Feature

Steam turbines: how big can they get?

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It is commonly believed that the specific costs of power plants fall as unit capacity increases (Figure 1). But, in reality, near the maximum size attainable the curve becomes almost horizontal, and specific costs can even increase as we approach this point. In other words summits are rarely achieved without pain. Nevertheless it is human nature to aspire to ever bigger unit sizes and every time a breakthrough in scale is achieved it soon becomes the new benchmark for the industry.

Although the steam turbine is a very mature technology, it continues to evolve and improve, and even in the last decade there has been great progress – in steam path design and in the adoption of elevated steam parameters – described by Dr Wilfried Ulm of Siemens as "an almost unnoticed revolution in steam turbine technology."

For the best modern turbines, the internal efficiencies of their high pressure (HP) and intermediate pressure (IP) cylinders/sections can reach 94% and 96%, respectively, or even greater, and with the application of advanced steam conditions the gross efficiencies of the most efficient steam turbines approach 50%. Even wet steam turbines used in nuclear plants, fed with saturated steam, can achieve efficiencies as high as 36% or more.

"Good old steam turbines", with their high reliabilities, remain the workhorses of the power industry.

The cross–compound solution

The main factor limiting the unit size of condensing steam turbines is their annular exhaust area, determined by the available length of the low pressure (LP) last stage blades (LSBs) and the number of the exhaust flows. In turn, the maximum LSB length is limited by the strength of the blade and its ability to withstand centrifugal stresses in the root section.

Since the early 1960s, the quest for increased unit size frequently led to use of the cross–compound (CC), ie double shaft, configuration to provide the required large exhaust area without an excessive increase in LSB length and without adding unduly to the number of cylinders on the turbine shaft. This approach was particularly favoured in the USA, with its 60 Hz grid frequency. The centrifugal stresses in the LSBs of a full speed 60 Hz turbine, with a rotation speed of 3600 rpm, would be 1.44 times greater than those in a 50 Hz machine with a speed of 3000 rpm. So, with the LSB technology available in the 1960s/early 70s, a supercritical-pressure steam turbine for 3600 rpm, with steam conditions of 24.7 MPa, 538/538°C (typical for those years), and unit capacity of about 1300 MW, would need four double flow LP cylinders. If these were attached to the same shaft together with the HP and IP cylinders, the result would have been a monstrous caterpillar-like six-cylinder machine with numerous potential problems in service. In addition, with the technology of the day, it would have been rather problematic to manufacture a generator of such an output.

The solution adopted at that time by Brown Boveri Company (BBC, later merged into ABB, then evolving into ABB Alstom and finally Alstom) was to employ two full speed shafts, with the six cylinders evenly distributed between them: one high temperature (HP or IP) cylinder and two LP ones on each shaft. The first turbines of this type, installed in 1972-3 at the US power plants Cumberland and Amos, had LSBs with a length of 762 mm (30 in). This was increased to 787 mm (31 in) for subsequent machines of this type, installed at Mountaineer, Rockport, and Zimmer. These turbines remain the largest in service in fossil
fuel power plants.

This approach does not look optimal and is obviously inferior in effectiveness to the CC turbine concept, with all the LP cylinders located separately on the half speed shaft, making it possible to use much longer LSBs and to reduce the number of LP cylinders. An example of a modern CC turbine for fossil fuelled power plants (a 1000 MW class MHI machine) can be seen in Figure 2. Modern large CC turbines, mostly for 60 Hz, with unit sizes in the range 800-1050 MW, follow this approach, with two double flow LP cylinders on the half speed shaft and the HP and IP cylinders on the full speed shaft. MHI steam turbines of this class have LSBs with a length of 1169 mm (46 in) long. Examples are the 1050 MW turbine with steam conditions of 24.5 MPa, 600/610°C at Tachibana-wan unit 2, commissioned in December 2000 (see MPS, November 2001, pp 41-47), and two earlier 1000 MW turbines, with somewhat lower steam temperatures, at Misumu unit 1 and Matsura unit 2.

The scheme used in these machines makes it possible to cope with large volumes of steam flow through the LP exhausts, without an excessive increase in the number of them, and simultaneously provides high internal efficiency in the full speed HP and IP cylinders.

**Tandem–compound technology**

Nevertheless a tandem–compound (TC), ie single shaft, configuration would appear preferable in terms of capital expenditures and service requirements. Recent achievements in creating longer LSBs allow full speed 1000 MW TC turbines to be designed for 60 Hz, ie with a speed of 3600 rpm.

In particular, MHI has announced a new standard series of integrally shrouded LSBs, including 1143 mm (45 in) titanium blades for 3600 rpm and 1220 mm (48 in) steel blades for 3000 rpm. On this basis, MHI declared its readiness to produce a 1000 MW class 3600 rpm four cylinder turbine with four LP flows (TC-4F45) as an alternative to the CC configuration that has been used in recent years.

The 48 in steel LSBs have in fact already been used, in the 600 MW 3000 rpm TC turbine of the Hirono 5 plant in Japan, commissioned in 2004. The Hirono 5 turbine consists of two cylinders: an integrated HP–IP cylinder; and a double flow LP cylinder. This 600 MW turbine (TC-2F48) would appear to be the largest two cylinder machine in service and has the world’s biggest output per LP exhaust. According to MHI, in the future it plans to increase the unit capacity of such TC two cylinder turbines up to 750 MW. Thus the output of a three cylinder machine (HP–IP cylinder + two LP cylinders) could be pushed to 1000 MW.

As of 2004, the largest TC steam turbines produced by Japanese manufacturers were the two MHI supercritical-pressure turbines with a rated output of 735 MW each installed at Ratchaburi 1 and 2 in Thailand, which entered operation in 2000. The maximum gross output of such a turbine is up to 841 MW, with a speed of 3000 rpm. The configuration of this turbine is similar to the proposed 1 000 MW turbine: a four cylinder machine with double flow HP and IP cylinders and two double flow LP ones.

The first 1000 MW full speed TC turbines for 60 Hz (that is, with a speed of 3600 rpm) were manufactured by Toshiba and went on line at Hekinan (units 4 and 5) in 2004.

The readiness to produce TC steam turbines of the 1000 MW class has also been announced by Hitachi – as an alternative to CC machines, such as those recently installed, for example, at the 1000 MW 50 Hz units Haramachi 2 and Hitachinaka 1.

Even though these turbines would have shorter LSBs than the MHI LSBs mentioned above, they will also have only two double flow LP cylinders. They will be furnished with 1016 mm (40 in) long LSBs for 3600 rpm and 1092 mm (43 in) long LSBs for 3000 rpm. The same 40 in titanium LSBs have been used in Hitachi’s standard 700 MW three cylinder (HP–IP + two LP) turbines for Japanese 60 Hz power plants and are also to be used in the 495 MW turbine of the first Canadian supercritical power plant, Genesee 3 (with one double flow LP cylinder – TC2F-40), and the 870 MW turbine for the CBEC 4 supercritical plant (with two double flow LP cylinders – TC4F-40). The latter is the first of a new wave of supercritical-pressure units to be launched in the USA after a long hiatus (see MPS, April 2004, pp 33-37).
Their analogue for 50 Hz applications is a 700 MW three cylinder turbine with steam conditions of 25 MPa, 600/600 °C and 1092 mm long LSBs (TC-4F43). A similar 700 MW 3600 rpm turbine, with 593 °C (1100 °F) main and reheat steam temperature, was produced by Toshiba for the Nakano plant. For the time being, these 700 MW turbines can claim to be the world’s largest machines in service using integrated HP–IP cylinders.

The largest TC steam turbine for fossil fuelled power plants with a grid frequency of 50 Hz was manufactured as long ago as the late 1970s by LMZ of Russia. This turbine, with a rated output of 1200 MW, maximum continuous rating (MCR) of 1380 MW, and steam conditions of 23.5 MPa, 540/540°C, has been in operation since 1979 at the Kostroma power plant (unit 9). The turbine consists of five cylinders (loop flow HP, double flow IP, and three LP). The LP sections have titanium LSBs of 1200 mm in length.

Up to 2002, the largest TC turbines operating in Western Europe at fossil fuelled power plants had a unit gross capacity of 933 MW. These were two ABB five cylinder turbines for Lippendorf, a lignite fired plant in Germany, which has steam conditions of 25.9 MPa, 550/580°C. As distinct from other modern turbines mentioned above, with their integrally shrouded LSBs, these turbines have free-standing LSBs (without shroud or interblade ties) of 1050 mm (about 41 in) in length.

With a net efficiency of 42.4%, Lippendorf was for a time, the most efficient solid-fuelled power plant in the world. In 2000, this level was exceeded by the Boxberg Q unit, with a net output of 907 MW and steam conditions of 26.6 MPa, 545/581°C. According to heat rate field tests, its net efficiency was estimated to be 42.7% (see MPS, October 2001, pp 21-23).

As to the unit capacity of the Lippendorf turbines, this level was surpassed by another Siemens steam turbine, that of Niederaussem K (Figure 3), which entered commercial operation in autumn 2002. It has a gross output at the generator terminals of 1012 MW (with a net capacity of 965 MW). The turbine configuration, with a single flow HP cylinder, double flow IP, and three double flow LP cylinders, is identical with that of Boxberg Q. The main difference is that the LSBs were increased in length, from 978 mm (38.5 in) in the Boxberg turbine to 1146 mm (45 in) at Niederaussem K, increasing the annular exhaust area by 25% – from 10.0 m2 per flow up to 12.5 m2.

Using these longer LSBs, but in warmer cooling water conditions, has enabled Siemens to offer a 1000 MW turbine with only two double flow LP cylinders for the Yuhuan units in China (see MPS June 2005, pp 27-31).

A transition to titanium 1423 mm (56 in) LSBs, with an annular exhaust area of 16.0 m2 per flow, has enabled Siemens to design a 600 MW turbine for the North Rhine Westphalia Reference Power Plant consisting of only three cylinders – ie with just one double flow LP cylinder – without sacrificing any efficiency.

This opens up the possibility in principle of increasing the output of a five-cylinder turbine to 1700-1800 MW.

**Nuclear, wet steam machines**

In contrast with steam turbines for fossil fuelled power plants, all wet steam turbines for nuclear power plants, including those of the highest unit capacity, are of the single shaft, or tandem–compound, type.

This is mainly explained by the fact that both the HP and LP sections of these turbines work with wet steam. The rotating blades of both sections are exposed to water drop erosion (WDE) and similar approaches to coping with the problem are required in each section, bearing in mind that the WDE rate is proportional to circumferential speed.

If an extremely large exhaust area is needed to achieve the required steam flow, the turbine has to be designed to be half speed, that is, with a four pole generator. The main problem with such low speed machines is their manufacture, requiring heavier machining facilities, larger areas, more advanced technologies, and greater capital expenditures.

An advantage of high speed turbines is their higher internal efficiency in the HP stages, with longer vanes and buckets. At the same time, according to some estimations, over a
unit capacity of about 1070 MW the specific metal consumption (the turbine’s total metal mass divided by its output) for a half speed turbine comes close to that of a full speed turbine of the same output (about 2 kg per kW).

The maximum output achievable for full speed (high speed) wet steam turbines without excessive exhaust losses on one hand or an absurd increase in the number of LP flows on the other, is, as with other turbine types, not fixed but depends on available LSB lengths, which tend to grow over time as the technology improves.

For 50 Hz, the limit currently lies at a little over 1000 MW. For 60 Hz the level is about 1.5 times less.

Attempts so far to force the level up (eg, by using a Baumann stage) appear ineffective. But the use of titanium LSBs with a length of 1500 mm (59 in) would enable a full speed (3000 rpm) wet steam turbine to achieve a unit capacity of about 1250 MW with two or three double flow cylinders, depending on the cooling water temperature.

The largest high speed wet steam turbine to enter operation to date is rated at 1032 MW. This is the 3000 rpm Siemens turbine at the Trillo nuclear plant in Spain. It consists of one double flow HP cylinder and three double flow LP cylinders with 1118 mm long LSBs (TC-6F44).

LMZ’s standard 1000 MW high speed wet steam turbine has one double flow HP cylinder and four LP cylinders with titanium LSBs 1200 mm in length (TC-8F47). A larger high speed wet steam turbine, rated at 1200 MW, was proposed by BBC for the Graben nuclear plant in Switzerland, but this project never materialised.

Transition to the low speed (half speed) concept, 1500 rpm in the case of 50 Hz applications, allows the turbine unit output to be increased by a factor of at least 1.5. The largest such low speed wet steam turbine presently in service, Alstom’s Arabelle, has a rated gross output of 1550 MW (see Figure 4). Four such turbines have been in operation since the late 1990s at the Chooz B and Civaux nuclear plants in France.

The Arabelle turbine has one integrated HP–IP cylinder and three double flow LP cylinders. The LSBs are 1450 mm (57 in) long, giving a total annular exhaust area of 115.2 m2. The same area could now be achieved with four exhausts, using the 1830 mm (73 in) LSBs developed by Alstom in more recent years. Such a turbine, with two LP cylinders, would be about 6 m shorter.

On the other hand, a six exhaust turbine employing the 1830 mm LSBs would have a total annular exhaust area of about 172 m2. This would reduce the energy losses associated with the exit steam velocity to tiny values, despite an enormous steam flow rate.

The Arabelle HP and IP sections are both single flow, which results in higher efficiency thanks to longer blades and reduced secondary energy losses.

In contrast Alstom’s low speed turbine design of the same size class for 60 Hz applications, with a unit capacity of 1200-1500 MW, uses a double-flow HP cylinder plus three LP cylinders. This is because of the greater rotation speed (1800 rpm) and, as a result, the necessity to limit the HP stage dimensions, keeping in mind the WDE threat. With a 1194 mm (47 in) LSB, which provides a circumferential speed close to that of Arabelle, such a turbine has a total annular exhaust area of only 80.4 m2, giving significantly greater exhaust energy losses than its European, 50 Hz, counterparts.

For comparison, even to achieve a total annular area of as little as 74.4 m2 with a full speed 60 Hz turbine of the same unit capacity would require 12 exhausts (that is, six LP cylinders) with an LSB length of 852 mm (33.5 in), resulting in a considerably greater circumferential speed.

To provide a somewhat smaller unit capacity, around 1000 MW, with a speed of 1800 rpm, Alstom has developed four cylinder turbines, with one single flow HP cylinder and three double exhaust LP cylinders. Such machines have been supplied to South Korean nuclear plants.

Arabelle’s unit capacity record will be overtaken by the Siemens 1500 rpm turbine being
supplied to Finland's Olkiluoto 3 EPR plant, currently under construction, which will have a rated output of 1720 MW gross (see MPS, August 2004, pp 43-46). The Olkiluoto turbine (Figure 5) will comprise one double flow HP cylinder and three double flow LP cylinders with 1675 mm (66 in) long titanium LSBs, providing an annular exhaust area of 25 m² per flow (150 m² in total). The turbine is furnished with two vertical moisture separator reheaters instead of horizontal ones (as situated on either side of the Arabelle units), which makes it more compact.

Not to be outdone an advanced version of the Arabelle turbine (Figure 6), with a rating of 1750 MW gross, is to be supplied by Alstom to the Flamanville 3 EPR plant in France, due to start up in 2012. This will also use vertical MSRs.

For the 60 Hz US market, MHI is proposing a 1700 MWe version of its APWR, with a steam turbine having a 70 in integrally shrouded LSB (compared with 54 in in the case of its Japanese APWR design) – see MPS December 2006, pp 33-35.

In October 2006 it was announced that under an agreement with GE, Doosan Heavy Industries and Construction will provide two 1455 MW 1800 rpm steam turbines for two units of the Shin Kori plant. These will be the largest 60 Hz steam turbines in the world when they enter service in 2013.

Meanwhile, the largest 1800 rpm (60 Hz) wet steam turbines in operation to date are the Hitachi machines at Hamaoka 5 (1380 MW) and Shika 2 (1358 MW), which entered service in 2005/6.

These turbines consist of one double flow HP cylinder and three double flow LP cylinders. The LSBs are 1320 mm (52 in) long, providing an annular exhaust area of 16.7 m² per flow.

**Blade design challenges**

The increase in blade length inevitably reduces the aerodynamic qualities of the LSBs and makes their design more complicated because of the large length-to-mean-diameter ratio and the increased pitch of the meridional stage profile. The maximum length-to-mean-diameter ratio for the longest full speed LSBs today reaches 0.40-0.41, whereas for half speed LSBs it remains less than 0.36. The closer the LSB is to its maximum length, the smaller the gain in efficiency and the higher the costs. With an optimal circumferential-speed-to-steam-velocity ratio, the increased mean diameter means an increased enthalpy drop and, as a result, a greater difference in the specific steam volume values between the blade row entrance and exit. High, supersonic, steam velocities and their great variations lengthwise with row height hamper the attainment of optimal aerodynamic performance. Of particular importance is that at low-flow operating conditions the last LP stages are especially prone to reverse vortex motion of the steam which can seriously threaten blade integrity. This applies not only to the LSBs but also to the second and even third LP stages counted from the exit, and the longer the LSBs and the greater the pitch of the meridional profile, the more probable is the appearance of this reverse vortex motion.

It seems likely that just this phenomenon caused the failure of a rotating blade and high-cycle fatigue cracks at the root prongs of many other blades in the 12th (3rd from the exit) LP stages of the Hamaoka 5 turbine in June 2006 (see MPS January 2007). Comparisons of steam flow patterns in the LP steam paths of the Hitachi steam turbines of Hamaoka 5 and 4, with unit capacities of 1380 MW and 1137 MW and LSBs 52 in and 43 in long, respectively, showed that at 5% load operating conditions the reverse vortex in the Hamaoka 5 case reaches the root zone of the 12th stage causing random vibration of the rotating blades, whereas in the Hamaoka 4 case (with shorter LSBs) the vortex area does not reach this stage. It is obvious that longer LSBs call for more attention to low-flow and high-backpressure operating conditions.

In addition, the erosion effects of wet steam become more pronounced the longer the LP stage blades and the greater their tip circumferential speed. For the longest full speed LSBs, the tip circumferential speed is already up to 750 m/s and could reach 830 m/s for the newly developed machines; for low speed LSBs the speed does not exceed 500-530 m/s.

For full speed LSBs, the only practical way currently to radically increase their length and annular exit area is to move to titanium alloys (for example, Ti-5Al, Ti-6Al-4V, or Ti-6Al-
For many years, the most widespread type of attachment bases for LSBs were prong-and-finger (fork shaped) roots with varying numbers of prongs. For example, Hitachi has made titanium LSBs with lengths of 1016 mm (40 in) and 1092 mm (43 in) with seven and nine prongs, respectively.

However, most LSBs are now designed with curved-entry fir tree roots, and these are currently considered to be the best way of attaching the longest LSBs. The compactness of the dovetails allows a thinner wheel configuration, reducing the centrifugal stress in the rotor body. In addition, the fir-tree roots are free from potential stress concentrators such as sharp edges or pin holes. This is especially important for blades made of titanium alloys, which are relatively brittle and sensitive to notches. A notable feature of the curved-entry fir-tree dovetails is the uniform distribution of load over all the blade root hooks. Insufficient rigidity of the prong-and-finger attachment base, unavoidable presence of stress concentrators, and uneven stress distribution in the root may have contributed to the failure and cracking of the 12th stage rotating blades at Hamaoka 5.

Modern rotating blades, including LP ones, are usually integrally shrouded, that is, made with shrouding elements milled together with the bucket airfoil (profiled body). The shrouding elements of individual buckets are connected together by means of special outside inserts and wedge-shaped grooves in the shrouds like a dovetail joint, or the shroud pieces are designed with special wedge-shaped edges that mesh with the blades when subjected to centrifugal forces. In addition, to increase the rigidity of the entire blade structure, the blades are supplementarily coupled with “snubbers” – integrally formed tie-bosses at the mid-span of the blade height. Their edges also engage under the action of centrifugal forces. As a result, when the turbine rotates, all the stage blades are tied together, forming a continuous ring of blades.

One of the major advantages of such an annular blade structure, compared with blade groups (several units of several blades each connected with wire ties), is that it has fewer resonance points during rotation. The resulting blade structure with two contact supports (tie-bosses at the blade mid-height and integral shroud at the blade tip) provides well defined and easily controlled vibration modes and significantly reduces the buffeting stresses arising when the LSBs are subjected to low-steam-flow and high-back-pressure conditions.

Free-standing LSBs, not connected by shrouds, mid-span damping wire ties, or tie-bosses have also been successfully used (eg by Siemens and ABB). Modern CFD methods combined with extensive model trials, together with precise manufacturing techniques, make it possible to completely eliminate the need for any vibration damping elements, including shrouds. Even though shrouding the blades typically reduces tip leakage losses, this is completely compensated for by the more effective peripheral water separation of unshrouded blades. In turn, mid-span damping devices increase the airfoil thickness in their neighbourhood, considerably increasing the profile losses. In addition, all the obstacles in the interblade channels (like tie bosses or wire ties) disrupt the steam flow and lead to additional energy losses. Of importance also is that any local wetness concentration in the stage channels greatly exacerbates blade erosion. This is a particular issue for wire ties and tie bosses between the blades and provides another reason for using free-standing LSBs, as well as shrouded blades without any additional ties in the preceding stages. But there is a length limitation for free-standing LSBs. So the latest titanium LSBs from Siemens, providing annular exit areas of 16.0 m² and 11.1 m² per flow for 3000 and 3600 rpm, respectively, are characterised by an “interlocked” design and feature an integral shroud, as well as a mid-span snubber.

**Rotors, bearings and thermal expansion**

The low pressure rotors of large modern steam turbines with their long LSBs and large
root diameter experience large centrifugal forces. To withstand them, the LP rotors are solid (forged without a central bore) or welded. Nevertheless, rotor strength can be among the factors limiting turbine output increase.

At turbine start-ups, the steam admission sections of LP rotors can encounter large tensile stresses due to the superposition of centrifugal and unsteady thermal stresses. To protect LP rotors from brittle fracture, their thermal stress state must be monitored during start-up, with the aid of mathematical modelling, as is done for high temperature (HP and IP) rotors. This is especially important for the steam turbines of fossil fired plants with elevated reheat steam temperatures.

As turbine capacity increases, so do the diameters of rotor journal necks and therefore journal bearings too, with a consequent increase in the loads on them. So, for example, in the 735 MW TC steam turbines for Ratchaburi in Thailand MHI used a journal bearing with a diameter of 535 mm (at the generator end of the LP cylinder). This was declared to be the largest diameter journal bearing ever employed in 3000 rpm turbines. But this was not the case as LMZ had already deployed journal bearings of 620 mm and 575 mm in its 1200 MW supercritical-pressure machine and in 1000 MW wet steam turbines for nuclear power plants.

To provide vibrational reliability and decrease bearing friction losses, large steam turbines are often equipped with segmented, or multi-wedge, bearings. In this design, the journal neck interacts not with an entire bush of the bearing but with a few self-adjusted segments, each of which can turn independently. The lubricant is introduced to each segment, forming separate oil-covered wedges, which hold the journal neck. This noticeably increases rotational stability. Segmented journal bearings are however much more complicated and require more accurate assembly compared with more traditional bush journal bearings with an elliptical bore. Also, field experience suggests that in practice segmented bearings are not much more effective than bush bearings. Indeed, improvements in the design of bush bearings has reduced vibrations to a level even lower than that associated with segmented bearings, while at the same time reducing the required lubricant flow and bearing friction losses.

As a rule, the high temperature rotors of modern large steam turbines for fossil fuelled power plants with elevated main and reheat steam temperatures are made of 12%Cr forged steel, which has sufficient creep rupture strength. However, a disadvantage of this steel is its high hardness. For this reason, in the journal and thrust collar sections of such rotors overlay welds are built up with a low Cr weld material to reduce bearing wear. MHI has proposed a different approach: “hetero-material” welded rotors, with the central (high temperature) part made of 12%Cr steel and the ends made of 21/4%Cr-Mo-V steel.

As we have seen, the largest modern TC steam turbines comprise up to five cylinders and, depending on the configuration, the total thermal expansion can be as much as 55 mm relative to the cold state.

The traditional way of dealing with thermal expansion is for the bearing pedestals to slide on the foundation frame along the turbine axis. The axial thermal expansion occurs relative to the turbine’s fixed-point, under the LP cylinder. For turbines with several LP cylinders, each of them usually has a separate fixed-point, and the pedestals of the bearings between the LP cylinders have some flexible elements allowing the cylinders to move slightly relative to each other. The axial movement of the bearing pedestals along the foundation frame is hampered by significant frictional forces on the sliding surfaces. As a result, many large steam turbines encounter serious problems with their freedom for thermal expansion. Hampered thermal expansion can result in distortion of the casings, torsion in the foundation frame crossbars, increased vibration, damage to turbine bearings and couplings, etc. Problems with the freedom of thermal expansion also frequently hamper turbine start-ups because the monitored relative rotor expansion (RRE) values for the HP and IP cylinders reach their limits.

Investigations of a wide range of large steam turbines in service have shown that the major causes of loss of freedom for thermal expansion can be: increased friction at the sliding surfaces between bearing pedestals and foundation frame; increased transversal load on the turbine from steam lines connected to it; poor transfer of axial thrust from one cylinder to another; and insufficient rigidity of the foundation crossbars.

The larger the turbine’s output and the higher its steam conditions, the more massive and rigid the steam lines become and if their thermal expansion is not properly allowed for...
they can limit the turbine's ability to accommodate temperature changes. The effect is particularly problematic when the steam lines are asymmetric relative to the turbine axis, eg when the boiler is on one side of the turbine.

Typical countermeasures to accommodate turbine thermal expansion include the placing of special bands or removable plates on the sliding surfaces under the bearing pedestals, electrochemical treatment of the key surfaces, and adjustment of the support and suspension systems for steam lines. Some manufacturers also furnish their turbines with special rods to transmit the pushing and pulling forces directly from one cylinder to another.

Another approach, increasingly used, is to have the bearing pedestals rigidly mounted on the foundation frame, but to allow the outer casings of the HP and IP cylinders, as well as the inner casings of the LP cylinders, to slide in the axial direction, with the aid of longitudinal keys on the bearing pedestals. The common anchor point and the origin of the axial thermal expansion is the pedestal of the intermediate bearing between the HP and IP cylinders, which is designed as a combined journal-and-thrust bearing. With this arrangement, both the casings and rotors of the HP and IP cylinders expand or contract in unison and this decreases variations in the RRE for both cylinders, decreasing changes in axial clearances in the steam paths. As to the LP cylinders, their outer casings rest on the condensers while a system of push rods connects all their inner casings together so that thermal expansion of the LP rotors and the inner casings is in the same direction, thus reducing RRE and variations in axial clearances.

This system has been used in the 1000 MW class Siemens steam turbines at Boxberg Q, Niederaussem K, and Yuhuan.

The problem of thermal expansion is lessened of course if the overall length of the turbine can be lessened, by reducing not only the number of cylinders but also the number of bearings. It is worth noting that for some years there have been large steam turbines with journal bearings common to adjacent cylinders. With these designs a five cylinder 1000 MW class machine for fossil plant applications has six bearings instead of ten. The downside of common bearings is that their assembly and disassembly is more complicated.

Where now?

In principle modern technologies allow the design of steam turbines with a unit capacity of up to about 2000 MW – specifically 50 Hz wet steam turbines for nuclear power plants (provided, of course, a nuclear reactor of sufficient power is available).

There already exist, or will be available in the very near future, steel LSBs with a length of 1800-1850 mm (71-73 in) for 1500 rpm. They allow construction of a four cylinder turbine with six exhausts providing a total annular exhaust area of 168 m2, about 10% more than a five cylinder, TC-8F57, machine.

The LP rotor for such LSBs will have a body diameter of about 3000-3150 mm and, being of a welded type, will need very heavy billets, of as much as 40-45 t in mass. Manufacturing such billets and transporting such welded rotors by rail will encounter serious problems. It is not surprising that currently the specific costs associated with such a turbine would exceed those for units of smaller capacity.

The same could be said for high speed (3000 rpm) turbines for fossil fuelled power plants, when compared with the 1200 MW five cylinder supercritical-pressure machine. Transition to elevated main and reheat steam temperatures, of up to 600 °C, and the use of six and eight LP flows with 1400 mm long titanium LSBs would allow the output to be increased to about 1350-1400 MW and 1700-1800 MW, respectively. But it would not result in reduced unit costs, as yet. However if the patterns of the past are followed such large machines, once established, will become economically attractive in the future.

Source: Alexander S Leyzerovich, consultant, Mountain View, CA, USA