HIGH SPEED AND LARGE CAPACITY COMPRESSOR-DRIVING TURBINES FOR CHEMICAL PLANTS

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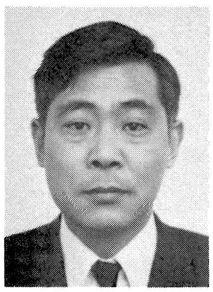
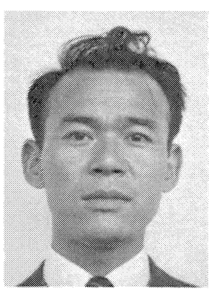


Makoto Teramoto became manager of the Mitsubishi Heavy Industries' Hiroshima turbine designing section in 1973. After graduating from Kobe Mercantile Marine University in 1956 with a Bachelor of Science in Mechanical Engineering, he worked for six years as a marine engineer for ocean liners. He then moved to Mitsubishi Heavy Industries' turbine section where he has been involved in the design of steam turbines for the past 16 years.



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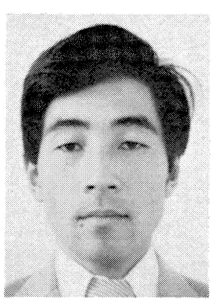

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ABSTRACT

This paper gives a brief description of compressor-driving steam turbines for use in chemical plants.

Turbines used for ethylene plants, ammonia plants and LNG plants, in particular, are large-capacity, high-speed variable-speed steam turbines. Their present state and the state of their development are described. Rotor dynamics technique which is the basic design technique for the attainment of their stable operation and blade design and manufacturing techniques are described in detail. Further, the construction of each component part is described in full.

INTRODUCTION

The petrochemical industry has made a rapid expansion in recent years. Particularly, plants producing ethylene as a basic material for various kinds of petrochemical products, ammonia as a material for nitrogenous fertilizers and liquefied natural gas plants supplying low pollution energy have been expanded for lower manufacturing costs.

The expansion of their producing scales caused a remarkable increase in gas quantity in their producing processes and large capacity turbo-compressors have been adopted for their compressing processes. These large compressors are the hearts of these plants and it is important that they should have a long-term stability and high performance. As their driving machines, steam turbines have been adopted for the effective use of the heat energy of various decomposition reactions, depending on processes.

Compressor-driving turbines, which have higher speeds, efficiencies and larger capacities, have been required in accordance with the rapid growth of plant capacity of petrochemical industries in our country. Such requirements have been satisfied by our research and development work. As the result of such effort, those having an output of 32000 kW and a speed of 12000 rpm as represented by synthesis gas compressor-driving turbines for ammonia plant have been developed and are in operation already. Up to the present, a total of 246 compressor-driving turbines with a total output of 3264 MW have been delivered. Most of these turbines are for ethylene plants, ammonia plants and LNG plants.

This paper presents a brief description of the turbines used in these plants, various performances and structures required with these turbines.

GENERAL CHARACTERISTICS REQUIRED WITH COMPRESSOR-DRIVING TURBINES

Centrifugal compressors are widely used in chemical plants as they have flat characteristics against fluctuations in suction gas quantity and can be stably operated in a wide range. Turbines used for driving centrifugal compressors, therefore, should have such performances as will give full play to the characteristics of the compressors. From this viewpoint, we explain various characteristics specially required with centrifugal compressor-driving turbines.

These turbines are variable-speed turbines and are generally used from 80 to 105 percent of their design speeds, not clear in the initial stage of plant design and the characteristics of catalysts change during plant operation. In the design and production of turbines, therefore, it is necessary to bear in mind that they can be operated under maximum load at any point of the above mentioned operation range.

Since start-up of the turbines is carried out supplying gas to the compressors, they are sped up under a considerable load. Patricularly, extraction turbines used in ethylene plants and ammonia plants are in some cases required to take out the maximum output on the low pressure side at the time of start-up due to steam balance. Figure 1 shows the concept of the load characteristics of these turbines with respect to speed comparing with those of marine turbines and generator turbines.

Recent chemical plants constitute a gigantic industrial complex. Unexpected shut-down of one of them leads to the shut-down of all the related plants, giving a great damage to the enterprises concerned. The American Petroleum Institute Specifications (API) are wholly applied to the design and manufacture of the turbines used in those plants.

Also it is a general tendency that when a large output is required, both-end drive turbines are demanded.

TURBINES FOR VARIOUS KINDS OF PLANTS

Present ethylene producing processes are roughly divided into two processes: (1) thermal cracking of hydrocarbons and (2) separation of ethylene from various kinds of hydrocarbons contained in the cracked gas arising from the thermal cracking and purification of the ethylene separated. Therefore, three kinds of compressors are used in these plants; namely, cracked gas compressors, propylene refrigerant compressors and ethylene refrigerant compressors. Steam used for these compressors is generated by use of the heat of waste gas arising from thermal cracking furnaces. Figure 2 shows a schematic diagram of the steam system of a standard ethylene plant constructed recently.

The cracked gas compressor is used for compressing raw gas and requires the largest power in the ethylene plant.

The propylene refrigerant compressor is used for the refrigerating cycle of the ethylene purifying process. A long time low speed operation for start-up causes a rise in gas temperature and gives a bad effect on the compressor. Therefore, the compressor should be sped up from low speed to the rated speed within several minutes. For its main steam, steam extracted from the turbine for the cracked gas compressor or high pressure high temperature steam is used.

For the ethylene refrigerant compressor, a condensing turbine with a comparatively small output is used and its main steam is obtained from the steam extraction line of each large-type turbine.

Figure 3 shows a cross sectional drawing of a typical turbine in the ethylene plant.

Ammonia is produced by compressing and refrigerating the synthesis gas which is a mixture of nitrogen in air and hydrogen obtained by the reforming of hydrocarbon by steam. Therefore, its production generally requires three kinds of compressing processes: compression of air, compression of synthesis gas and refrigeration of ammonia. Turbines used for driving these compressors are slightly different in speed and output by plants. Figure 4 shows a schematic diagram of the steam system of a typical ammonia plant with a capacity of 1360 tons/day.

Barrel type compressors are adopted for synthesis gas compression as its discharge pressure is as high as 200-340 kg/cm^{2}g. They are operated at high speeds, and require large motive powers. In order to make effective use of a large

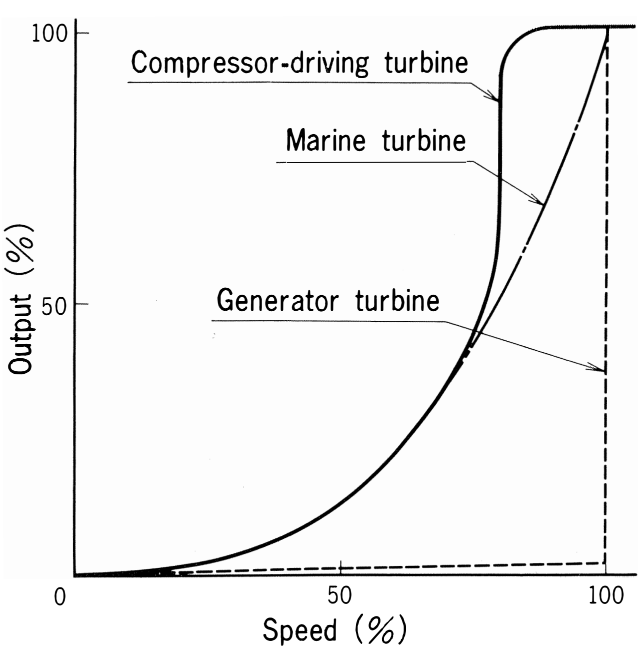


Figure 1. Output Characteristics of Various Kinds of Turbines.

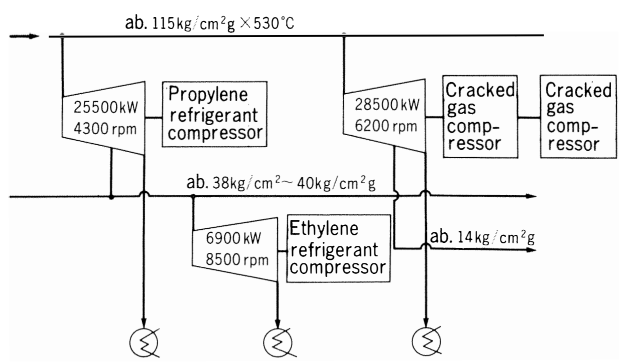


Figure 2. Example of Turbines and Compressors Used in Ethylene Plant.

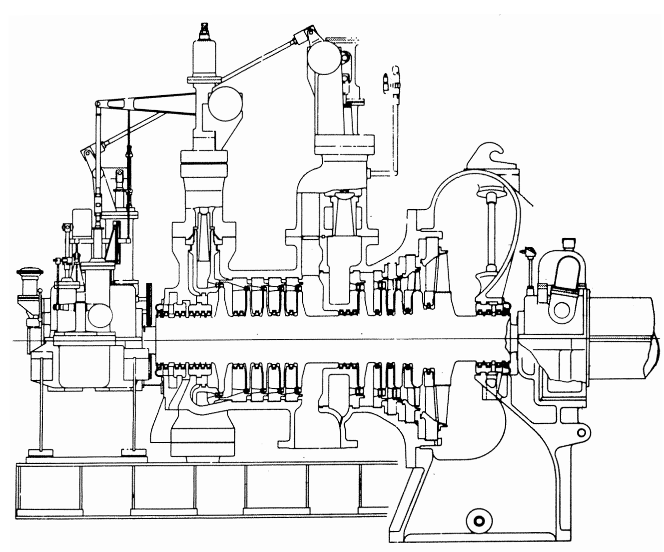


Figure 3. Cracked Gas Compressor-Driving Turbine in Ethylene Plant.

amount of steam generated from the waste heat boiler after reforming, a very large amount of the steam is required to flow into the turbine. On the other hand, as the process requires much steam, a considerable amount of steam must be extracted. A greater portion of motive power for the compressor can be supplied from the high pressure part only.

Figures 5 and 6 show cross sectional drawing of typical turbines, of which Figure 5 shows single casing single flow type and Figure 6 shows double casing double flow type. Both are of both-end drive type. A large number of the former are already in use in plants of 1360 tons/day capacity class. The latter is already installed in a plant with a capacity of 1500 tons/day but can be used in plants with capacities of up to 3000 tons/day by combining blades which have been in actual service.

In the background of the debut of LNG as a low-pollution fuel and also as a star item for the diversification of energy resource, there are its ease of handling and the application of refrigerant compressors required for the liquefaction of natural gas which were developed as the result of an increase of ethylene plants in size. This is same with turbines used for driving compressors, experiences in many ethylene plants are applied in the design and manufacture of these turbines.

Ordinarily, a plant is composed of several units and one unit is equipped with three large compressors. As these compressors require almost the same motive power and speed, many condensing steam turbines of the same or similar specifications are manufactured. The main steam line is common to all turbines and is generally of 60 $kg/cm^2g \times 440°C$.

Figure 7 shows a cross sectional drawing of a LNG plant in operation now. This plant is composed of five units with a total annual production capacity of five million tons. Our company delivered a total of 15 turbines of the same type to this plant.

As many large machines are installed in one plant and many persons and days are required for their maintenance, maintenance-free machines are in demand. In plants of this kind, continuous operation for two to five years without overhauling is being planned.

STRUCTURE OF TURBINE

Large factors which determine the overall structure of a turbine are rotor bearing system design technique and blade design technique. Here we describe our view on these techniques and present studies first and then explain the structure of each part.

Rotor Bearing System Design Technique

The permissible value for shaft vibration during turbine operation is specified by the API specifications as shown with solid line in Figure 8. Black points in the figure denote the

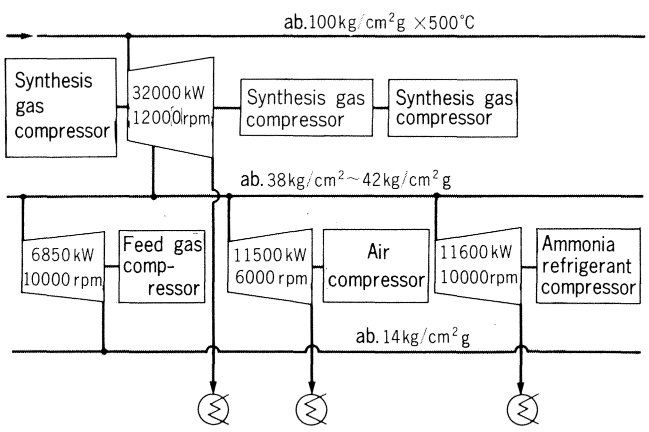


Figure 4. Example of Turbines and Compressors in Ammonia Plant.

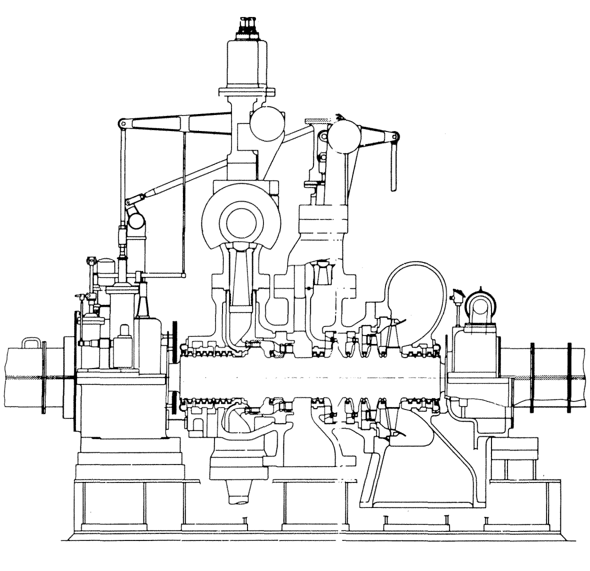


Figure 5. Synthesis Gas Compressor-Driving Turbine in Ammonia Plant. (Single-Casing Type)

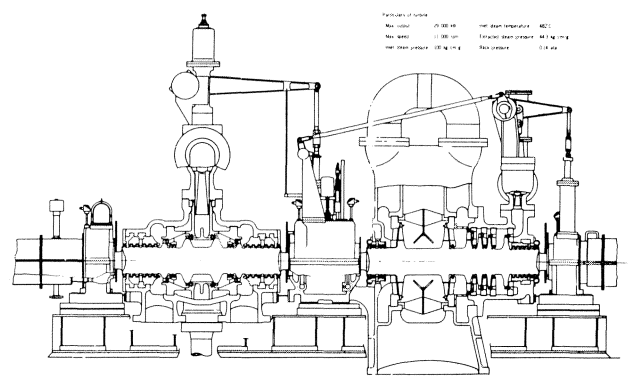


Figure 6. Synthesis Gas Compressor-Driving Turbine in Ammonia Plant. (Two-Casing Type).

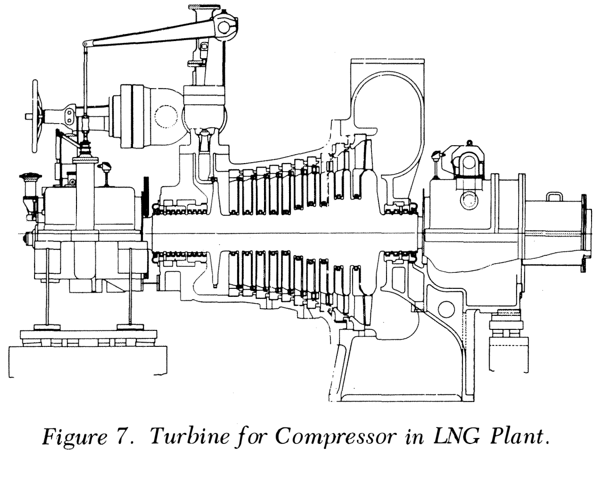


Figure 7. Turbine for Compressor in LNG Plant.

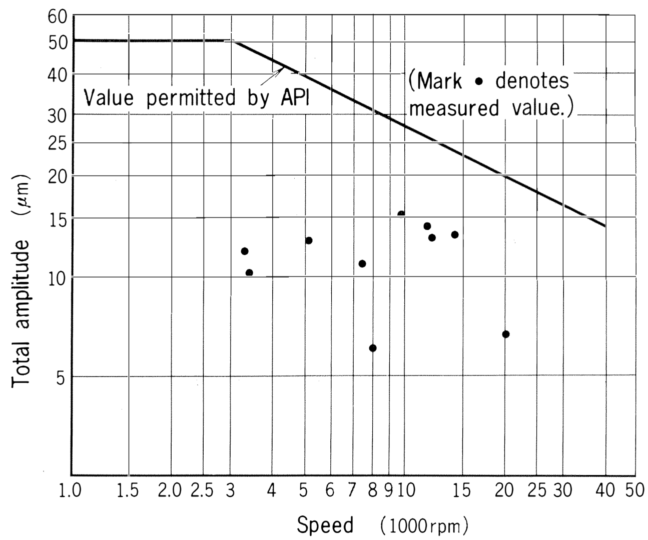


Figure 8. Comparison Between Permissible Shaft Vibration and Actually Measured Value of Shaft Vibration.

values of shaft vibration measured on turbines delivered from our works recently.

The problems of shaft vibration can be roughly divided into two kinds, synchronous vibration caused by unbalance remaining on the rotor and non-synchronous vibration due to the instability of the rotor bearing system.

Generally, the natural frequency of a rotor supported on both ends is expressed as the function of supporting stiffness as shown in Figure 9. On the other hand, during operation, the rotor is put in a floating condition on a thin oil film formed on the bearing and the bearing itself is elastically supported on the pedestal. If these supporting stiffnesses are plotted on the figure as the functions of speed, the critical speed of the rotor bearing system can be obtained as the points of intersection of both. In the figure, two supporting stiffness curves are shown. This is because the dynamic characteristics of oil films in vertical and horizontal directions are generally different, when a cylindrical bearing is used. It is usual that this critical speed is excluded from the turbine operating speed range and this is also specified by the API. According to our estimation by the method mentioned above, there exist many critical speeds, which makes the overall design of turbines difficult and greatly narrows their operating ranges.

According to the results of a certain recent research on high speed turbines, it is important in the design of rotor bearing systems to consider the effect of vibration damping occurring on the oil film of the bearing [1]. Figure 10 shows the results of analysis by Dr. Balda [2] of a simplified rotor bearing system. It can be found that when an oil film bearing is used vibration magnification at the time of resonance can be greatly reduced by making ω_a/ω_∞ small, or making the value of K/C small. That is, a rotor bearing system insensitive to unbalance can be manufactured by making the bending stiffness of the rotor higher than that of the oil film of the bearing.

Our company conducted studies on this problem under the leadership of our Technical Headquarters and could obtain the characteristics of the complicated turbine rotor bearing system accurately. In the instance shown in Figure 9, resonance magnification for each critical speed is shown with the size of circle. In this instance, the first and second critical speed can reduce resonance magnification very low in the range where the motions of the rotor are approximately seemed to be those of a rigid body. It is found from out measurement on actual turbines that vibrations at a critical speed where resonance magnification is low are very small and the continuous operation is entirely safe.

Ordinarily, in the design of rotor bearing systems, the third critical speed is put sufficiently high from the operating range. It is necessary, however, to note that when the natural frequency of the overhang part of the rotor is low, the resonance magnification of local vibration is large and the third critical speed causes unexpected vibration trouble.

The rotor balancing operation should be determined according to the results of study on the resonance magnification of the rotor bearing system. The turbine rotors manufactured by our company are balanced step by step after installing moving blade of each stage so that distributed unbalance inducing a mode which causes a comparatively large bending of the rotors does not occur.

Non-synchronous vibration shows various phenomena according to the mechanism by which it is induced. The matters to which special attention is paid in the design of turbines in our works are shown in Table 1 for reference.

As the material for rotors, Ni-Cr-Mo-V forged steel having high toughness at low temperatures and high creep strength at high temperatures is widely used. Further, a high strength material with 12% Cr was recently developed for use as a material for high speed turbines (about 20000 rpm) and has already been applied in some turbines. The journal bearing is a crash type cylindrical bearing and the bearing liner is a standardized one having steady quality obtained by mass production. A pressure dam is provided on the upper half of the bearing surface. It works as a stabilizer in high speed operation. Tilting pad journal bearing is applied for higher peripheral speed use. As for thrust bearings, a Kingsbury type bearing with load equalizer is used on the main thrust side and a Mitchell type or Tapered-land type bearing is used on the anti-thrust side. Each compressor and turbine are also equipped with thrust bearings respectively and they are combined with a gear coupling. Therefore, when the rotor expands or contracts during load transmission, force required for sliding the tooth surface is applied to the thrust bearings additionally as reaction force. This force is considerably large in large-output turbines. Therefore, thrust bearings having a sufficient load carrying capacity should be used.

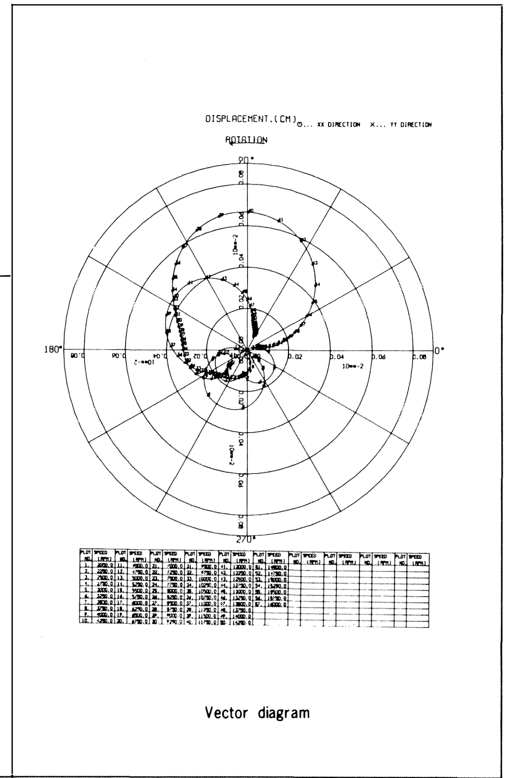
Stage and Blade Design

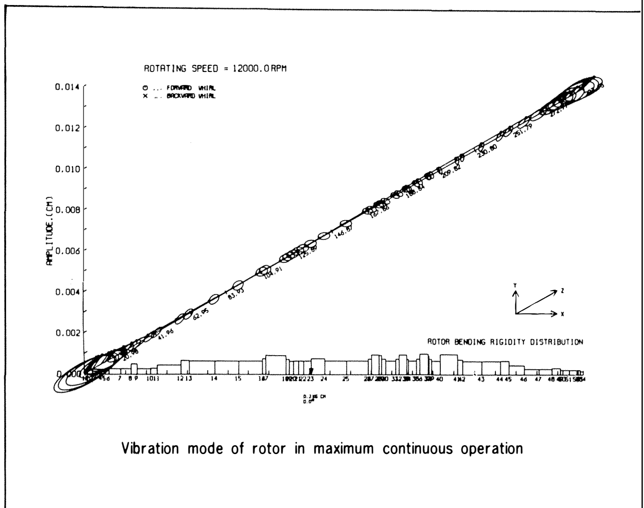
Steam turbines can be roughly divided into impulse turbine and reaction turbine. Both types have their respective features. The compressor-driving turbines manufactured by our company adopt the impulse type because of their following merits.

1. Impulse stage need not be equipped with a complicated fins for preventing steam leakage in the vicinity of the moving blades. This makes quick start-up and stopping possible and makes overhauling and assembly easier. Besides, there is no unstable vibration of the rotor bearing system caused by improper clearance of the fin.
2. There is no pressure difference between the steam inlet side and outlet side of the moving blades, and consequently, there is no excitation force which causes axial vibration of the moving blades. This makes comparatively easy and highly dependable anti-vibration design of moving blades possible.

3. Rotor bearing system is stable as the total number of stages is small and axial length is short.
4. Thrust bearings are simple in structure and have high reliability since axial thrust applied to the rotor is small.

The moving blades are subject to stresses caused by the large centrifugal force produced by rotation, thrust force of





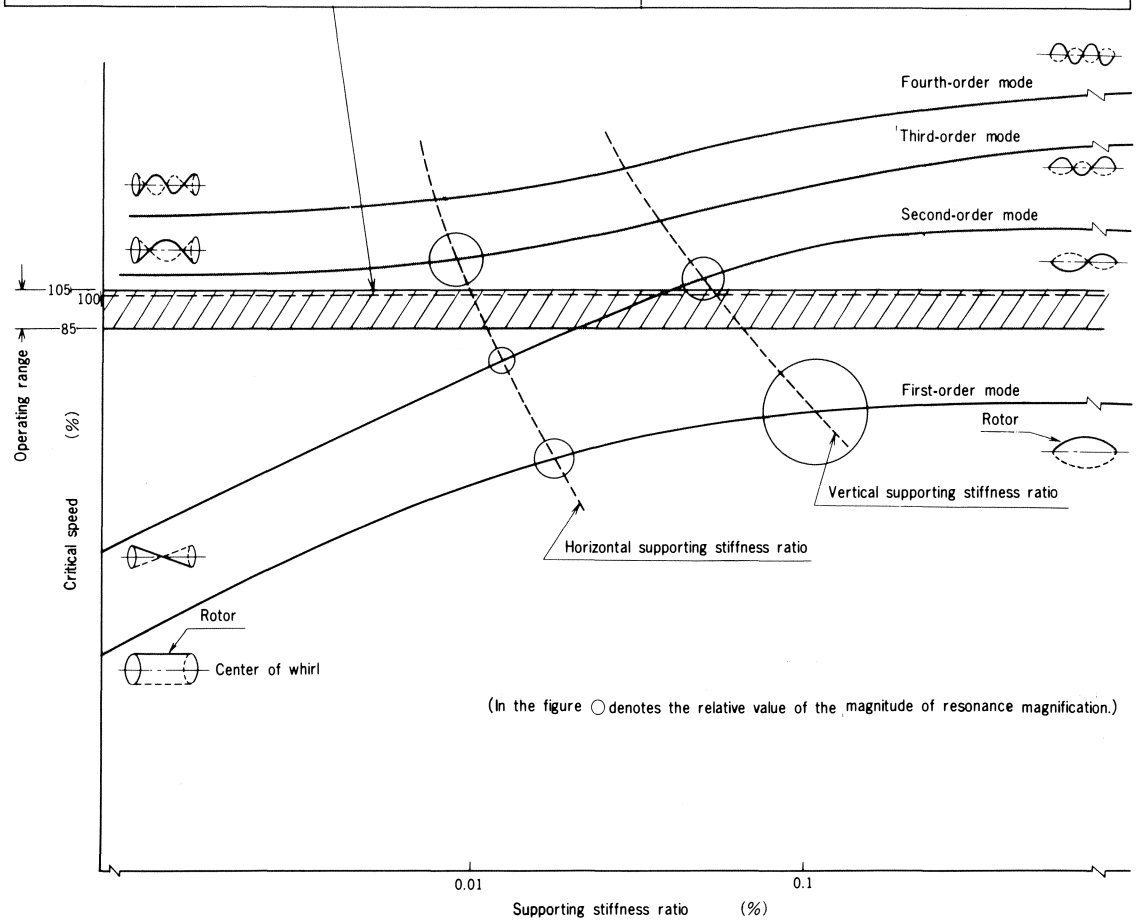
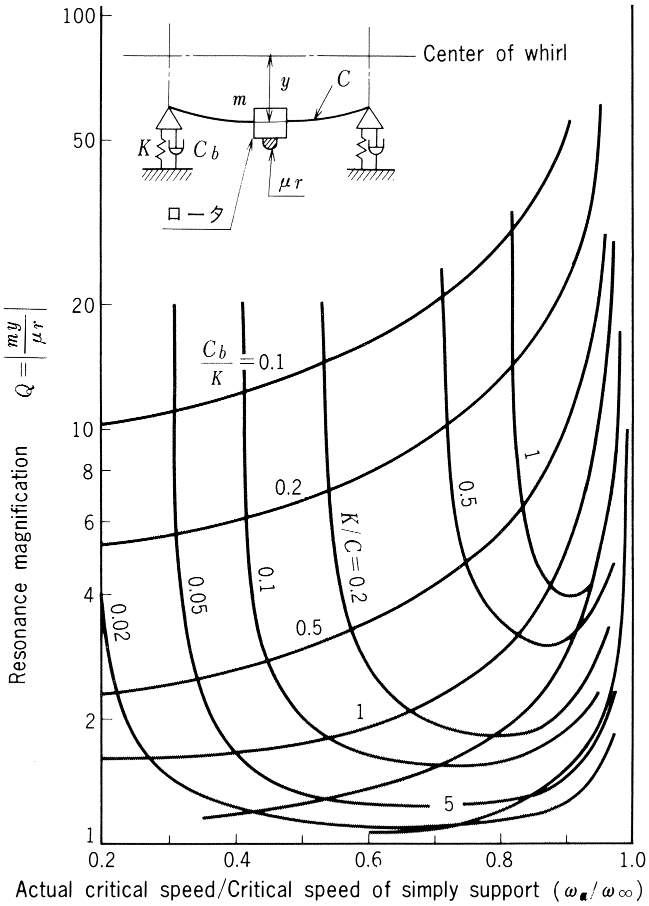


Figure 9. Critical Speed Diagram.



m: rotor mass
y: rotor displacement
K: spring constant of supporting system
Cb: damping factor of supporting system
μr: unbalance moment
C: bending stiffness of rotor
$\omega\infty$: critical speed of simple support
ωa: natural frequency of actual system

Figure 10. Magnification of Simplified Rotor Bearing System.

steam and vibration of the blades themselves. Stresses caused by centrifugal and thrust force of steam are static stresses and can be known accurately by calculation. On the other hand, dynamic stress due to vibration can be predicted only by knowing the vibration exciting forces during operation and the resonance response characteristics of the moving blades accurately. Most of the moving blade failures in the past presented a beach marks on their fracture surfaces, which resulted from the vibration fatigue rupture of their materials.

The natural vibration modes of moving blades differ variously according to their shapes and dimensions [3]. Here, we describe the points requiring special attention in the design of compressor-driving turbines.

Exciting forces are mainly caused by steam disturbance force. Table 2 shows various factors causing exciting forces. It can be found from the table that matters for consideration differ by stages.

Exciting forces in the control stage and extraction pressure control stage are mainly due to partial admission for adjusting

TABLE 1. KINDS OF NON-SYNCHRONOUS VIBRATIONS AND COUNTERMEASURES

Cause	Countermeasures
Vibration due to instability bearing oil film	(1) Adoption of antiwhip cylindrical bearing or tilting pad bearing (2) Increasing of the stiffness of rotor and to maintain critical speed higher than half the maximum operating speed (3) Increasing of the weight of rotor to reduce the effect of the unbalance force of steam
Vibration caused by rotor internal friction	Rotor is of forged and integral type
Vibration resulting from dry friction	(1) Casing and diaphragm are supported on the same plane as the center of rotor. Therefore, the center of labyrinth packing does not shift by thermal deformation and does not contact with the casing nor diaphragm. (2) Even if a rotor contacts with labyrinth packing, it is stable as the packing can move back.
Unstable vibration due to non-uniform steam pressure (steam whirl)	There is no fin around the moving blades.

TABLE 2. CAUSES OF EXCITING FORCES

Control stage	1. Exciting force generated by impact force caused by partial admission 2. Exciting force caused by wake at the trailing edges of nozzles
Full admission stage (100% admission stage)	1. Exciting force caused by wake at the trailing edges of nozzles 2. Exciting force generated due to existence of inlets or outlets on several places on the entire circumference 3. Structural members in the fore and after of blades (for example, exciting force generated in diaphragm support, rib & stay at exhaust port) 4. Exciting force caused by irregular nozzle passage horizontal split discrepancy between inner and outer walls of diaphragm or leakage of steam 5. Exciting force caused out of circular shape diaphragm 6. Exciting force caused by non-uniform nozzle shapes 7. Exciting force caused by mechanical external force transmitted from rotor shaft

the amount of inflowing steam. In this case, the moving blades are subjected to fluctuations of force in the moment they go into or out from the active arc zone during one rotation as shown in Figure 11. Forces created at this moment can be obtained from the data of measurements using a wind tunnel or experimental turbine. Stresses generated in the moving blades can be calculated as a transient response calculation.

Output in the control stage is generally large. Particularly there are turbines for driving ammonia synthesis gas compressors in which single stage output is as large as 10000 kW. In this case, wake created in the stream behind the nozzle is so strong that resonance occurring as the moving blades pass through it should be avoided. In order to reduce this exciting force, a very fine pitch nozzle as shown in Figure 12 is used.

In the 100 percent admission stage where nozzles are arranged on the entire circumference, uniform steam flow ought to be made over the entire circumference theoretically. Actually, however, non-uniform steam flow as described in Table 2 is present. Of the medium length moving blades in the intermediate stage it is necessary to pay attention for that stresses generated at the time of resonance of the tangential end-supported type vibration [3] with the exciting force generated when the moving blades pass through the nozzle wake. In this type of vibration mode the middle part of the blade bends while the ends are supported.

In the last stage of condensing turbines, the natural frequency of the moving blades is low as compared with the turbine speed and it is necessary to pay attention to its resonance with the exciting force caused by the turbulence of steam in the exhaust chamber. This exciting force can be considered by dividing it into integral multiple components of the turbine speed. Its resonance with the moving blades can be understood from the Campbell diagram as shown in Figure 13. According to this example, the resonant point is present at a point corresponding to 94 percent of the design speed.

As compressor driving turbines have a wide operating speed range as mentioned above, it is impossible to avoid the resonance over the entire operating range by adjusting the frequency of the moving blades. Therefore, it is important to accurately estimate the vibration stresses at the time of resonance and thereby provide a sufficient safety factor to the fatigue strength of the material.

In answer to this problem, we have kept up studies on the resonance response characteristics of the moving blades, the

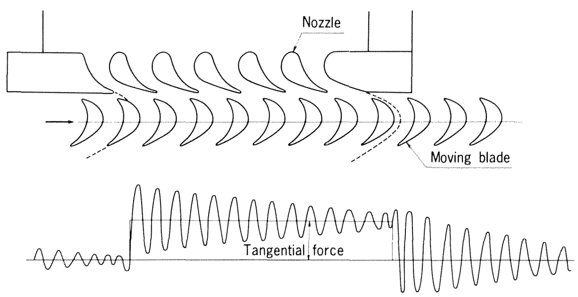


Figure 11. Vibration of Moving Blades in Governing Stage Due to Partial Admission.

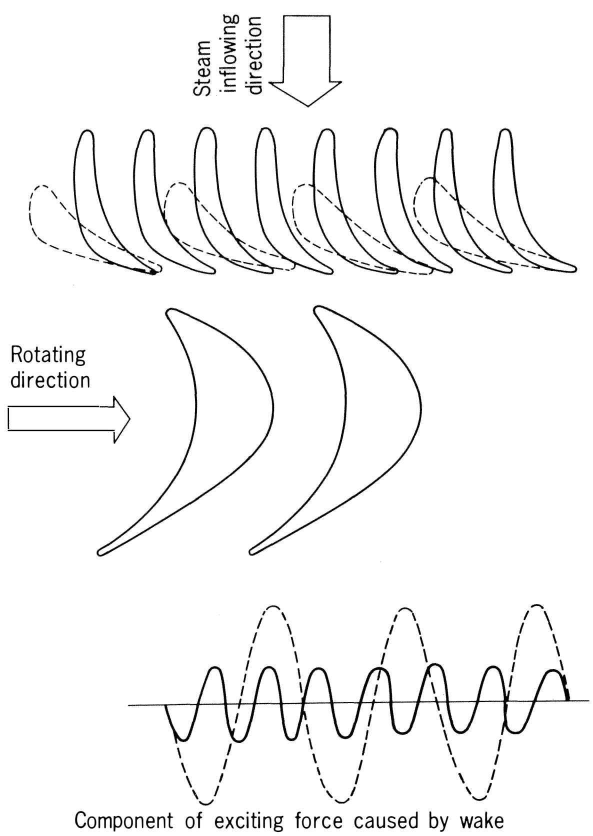


Figure 12. Fine Pitch Nozzle.

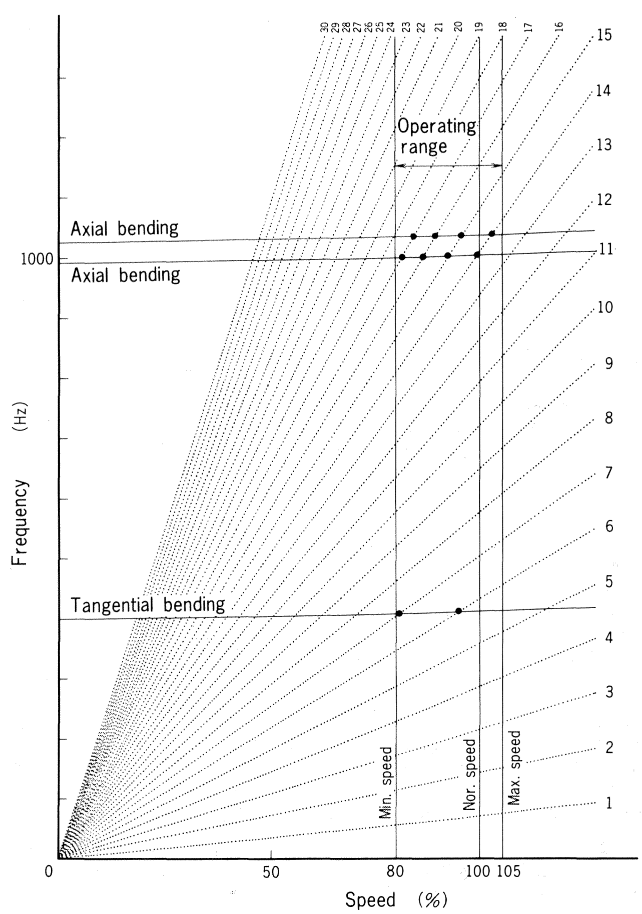


Figure 13. Campbell Diagram.

magnitude of exciting force and the fatigue strength of the material.

The moving blades manufactured recently, particularly those for high speed turbines, tend to become wider and there are many instances in which actual phenomena cannot be explained by the conventional method. In these instances, analysis by the finite element method (FEM) and observation by laser holography are useful. Figure 14 shows an example of the results of such analysis and observation. The result of analysis of FEM shows a good agreement with the phenomenon of local vibration of moving blade group.

Figure 15 shows a testing device used for the study of exciting forces applied to low pressure stage moving blades.

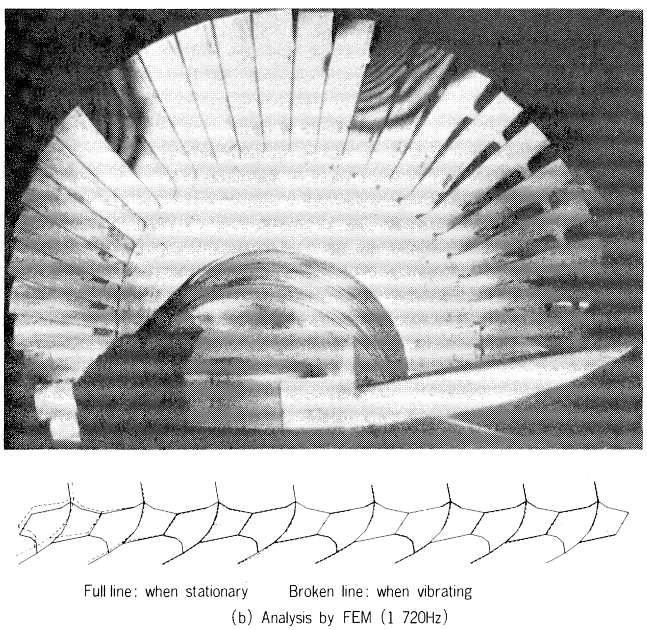


Figure 14. Comparison Between Results of Measurement and Analysis of Partial Vibration of Lashed Blade.

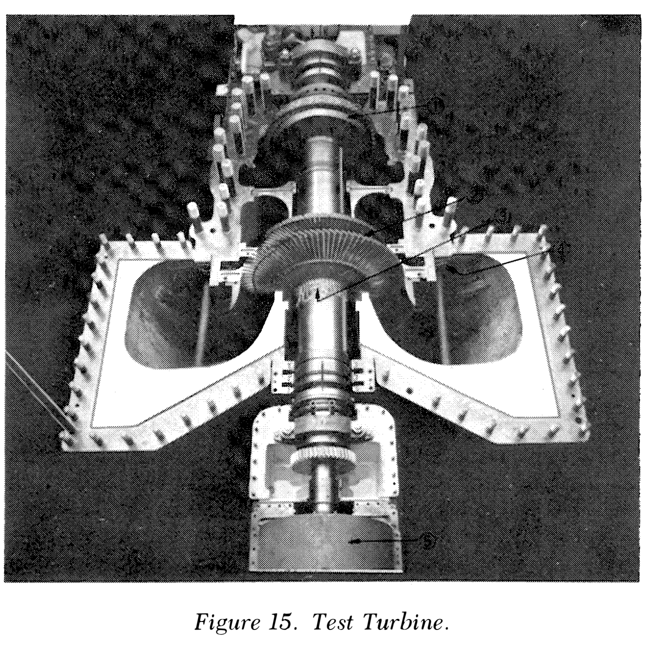


Figure 15. Test Turbine.

This test turbine is composed of a low pressure turbine of the same shape as actual turbines and a high pressure turbine which works as a loading device. It can produce entirely the same state as that at the time of operation of an actual turbine on the low pressure turbine side. By using this device, we could obtain the effects of the various shapes of the last stage diaphragm moving blades and exhaust casing on the vibration of the moving blades. Figure 16 shows an example of the data which were obtained from actual load test using that turbine.

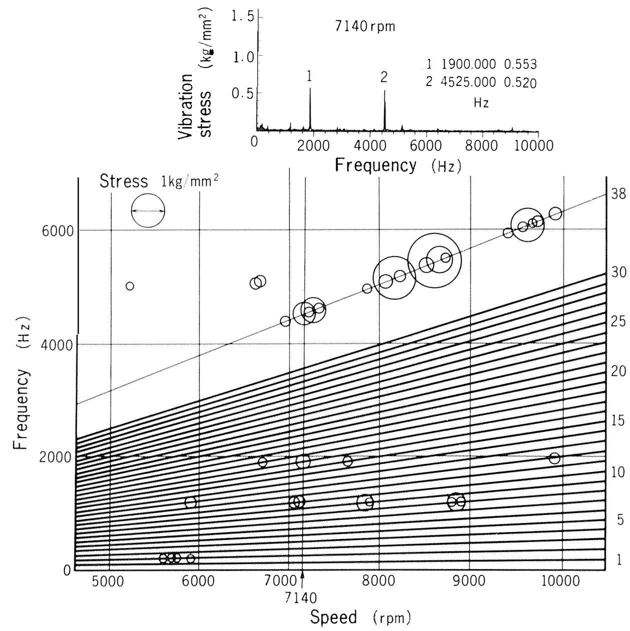


Figure 16. (1) Result of Measurement of Vibration Response of Last Stage Blades.

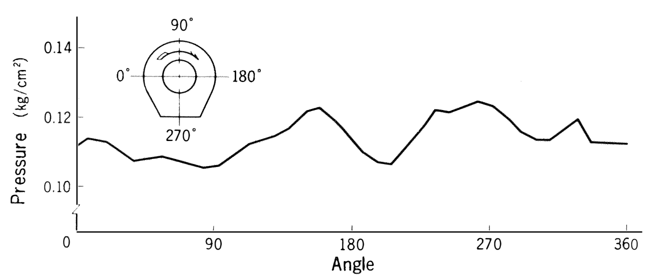


Figure 16. (2) Static Pressure Distribution Behind Last Stage.

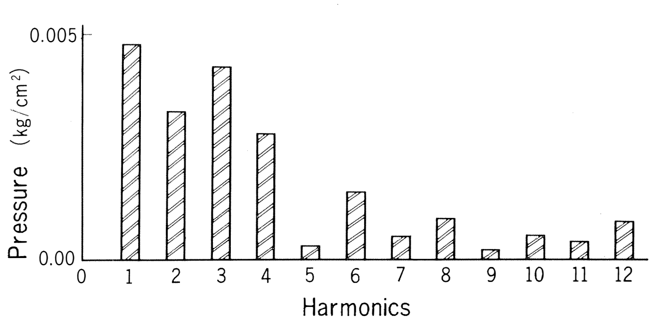


Figure 16. (3) Harmonic Analysis of Pressure Distribution.

Further, long-term endurance tests of newly developed moving blades is conducted by use of this testing device.

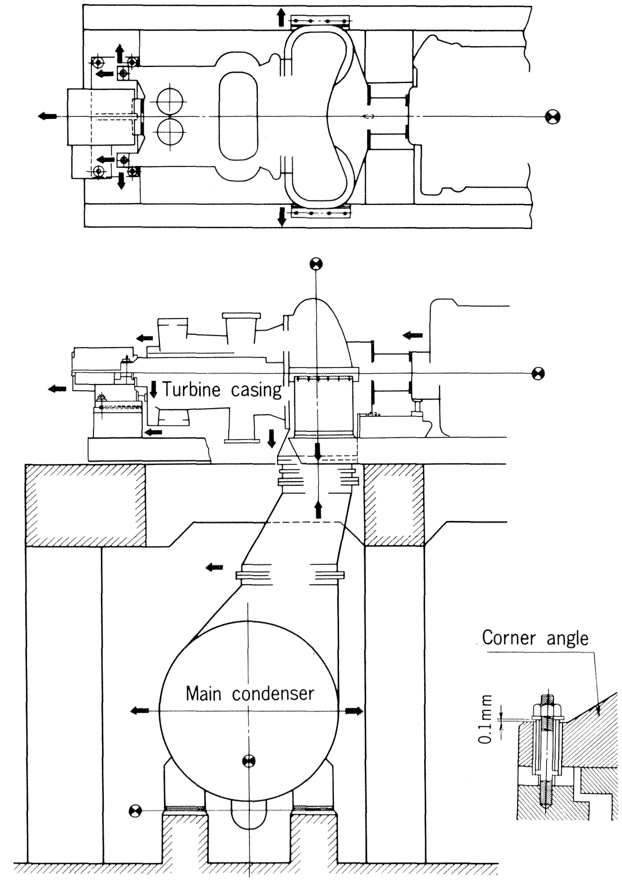
Figure 17 shows a part of the results of study on the fatigue strength of blade materials conducted recently [4]. In the design of compressor-driving turbines, it is necessary to pay attention to the fatigue strength in the various kinds of environment because corrosive substances are liable to be contained in steam and the stellite plate for preventing drain erosion in their low pressure stages might be silver soldered on the moving blade surfaces. As blade material, 13% Cr steel having corrosion resistance, high elevated temperature strength and high damping factor is mainly used.

For the moving blades of control stage, double tee roots are used while for high speed moving blades, side entry type fir tree roots are adopted for the fixture to the rotor discs. Both types have on their upper part a shroud integral with the moving blades and their top is lashed with a shroud band. This structure is for increasing vibration damping capacity. For the last stage moving blades where each blade length is large and centrifugal force is strong, side entry type fir tree roots are used and most of blade airfoil parts is manufactured by the precision forging method.

Structure of Each Part

All parts of the turbine casing are of horizontally divided type and the casing is supported on pedestals at the horizontal plane which coincide with the shaft center. Figure 18 shows the relation between casing supporting method and thermal expansion at the time of operation. In two-casing type turbines used in ammonia plants, as a special example, a bearing pedestal integral with the low pressure casing is fixed between two casings the the other casing and the rotor are arranged on both sides of the bearing stand as a fixed point so that they can make thermal expansion as shown in Figure 6, too.

The nozzle boxes are directly connected with independent regulating valve chests respectively and forms a steam chest as shown in Figure 19. Their ends are fixed to the casing and are freely expandable. This type is used for high temperature high pressure turbines and is capable of quick start-up and quick change of load. For turbines for medium temperature and



Mark ⊖ denotes that the center line with this mark is immovable
Mark ➡ denotes expanding direction

Figure 18. Thermal Expansion of Turbine Casing.

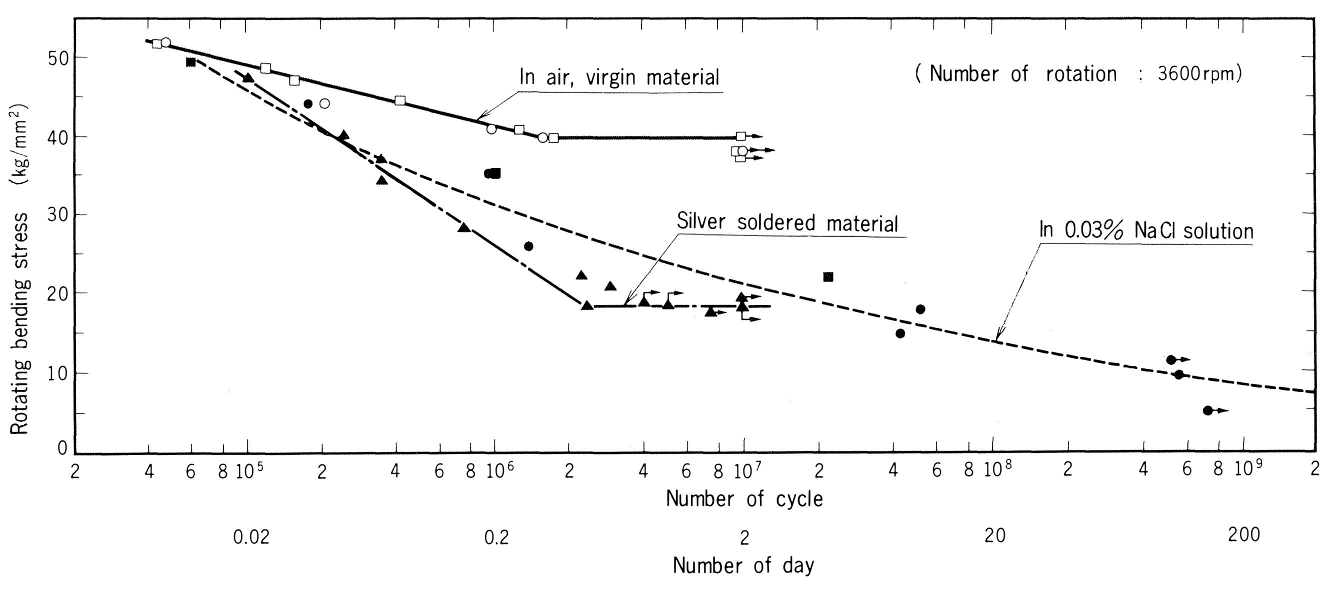


Figure 17. S-N Curve of SUS 410 J 1 Material in Various Surface Conditions.

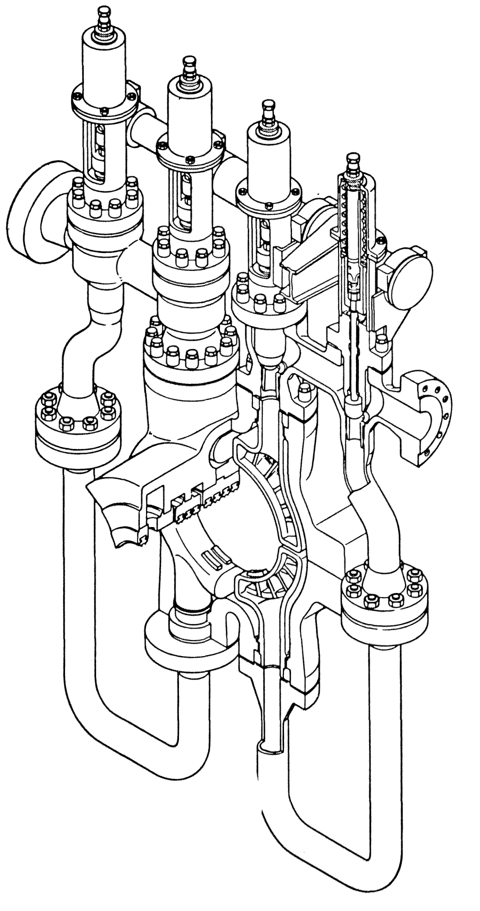


Figure 19. Nozzle Box and Governing Valve.

pressure steam, a type having a steam chest of solid casting with the regulating valve chest is adopted.

The regulating valve is single beat type and is equipped with a pressure recovering diffuser. The opening and closing of the valve is performed by means of a lifting bar and besides, a spring is equipped on the upper part of the valve chest so that in case of emergency the main trip and throttle valve and regulating valves close completely at the same time for double closing. This mechanism increases safety against emergency.

In the start-up of the turbine, the regulating valves are in entirely open state till the turbine speed comes to the lowest governing speed and the regulation of turbine speed is performed by the main trip and throttle valve only. As a considerable load is already applied by this time, the speed regulation is performed by opening the main trip and throttle valve slightly. The main trip and throttle valve can be fixed at any opening and in order to prevent steam turbulence occurring at the time, a special rectifier is equipped.

The diaphragm is divided into two parts (upper and lower). Fabricated nozzle type is adopted for the high pressure part and welded nozzle type is adopted for the low pressure part. The diaphragm expands while being held its center surely even if there arises a difference in thermal expansion between casing and itself, so that steam jet from nozzles comes into moving blades without any deviation and turbluence, and there is no contact of the diaphragm labyrinth packing with the rotor. Furthermore the continuity of jet at the horizontal split portion would not be disturbed.

Speed Governors and Safety Devices

The turbine speed is governed by a Woodward governor. Signals from the governor is amplified via the oil pilot valve to open or close the regulating valves. Pressure (main steam pressure, extraction steam pressure or mixed steam pressure) is detected by an Askania and controlled by converting it into power piston displacement. Figure 20 shows a diagram of the

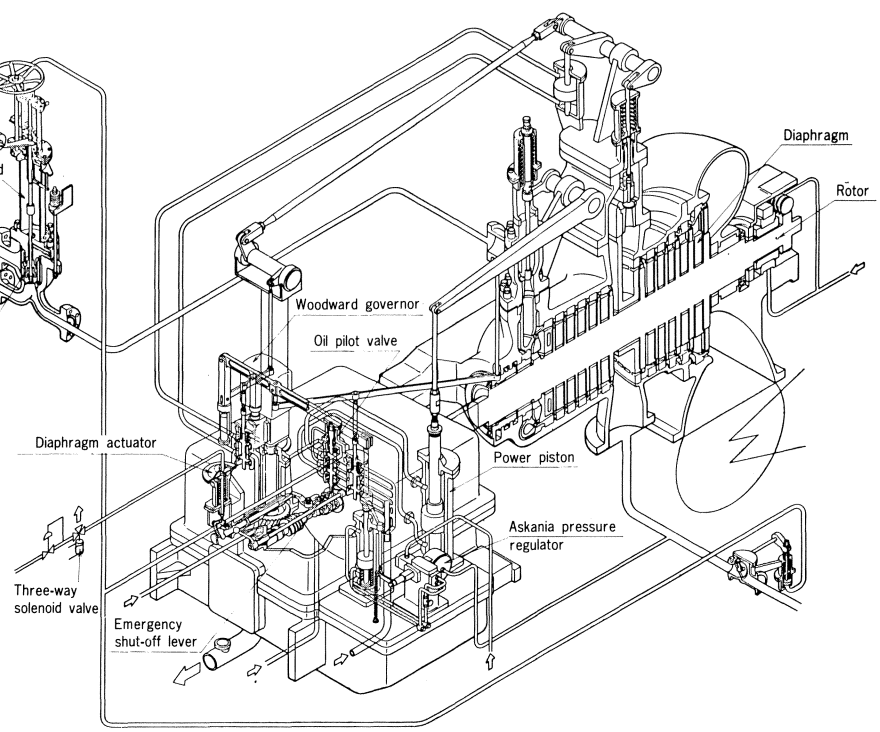


Figure 20. Schematic Diagram of Governing System and Safety Devices.

control system. The link mechanism with a long lever does not need a large angle of rotation, hence it undergoes little wear, works smoothly and has a good controllability.

The overspeed of the rotor is liable to cause serious accidents. In order to prevent accidents the rotor is equipped with an overspeed governor, which actuates the emergency circuit breaker to close the main trip and throttle valve and regulating valves instantly. For other abnormalities as shown in Table 3, a three-way solenoid valve is installed in the air pipe, which actuates the diaphragm actuator for immediate shut-down.

TABLE 3. EMERGENCY TRIP FUNCTIONS

- Overspeed of rotor
- Decline of lubricating oil pressure
- Wear of thrust
- Bearing temperature rise
- Abnormal rotor vibration
- Stoppage of air supply for instrumentation
- Oil level drop of sealing oil head tank
- Temperature rise of gas discharged from compressor
- Pressure rise of extracted steam

Supervisory Meters

In order to discover any abnormality during operation in an early stage and take quick countermeasures, a steam pressure meter and a thermometer are installed on the turbine and besides a shaft vibration meter and a bearing metal thermometer are fitted on each bearing.

CONCLUSION

Chemical plants are still showing an expanding tendency. Ethylene plants with an annual production capacity of 700000 tons, ammonia plants with daily production capacity of 2000-3000 and LNG plants with 2 million tons, per unit will make debut in the near future. Research and development on compressor-driving turbines is being carried on in our company, and such work will be sure to be useful to construct large turbines in the future.

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