DEVELOPMENTS IN STEAM TURBINES FOR THE SUGAR INDUSTRY

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Abstract

Demands for higher outputs have been met by careful tailoring of steam turbines for both mechanical and electrical power generation to meet the needs of thermal efficiency and reliability at an economic cost. Major system components and the methods chosen for speed control are described. Important operational requirements are featured.

1. Introduction

Steam turbines have been used in sugar industries for very many years. The sugar manufacturing process requires steam for its heat energy and for electric power for driving motors in various applications.

Where there is an approximate steam and power balance, back-pressure turbines are used. The steam raised in the boiler at a higher pressure and temperature than is required for the process is first passed through the turbine where power for either mechanical or electric drive is generated. The steam is then exhausted to process still containing the bulk of its energy in the form of its latent heat, which is then used in various ways resulting in the conversion of the steam into hot feed water for return to the boiler.

The thermal efficiency of the cycle for power delivered at the turbine gear output coupling will be about 80%, depending on the output and the operating conditions. The back pressure turbine is a very valuable asset to the sugar cane industry and to the conservation of fuel generally.[1]

W. H. Allen Sons & Co. Ltd. and Belliss & Morcom have supplied steam turbines of all types for the cane sugar industry in South Africa since 1930 and have just achieved 50 years of supplying steam turbines to the British Sugar Corporation which is a beet sugar industry where back-pressure turbo-alternator sets are used.[2]

However there are now pressures in various areas to conserve bagasse for other purposes e.g. paper making etc. Also as mills have grown, so have the electrical requirements. In many cases these have been met by additional small output sets. There has therefore arisen the need for sets of 3 MW to 8 MW (e.g. 3,3 MW shredder drive at AK) for either additional power and/or to reduce to number of individual units.

Against these demands is the need for economic units in terms of capital cost. In the late 1960's two ranges of multi-stage turbines were developed to meet the various requirements. A few are already in service in this country and many in all parts of the world, and others are in course of installation and manufacture.

2. Multi-stage turbine ranges

(*a*) General

These turbines are of the impulse pressure-compounded type. The pressure drop occurs almost wholly in the fixed nozzles, only a small part being allowed to take place in the blades of low reaction type. This type of turbine can be made very robust in the moving parts including the blades and does not need the ultra fine clearances of the high reaction type where half the pressure of each stage occurs across the moving blades. The rotor in these impulse turbines is of the "stiff" type i.e. its first critical speed is well above the normal running speed. It is therefore not so sensitive to starting conditions. This type of turbine has been shown to be capable of great reliability and long life given reasonable operating conditions.

The solutions adopted for these turbines were:

(i) A standardised range of three turbines of medium efficiency level significantly higher than for single stage machines but lower than the optimum in the interests of capital cost. The maximum inlet pressure is 3 100 kPa, maximum inlet temperature 400°C and maximum exhaust pressure 415 kPa. The maximum output of each frame size is 6,5 MW, 3,5 MW and 2,2 MW and the corresponding rotational speeds of 6 000, 7 000 and 9 000 rpm respectively permit economic reduction gear sizes.

Examples of these turbines are the 3 MW turbo-alternator at Dalton, the 3,5 MW turbo-alternator at Sucoma, the 3,3 MW turbine shredder drive at Amatikulu and the new 5,0/4,0 MW turbo-alternators for Sezela, Gledhow, Pongola and Umzimkulu Mills.

(ii) A standardised range of three turbines of higher efficiency for use where this higher efficiency is more important than lower capital cost. The maximum inlet steam pressure is 4 500 kPa, steam temperature 455°C and exhaust pressure 700 kPa. The maximum outputs are 10 MW, 6,5 MW and 3,6 MW. To achieve the higher efficiency, higher running speeds of 8 000, 10 250 and 12 000 rpm are used. These turbines are predominantly for electric power generation and meet the demand for increased electrical power in relation to the low pressure steam demand. Examples of the largest size are the two new 7,25 MW turbo-alternators at Tongaat.

This type of turbine has wide industrial application and is already being further developed for higher outputs and steam conditions.

(iii) Attention was paid to the control system to achieve an economic balance between the requirements of turbine efficiency, control response and cost. It is necessary to keep valve sizes down to reduce actuator sizes and maintain quick governor response. Hence the choice was made of one valve for the smallest turbine and two for the larger turbines in the medium efficiency range of turbines, and four for all turbines in the high efficiency range. The means by which these valves are operated are described later.

(*b*) Standardisation

The basic cost of the turbine and reduction gear can be kept down by restricting the variations in design of the component parts as far as possible to those which are important to the thermal efficiency performance and affected by output and steam conditions.

This policy also assists in providing quicker deliveries through stocking of castings and forgings and spare parts. For example, one rotor can cover as a spare for the 4 MW and 5 MW sets recently ordered for Pongola, Gledhow, Sezela and Umzimkulu.

(*c*) Medium Efficiency Turbines (designated SLC)

(i) General construction

The keystones of this design were an efficiency significantly higher than for a single stage machine, low first cost, and robustness. The maximum steam inlet conditions were important, 3 100 kPa pressure, 400°C temperature at inlet, 415 kPa exhaust, since they permit plain carbon steel casings and rotor and do not present large thermal expansions. Five and four pressure compounded stages and moderate blade speeds were chosen which allowed a simple rotor construction with separate shrunk-on and keyed wheels. These in turn allow wheel material to be chosen independent of the shaft, and construction and assembly to proceed individually up to the final assembly time.

A maximum of two governor controlled nozzle valves were chosen, located in a side separately mounted steam chest to which is bolted an emergency-valve chest. The separate steam chest allows a very simple cylinder design with attendant benefits in production and cost. The top half cylinder can also be removed without breaking inlet or exhaust pipes which is simpler for maintenance. Supporting arrangements are as simple as possible consistent with robustness. The exhaust branch at the side allows the turbine to be installed at ground level if necessary, on simple foundations.

Both the 14 and 18 frames use a double panting plate to support the steam end. The largest 22 frame, in view of its size and because it has to deal with very large pipe sizes at its maximum power of 6,5 MW, has centre line and wider-spread supports at both steam and exhaust ends.

When epicyclic gears of the Allen-Stoekicht design are used it is necessary to use a lightweight gear tooth type flexible coupling to connect the turbine rotor and the comparatively lightweight sunwheel. In these cases, centre line stool support is provided at the exhaust end for all sizes. The maximum output of 6,5 MW for the 22 frame makes the epicyclic gear an attractive choice here with its smaller and more compact in-line arrangement. Fig 1 shows the sectional arrangement of the frame 22 turbine.

(ii) Control and Lubrication Systems

For the smallest turbine the package governor is powerful enough to operate the one control valve directly. By making the emergency valve combined with the stop valve and mechanically tripped, there is no need for a control oil system. The main lubricating oil pump is driven from the turbine shaft.

The middle size has two control valves and requires more power than is available directly from the governor. Hence a hydraulic relay fed by 620 kPa oil is used to amplify and convert the mechanical signal from the governor into a hydraulic one to the control valves.

For the largest size a separate control oil supply is used at a higher pressure of 1 700 kPa, to keep down the amplifying relay and actuator sizes. An accumulator is used to supply transient oil demands to keep the pump size and power down.

For these two turbines the emergency valves are oil operated from the control oil system.

Lubricating oil is fed by a separate pump driven from the reduction gear.

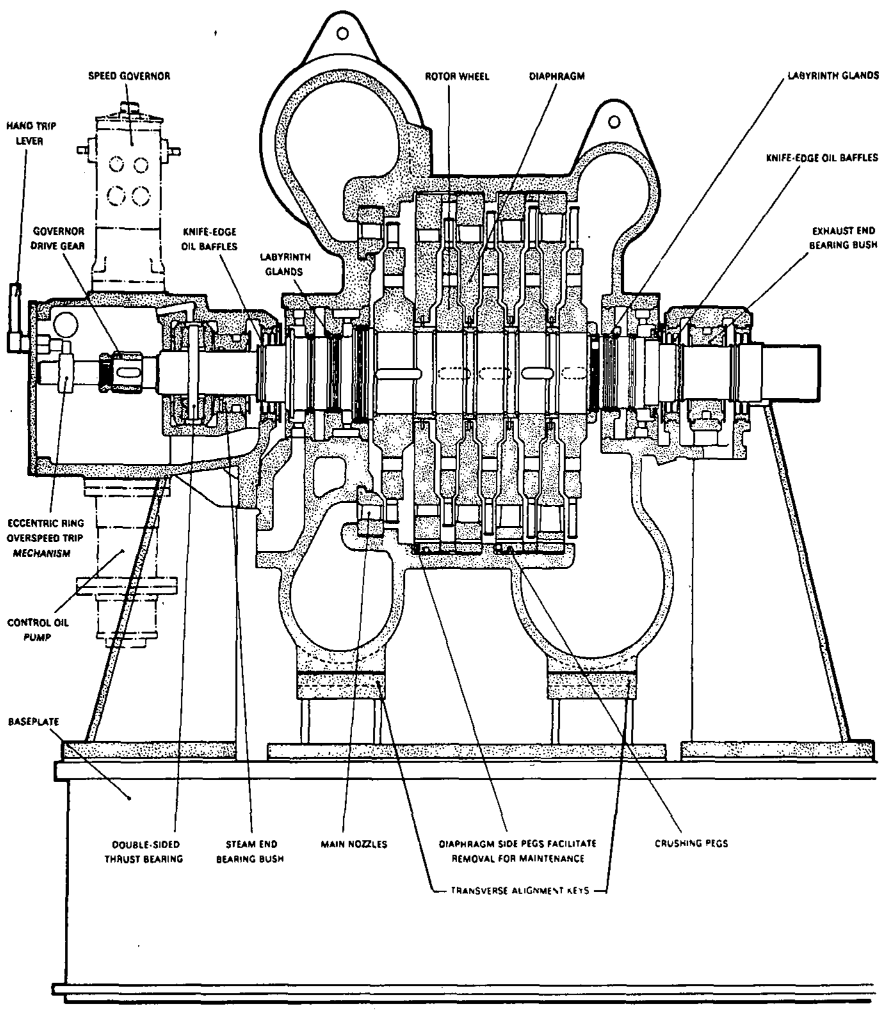

FIGURE 1: Longitudinal sectional arrangement of SLC 22 turbine.

Auxiliary oil pumps for control and lubrication are usually motordriven, with alternative turbine drives available.

When these turbines are used for mechanical drives with a wide operating speed range e.g. 70 %, it may not be practicable to use main oil pumps driven from either the turbine shaft or the gear without the pumps being excessively large at normal speeds. In such cases, either separate or additional motor or small turbine driven pumps are required.

The control and lubricating oil supplies are drawn from the common baseplate/oil tank. Suction strainers, pressure filters (75 micron filtration) and water cooled oil coolers are provided. The package governor uses a heavier oil than the turbine lubricating oil for performance reasons, but this is of small quantity (1,4 litres) and needs infrequent changing. It is a standard Grade S.A.E. 30 automobile oil so that supply and storage is no problem.

(*d*) High Efficiency Back-Pressure Turbines (designated HES)

(i) General construction

Above 6,0 MW, there is more justification for the highest turbine efficiency to maintain economy in bagasse fuel. Thus it makes sense to use the higher steam conditions to achieve this increase in power but without going outside the limits of comparatively simple boiler materials and water treatment.

To keep the turbine size, and hence cost, down, the rotational speed is increased enabling only a few extra turbine stages to utilise the increased heat energy available at higher efficiency. Fig. 2 shows a longitudinal section through such a turbine, of which type the two new 7,25 MW sets installed at Tongaat with inlet steam at 3 200 kPa 365°C are examples. The major differences compared with the medium efficiency turbines just described are a greater number of stages, higher rotational speed for a given diameter, "solid" rotor construction in low alloy steel to cope with the higher wheel stresses but still "stiff", and four automatic nozzle control valves.

The same policy of maximum standardisation has been followed.

The higher speeds and powers for these machines, make epicyclic gears an attractive proposition. These have been well proven now for 25 years of industrial and marine service and make a very compact in-line arrangement. They have been in service in the S.A. sugar industry at Amatikulu, Darnall, Felixton and elsewhere for about twelve years.

Fig. 2 shows a longitudinal section of the turbine coupled to an epicyclic gear.

(ii) Control and Lubrication System

Higher efficiency and power at economic cost has led to the development of more power from a given frame size of turbine and alternator rotor, and gears. For example, approximately the same size of turbine delivering 4 MW at Amatikulu is now producing 7,25 MW at Tongaat, and with appropriate steam conditions has been further developed to produce about 15 MW. Thus the acceleration rate of the system is much greater. The resulting requirement for fast governing response has been achieved in the Allen-Regulateurs Europa control system. The same basic type of "package" speed governor is employed but again this is no longer able to operate the four valves directly hence an intermediate relay and power valve

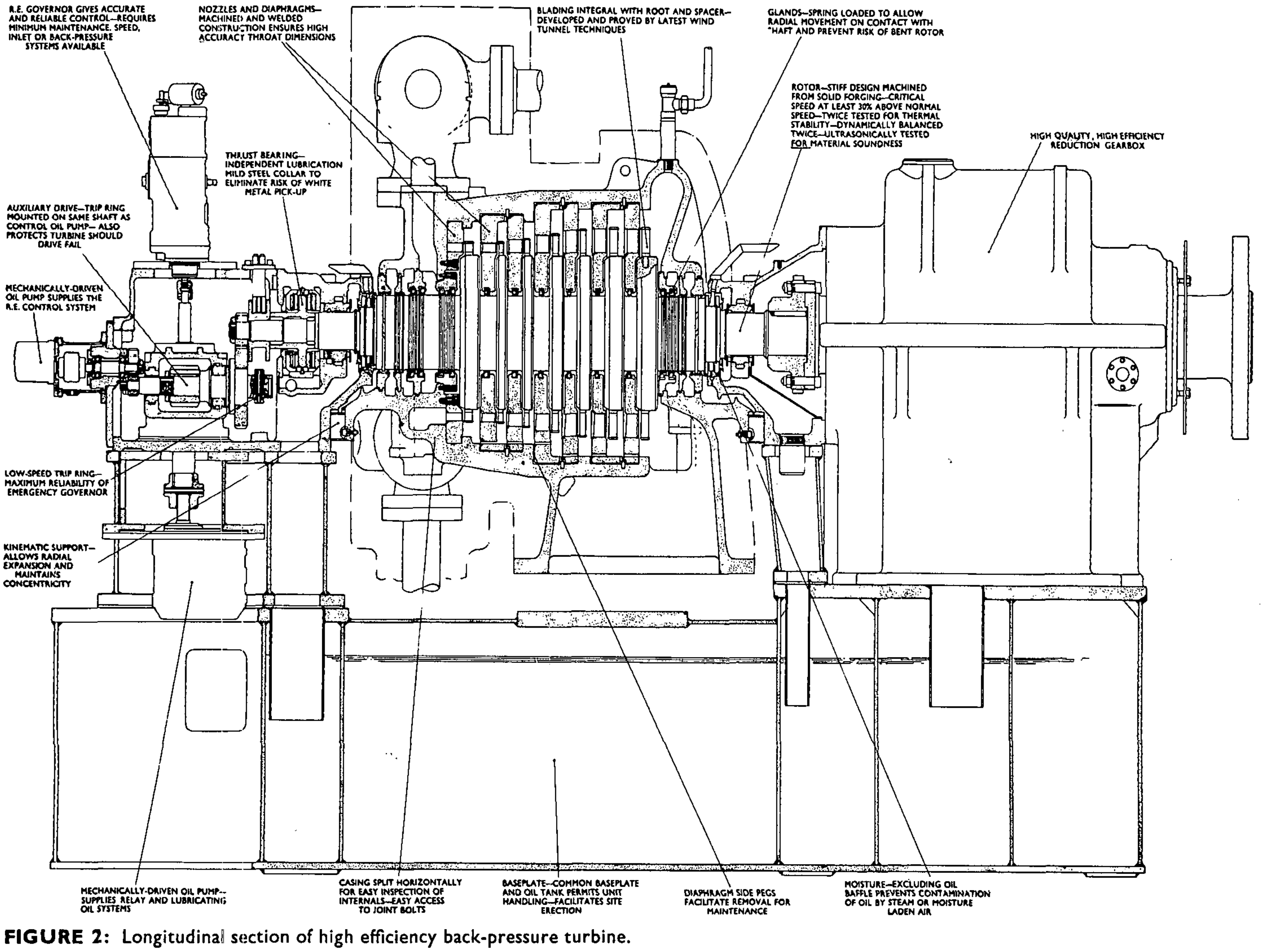

FIGURE 2: Longitudinal section of high efficiency back-pressure turbine.

actuator are used. High pressure oil at 10 000 kPa is used to keep down the size of valve and actuator and ensure rapid operation and enough force to overcome friction forces. This oil pressure is modulated by the speed response of the governor in a directly operated modulating oil pressure valve. The steam control valve is servo-operated in the actuator by a relay with positional feed back which keeps the steady high pressure oil consumption to a very low level. Thus only a small output control oil pump is necessary with an accumulator to supply transient demands. The control oil system is kept separate from the lubrication system, though using the same grade of oil, and is provided with separate small oil coolers and finer filtration (10 micron). Thus cleanliness and freedom from possible water contamination is maintained.

This governing system has proved an outstanding success since its introduction in 1970. The latest 5, 8 and 10 MW units in service with the British Sugar Corporation use it as will the 12 MW set now under construction.[2]

(*e*) Other Types of Multi-Stage Turbines

(i) Topping-Type Back-Pressure

Where there is a need to install new turbines for mechanical drives and at the same time more electrical power is required, these two requirements can lead to the installation of a topping turbine. More steam is generated at high pressure than is required for the mechanical drive turbines and the balance is passed through a back-pressure turbine driving an alternator and exhausting at a pressure just above that of the inlet pressure to existing turbines. Additional power is thereby generated. Such an installation exists at the Darnall Sugar Mill.

However for such an installation certain factors must be understood and followed where necessary.

(*a*) The pressure ratio between the inlet and exhaust pressures of the topping turbine is always comparatively small, and so consequently is the heat drop or heat energy available in the turbine for conversion into mechanical thence electrical power.

(*b*) For maximum thermal efficiency, the steam flow must be as specified and designed for. The pressure drops per stage are smaller and the heat distribution in the turbine is more sensitive to variations in the steam flow.

(*c*) The consequence of departures from the specified design parameters have a correspondingly larger effect on the thermal performance of the turbine, particularly the first stage. For example, reduced steam flow from the boiler will lead to a throttling loss the consequences of which will depend on the degrees of nozzle control provided. Reduced inlet pressure, at the same time maintaining the exhaust pressure, reduces not only the pressure ratio and hence the heat energy available in the turbine, but also the flow capacity of the turbine main nozzles. A reduced inlet steam temperature will also have a marked effect on the heat available. Such a reduction will seriously affect the power produced.

(*d*) The steam should be free from any impurities such as boiler salts or sugar which would adhere to the turbine blading and block up the passage areas. This not only reduces the steam throughput, but also by raising the stage pressure levels in the turbine, increases the end thrust on the blades and wheels and thus on the turbine thrust bearing.

These effects should be clearly understood in order to achieve the expected benefits and to avoid apparent shortcomings in operation.

3. *Single stage turbines*

This type of turbine, employing a velocity compounded two blade row wheel, is still very much in favour for knife, shredder and mill drives up to say 1 500 kW. Below this output the difference in efficiency compared with the medium efficiency multi-stage turbine narrows, but there is a big margin of capital cost in its favour. However, increase in steam conditions does not favour the single stage turbine in efficiency hence above about this power the medium efficiency multi-stage turbine is increasingly favoured.

It is assumed that there is familiarity with the application of single stage turbines and therefore it is not necessary to go into them further.

4. *System components*

(*a*) Governors

The usual method of turbine speed control is through a mechanical flyball type governor whereby variations in speed are reflected in change in the output signal, either by movement of a lever or change in hydraulic oil pressure. This governor is mechanically driven from the turbine shaft. The full load change corresponds to a definite permanent variation of speed, known as droop, and is usually about 4%. Smaller variations in load produce a proportionate permanent change in speed within this limit. Remote control is effected electrically through a motor attached to the governor or as is common nowadays by a pneumatic signal acting on the governor setting device.

An alternative electrical method of governing now available is to sense turbine speed through an electrical counter, process it through an amplifier and speed controller unit, and then feed the resulting electrical signal to an electro-hydraulic actuator where it is converted into a hydraulic output to operate the steam control valve.

The mechanical system has the advantages of a well tried and proven method simple to understand, adjust and repair. It is very responsive, different variations being available to suit particular needs. It is economic, and in its simpler versions as used to control the medium efficiency multi-stage and single stage turbines described it requires no more oil than the electrical system. The latter may claim an advantage in being independent of shaft drive and therefore the master speed controller unit or electric governor can be located off the turbine. However it is dependent on an external power supply. For more complicated control systems where the turbo-alternator runs in parallel with a public supply and import or export load control is required, the electrical signal can be imposed on the electric governor to control the turbine load. However it is only the speed sensor which changes. Hydraulic actuation of the steam valves is still necessary.

The multi-stage turbines described in this paper make full use of the mechanical system of governing which requires no specialist electronic engineers to diagnose or correct wayward behaviour, and the wearing parts are simple and economic to replace. The "package" type speed governor, either Woodward or Regulateurs Europa, is used as the speed sensor. Such governors are produced for a wide market, and a wide range of functions are available with precision of control. This means that the cost is economic and examination/repair facilities are widely available with repair-by-replacement a feasible proposition.

Pneumatic controls are well suited to these governors, and controllers and converters are now readily available and are economic. For electric import/export load control, an electric/pneumatic converter can be used to provide a modifying signal to the main governor controller. The Regulateurs-Europa package type speed governor used on the high efficiency turbines can be used for pneumatic and/or electric speed control as well as for manual overriding control.

For all turbines the speed response to load changes is fast. A full load to no load droop of 4% means a very small speed change for load changes of a few hundred kilowatts on an alternator drive as caused by the operation of large centrifugals. For knife or shredder drives larger load changes are also quickly responded to. Droop control is also adjustable externally to match the speed characteristics of other machines for parallel running. A wide turn down ratio, i.e. speed range, is available for mechanical drives, but naturally such a requirement must be specified at the time of order.

(*b*) Couplings

A metallic membrane type is sometimes used for connection between the turbine and the reduction gear. It is also used to provide a flexible connection between the gear and alternator when the latter is of the two-bearing type and hence self supported and independent of the gear low speed shaft. This type of coupling which has also now proved itself on the Tongaat shredder is now in common use with several important advantages over the more conventional gear tooth type. The main ones are that it permits axial, angular and parallel misalignment, in excess of that required, without creating undue end thrust or radial bearing loads. Other important features are that no lubrication or maintenance are required, it has a low initial cost, and will operate in high ambient temperatures and corrosive atmospheres and has low stress levels giving long working life.

(*c*) Reduction gears

Speed reduction is necessary between the turbine and the gear for the powers involved in these applications in order to keep the turbine sizes down and hence economic. Two basic types are available — parallel shaft and epicyclic.

For powers above say 3 MW, epicyclic gears are competitive in cost and certainly save space from their in-line arrangement. This can be important in installing a machine of higher output or an extra machine into existing power houses, many of which were designed and built long ago.

(*d*) Foundations

Until comparatively recently, turbine foundations for multistage turbo-alternators were of the fairly massive reinforced concrete block type, with spaces only for steam pipework and alternator air ducting as required. In the modern trend for reduced costs, these have now become table top type i.e. a base supporting the turbo-alternator with itself supported on a number of legs resting on a solid base foundation block.

It is essential therefore that the natural frequencies be carefully calculated in the design stage to make sure that they do not cause resonance and thereby magnify residual unbalance which may occur in the rotating machines. Admittedly this is not an easy calculation but it is necessary. Should difficulties occur in operation, correction can be an expensive business.

5. *Operation*

(*a*) Steam pipework

Now that more power is being developed from a given turbine size, the steam pipes are increasing in size relative to the turbine. It is very necessary that careful attention be paid to the supporting and thermal expansion arrangements of the pipework in order to keep the resulting forces and movements on the turbine flanges within the levels specified by the manufacturers. Steam turbines with their fine clearances between rotating and stationary elements, should never be regarded as convenient anchorages for the pipework.

Steam pipework must also contain adequate provisions for water drainage points not only to prevent water being carried into the turbine but also to allow for adequate flow of steam for pipe warming at start-up.

(*b*) Corrosion prevention

With more than one machine in a station connected to a common inlet or exhaust range it may be necessary to have a machine stationary whilst others are running. Then there can be a real danger from effects of standing corrosion, i.e. from the leakage of hot steam into a cold turbine cylinder. Condensation of this steam in combination with absorbed oxygen can cause severe corrosion of the turbine internals very quickly. There are various methods to combat this.

(i) Effective isolation by double isolating valves with an ample atmospheric drain between them. Single valves are seldom effective.

(ii) Provision for intermittent hot air circulation through the turbine during shut down periods longer than say one day. Such fan units are portable and typical sizes required are approximately 2–3 kW with an air flow of about 200 cfm for connection at the exhaust end, back flowing through the turbine and discharging through glands and opened drains.

(iii) Making the turbine internals of corrosion resistant materials e.g. turbine wheels and shaft (the blading is always of stainless steel), diaphragms, and protecting exposed internal parts of the cylinder.

However, such materials on rotating parts can lead to other problems and this kind of trouble is better prevented by the first measures outlined above and good maintenance.

(*c*) Fouling

Low boiler pressure due to erratic firing can cause carry over of water under a high load demand to a degree not catered for by separators and drains. Extra end thrust on the turbine blading and hence thrust bearing is generated, often sufficient to wear the thrust bearing. This can then lead to rotor movement and allow contact between rotating blades and stationary diaphragms with often serious consequences.

The carry over of boiler salts from contamination of the feed water (and inadequate control of boiler water dissolved solids) can lead to choking of the blade passages. This can also overload the thrust bearing with consequences as previously mentioned. If the deposits are water soluble such as sugar they can be removed by controlled injection of washing condensate through tappings on the inlet and outlet steam piping and can be carried out with only a temporary off-loading. Silica deposits are tenacious and can generally only be removed by mechanical cleaning during the off season.

Water for cooling alternator air coolers and turbine oil coolers must be kept free from foreign matter otherwise overheating with serious consequences may result.

(*d*) Speed control

Turbo-alternators are rated for a constant speed with a speed setting regulation of $\pm$ 6% about this level. On no account should lower levels than these be used for long term running otherwise the lower voltage level will probably cause the automatic voltage regulator to be operating at its extreme or even beyond. This may cause over-excitation of the alternator leading to overheating of the rotor through excess current.

(*e*) Daily routines

It is essential for long turbine life that careful attention should be paid to the daily routines such as:

(i) regular, accurate log readings;

(ii) quick investigation of readings which have changed for no apparent reason, particularly those on the lubricating oil system and alternator cooling.

(iii) regular inspection of lubricating oil for water contamination and regular cleaning of strainers and filters on lubricating oil and water systems. These periods to be adjusted to suit local conditions, but preferably on a routine basis.

Acknowledgement

The author records his thanks to the directors of Amalgamated Power Engineering Limited for permission to publish this paper, and his colleagues for their assistance.

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