle profile, to take into account shock loss and loss due to interference between shock wave and boundary layer on the nozzle surface as well as the losses to be considered for subsonic flow. Consequently, we adopt the optimum nozzle profile by synthetically evaluating much data obtained from cascade numerical flow analyses made by the modified FLIC (Fluid in cell) method [1] and from cascade tests conducted in the high speed wind tunnel. Figure 16 shows an example of test results comparing shape and profile loss between the straight back profile applicable for long blades in low pressure stages and the conventional curved back profile.

Improvement of flow pattern

As the blade length becomes long and hub/tip diameter ratio becomes small in the low pressure stages, the velocity triangle varies greatly from base to tip. Recently it has become possible to improve the performance of low pressure stages by controlling the reaction distribution in the direction of the blade length by means of three-dimensional flow pattern design method. In other words, by improving blade profile and controlling the stream line in the meridional plane, it has become possible to increase reaction degree on the base side and lower it on the tip side as shown in Figure 17.

With the reaction degree controlled, as shown in this figure, the inlet flow angle of steam on the blade base side becomes large, and the turning angle decreases; as a result, profile losses and secondary flow losses decrease. Furthermore, with a decrease of the reaction degree on the tip side, leakage loss on the tip side decreases. Figure 18 shows the

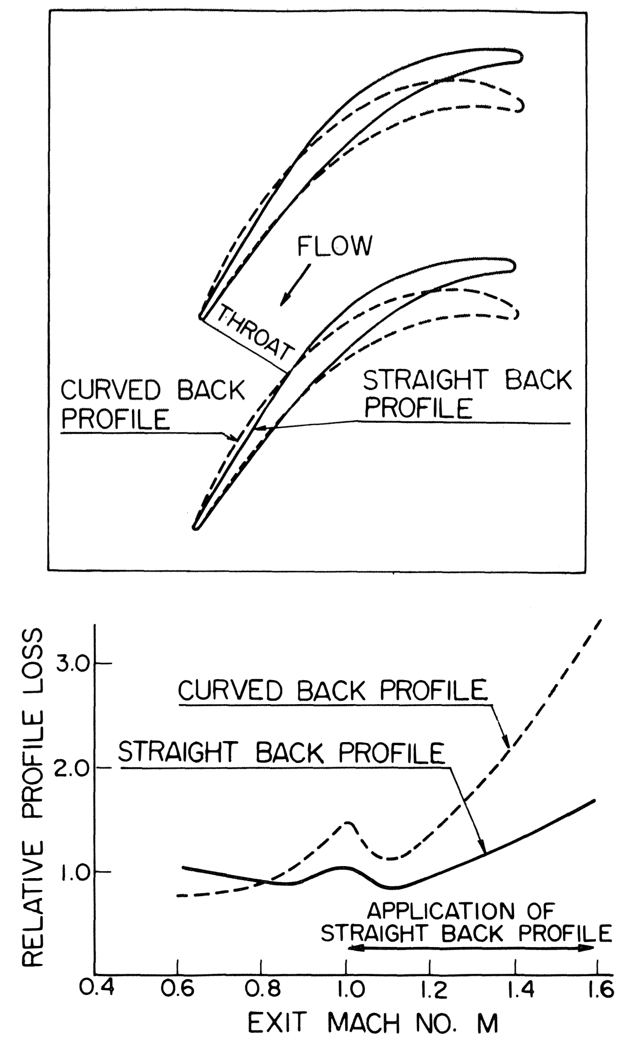

Figure 16. Comparison of Straight-Back and Curved-Back Nozzle.

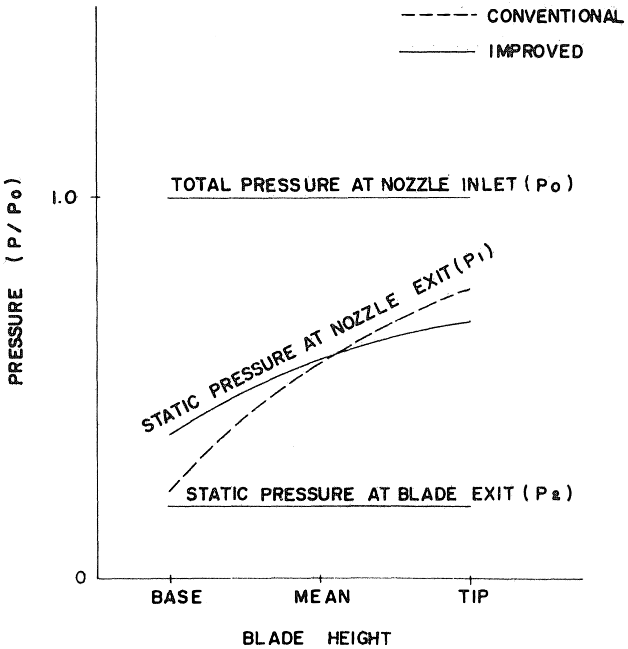

Figure 17. Pressure Distribution of Low Pressure Stage.

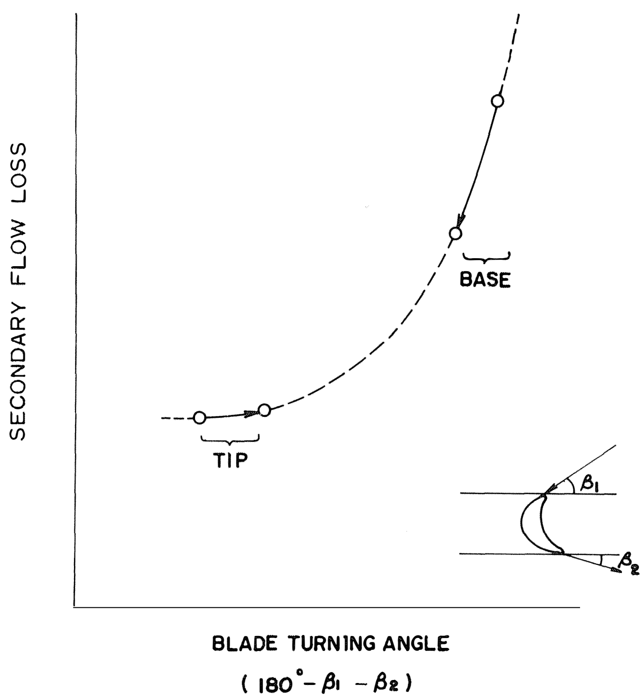

Figure 18. Secondary Flow Loss vs. Turning Angle (Effect of Optimized Reaction).

relation between the turning angle of steam flow and secondary flow loss obtained by cascade test and turbine traverse test. The effectiveness of the flow pattern improvement has been verified by overall performance tests and traverse tests using the test turbine. The test turbine facility with pitot tube traversing system is shown in Figure 19.

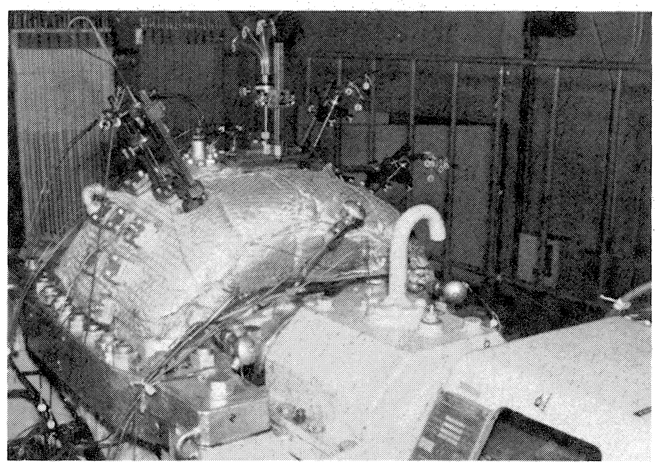

Figure 19. Turbine Test Facility with Pitot Tube Traversing Equipments.

Reduction of moisture loss

As mentioned, the stage moisture loss accounts for a large part in low pressure stages. In order to reduce moisture loss, it is necessary to know the behavior of droplets in low pressure stages and remove harmful droplets effectively.

Analyses and experiments have been conducted to determine the behavior of droplets in low pressure stages, and the optimized design methods for the axial distance, between nozzle and blade and for the drain catcher, have been established. Figure 20 shows the behavior of droplets in the low pressure stages.

The greater part of the fine droplets which have entered the nozzle passes with steam into the next stage, but a part of the fine droplets deviate from the steam stream line and are collected on the leading edge and pressure side of the nozzle to form a water film. The water film is broken by the steam flow at the trailing edge of the nozzle, and is discharged into the nozzle wake in the form of coarse droplets. The coarse droplets again form a water film on the blade surface, then move to the blade tip by centrifugal force, and are discharged. The orbits of droplets discharged in radial direction vary according to droplet diameters as shown in Figure 21. Therefore, it is seen that the minimum droplet diameter that can be removed by the drain catcher and the droplet removal ratio vary with the

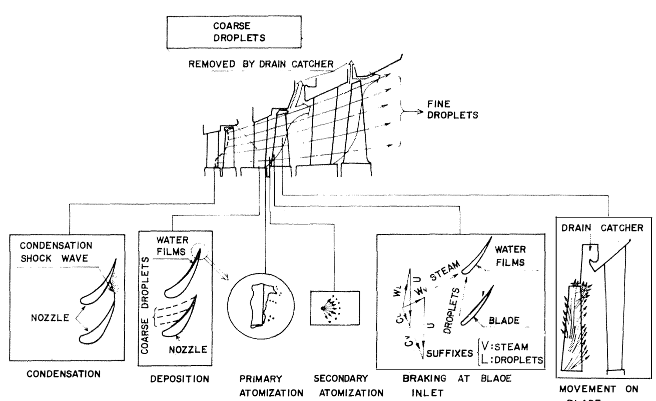

Figure 20. Behavior of Moisture Droplets in Low Pressure Stage.

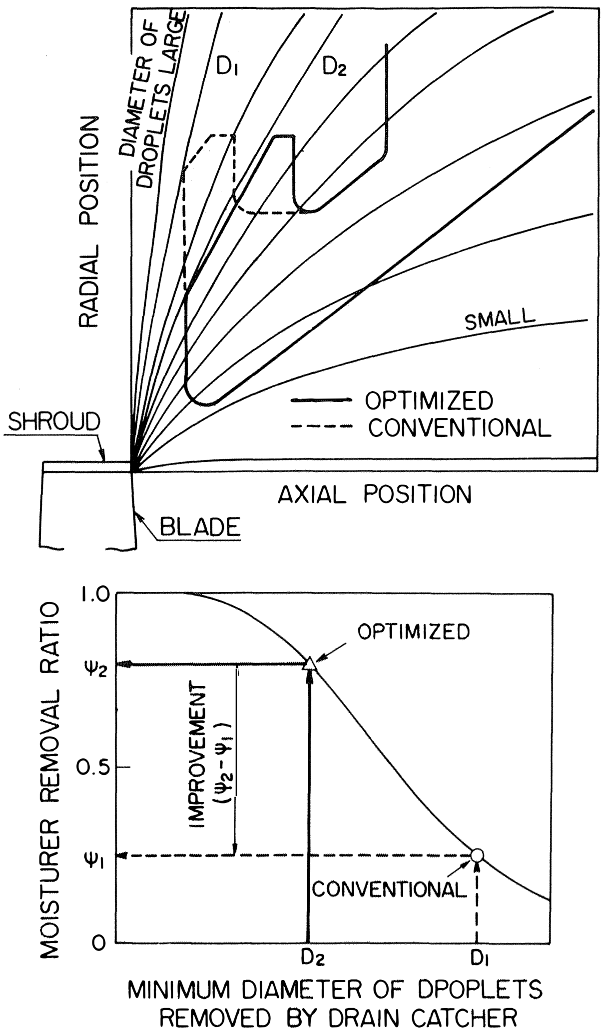

Figure 21. Orbit or Droplets in Low Pressure Stage (Optimized Design of Drain Catcher).

location and shape of the drain catcher. Thus, it has become possible to reduce the moisture loss by optimizing the drain catcher by calculating the diameter, distribution and orbit of droplets experimentally and theoretically. Figure 21 shows an example of the optimized design of the drain catcher.

Evaluation of performance improvement

The overall turbine efficiency gains obtained by applying the above-mentioned technologies to typical large scale mechanical drive condensing steam turbines are evaluated as shown in Table 1. In this example, it is seen that performance can be improved by about 3%. This value of efficiency gain is epoch-making, because large scale condensing turbines are originally designed to obtain the highest efficiency. Smaller steam turbines for which lower price is required rather than higher performance, have more efficiency gains as verified by modification of existing turbines.

MODIFICATION OF EXISTING TURBINES FOR ENERGY CONSERVATION

Some existing turbines have been modified for energy conservation by applying new technology. They are classified into three categories.

1. To replace the nozzle and blade with new ones to meet decreased operating load.
2. To apply new longer nozzle and blade at the exhaust end in the existing turbine casing to decrease leaving loss.
3. To replace the old machine with a new one of higher efficiency.

TABLE 1. EVALUATION OF THE OVERALL TURBINE EFFICIENCY GAINS (TYPICAL LARGE SCALE CONDENSING TURBINES).

		Efficiency Gains
1. Decrease of Exhaust Loss		0.5%
2. HP Stages Improvement		1.7%
Profile Improvement	1.2%	
Leakage Loss Reduction	0.5%	
3. LP Stages Improvement		0.9%
Profile Improvement	0.4%	
Flow Pattern Improvement	0.4%	
Moisture Loss Reduction	0.1%	
4. 1. ~ 3. Sum Total		3.1%

Figure 22 shows an example (above Case 1) of modification of synthesis gas compressor driver for methanol plant. In this case, steam consumption was decreased by 5% with the replacement of a low pressure stage blading. Figure 23 shows an example (above Case 2) of modification of a charge gas compressor driver for ethylene plant.

In this case, steam consumption was decreased 5% by replacement with longer blades. The modified turbine longitudinal section is shown in Figure 24.

Replacement of the existing turbines with new machines is planned as a part of energy conservation measure throughout the plant. Replacement is, in some cases, accompanied with the change in turbine type and steam condition.

PERFORMANCE TEST

It is important to verify actual turbine efficiency by performance test. Figure 25 shows an axial blower driving turbine under a shop performance test. The result of the performance

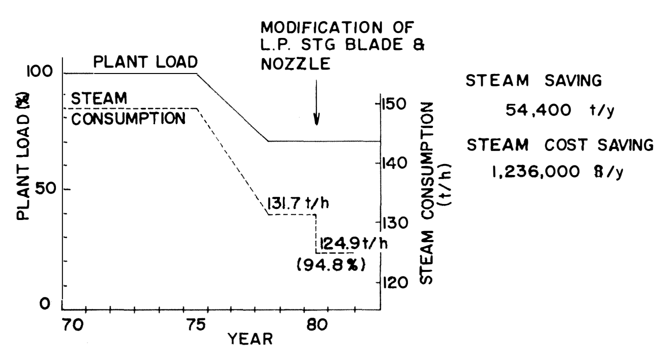

Figure 22. Save Energy Modification (Methanol Plant Syn. Gas Compressor Drive Turbine).

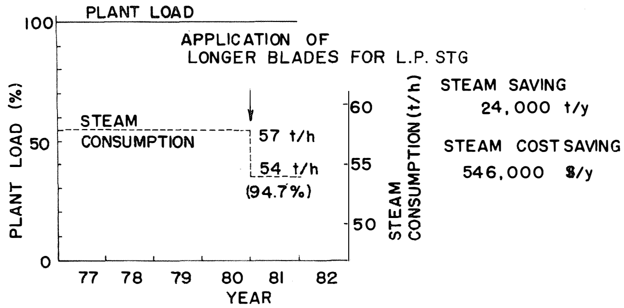

Figure 23. Save Energy Modification (Ethylene Plant Charge Gas Compressor Drive Turbine).

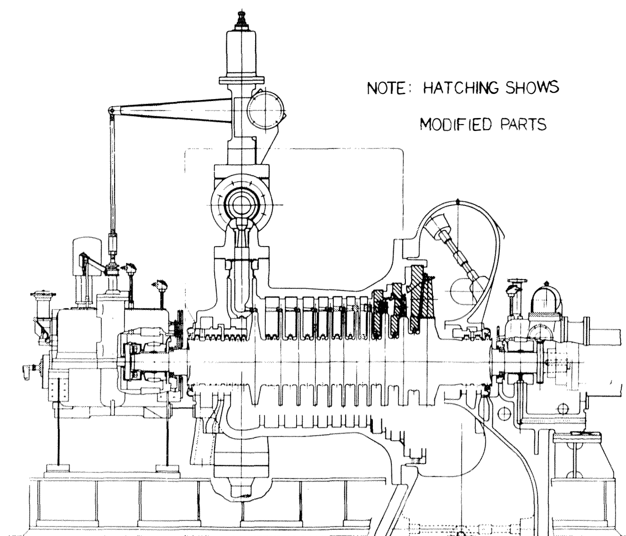

Figure 24. Modified Turbine Longitudinal Section.

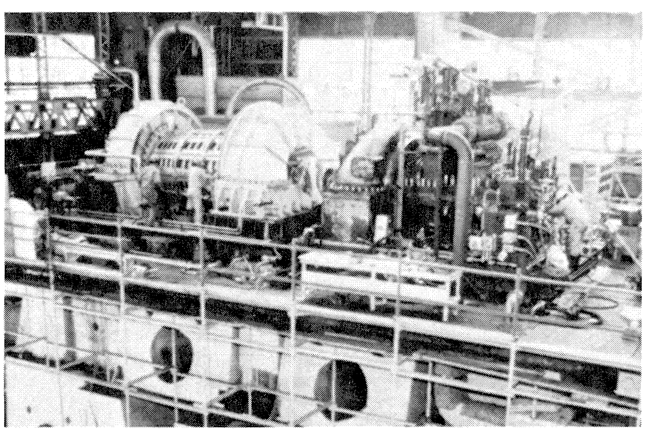

Figure 25. Photograph of Shop Performance Test Condition.

test showed good coincidence with the expected value. In this case, output, speed, primary flow, each pressure and temperature were measured in accordance with the method specified by ASME PTC-6, and compared with expected steam consumption rate as shown in Figure 26. Total instrument error

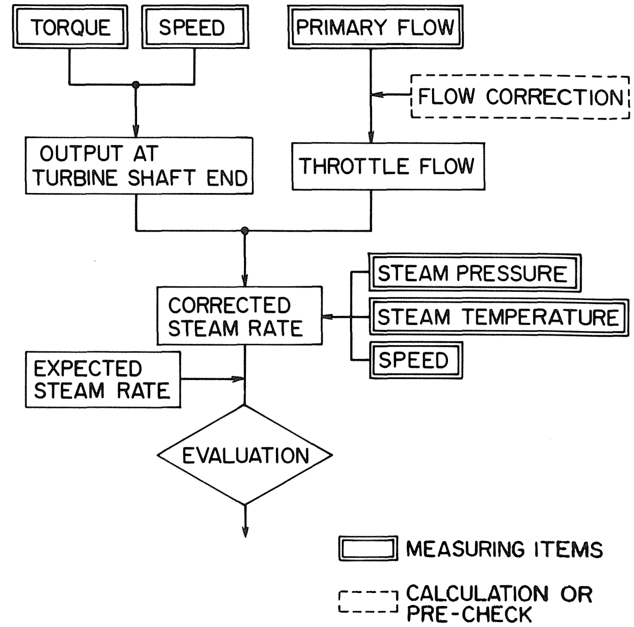

Figure 26. Summary of Shop Performance Test.

should be small enough to make a performance test significant. Therefore, the instrumentation program was completed after precise evaluation of total error which affects steam consumption rate (total error of 2.1% was achieved in the case mentioned above). Efforts to minimize total error are concentrated on increasing the torque and flow measurement accuracy.

When shop performance test is not practical because of the limitation of shop test facility, performance test is, in some cases, carried out in the field in cooperation with the user.

CONCLUSION

Some important approaches to improve the turbine efficiency were introduced. They are summarized as follows.

1. Decrease in exhaust losses by enlarging the exhaust annulus area and optimizing exhaust hood.
2. Decrease in losses in each stage by improving the blade profile, and the flow pattern and optimizing the drain catcher.

These technologies have been adopted to existing and new machines after confirming the effectiveness and reliability by cascade test and overall performance test using test turbines. It is hoped that users will show good understanding of new turbines having higher efficiency.

Also, it is important to confirm the actual turbine efficiency, so it is recommended that the shop performance tests or field tests be carried out. From now on, efforts will be continued, to make efforts to further improve turbine efficiency so as to contribute to energy conservation.

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