Power Engineering

Gas Turbine Combined Cycle Fast Start: The Physics Behind the Concept

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Nowadays all major gas turbine OEMs promote their products with an emphasis on "flexibility" in addition to output and efficiency. The most advertised flexibility feature is the fast start capability of advanced F, G or H class machines in simple and combined cycle modes. Alas, modern gas turbine based combined cycle (GTCC) systems comprise steel behemoths weighing tens of thousands of pounds and operate at extremely high pressures and temperatures while connected to each other via a maze of pipes and valves. This complex architecture presents formidable challenges to designers and operators alike to handle major operational transients with large flow, pressure and temperature (FPT) gradients without adverse impact on reliability, availability and maintainability (RAM). This is primarily achieved by advanced control schemes incorporating model based controls (MBC), design features such as terminal attemperators and cascaded steam bypass as well as material selection. As a result, in terms of dynamic response to transient events, the difference between a modern GTCC and its forerunners is as pronounced as that between cars with carbureted vis-à-vis fuel-injected engines.

The goal of this article is to provide the reader with relevant and easy-to-use technical information (in the form of simple charts, basic equations and representative physical quantities) to form an informed opinion on available technologies and their purported capabilities and benefits along with potential pitfalls and physical limits. The focus is on GTCC startup, which can be considered as a*primus inter pares* among all GTCC transients. Admittedly, an article limited to a few thousand words cannot do justice to the subject matter at hand. The reader is encouraged to consult the listed references for a thorough understanding and guidance for applying the basic principles to his/her own projects.

There are many considerations in a successful GTCC start from standstill, which are discussed in detail elsewhere [1-3]. Correct steam chemistry, establishment of steam seals, vibration, overspeed and thrust controls are all vital for acceptable component life and RAM. When all said and done, however, the single most important issue from a fast start perspective is steam turbine (ST) thermal stress management. Furthermore, if the heat recovery steam generator (HRSG) is drum-type, high pressure (HP) drum thermal stress management becomes an integral part of the problem.

In a nutshell, GTCC startup optimization problem can be formulated as to minimize the time required to reach the dispatch power (e.g., full load or a specific part load) without "breaking anything" in the process - literally. The failure mode to avoid is crack initiation and propagation. Failure to control thermal stresses results in cracks via low/high cycle fatigue (LCF and HCF) and brittle fracture. In fact, LCF is found to account for roughly two thirds of ST rotor life with the remainder attributable mainly to creep. In particular, thick-walled components such as HP drum, ST valves, casings and rotor are exposed to LCF due to thermal cycling (start-stop sequence or load up-down ramps) and associated thermal stress-strain loop.

Definition of key material parameters and their typical values			1
Modulus of Elasticity	E	26,000	ksi
Linear Coefficient of Thermal Expansion	α	$6\text{-}7 \times 10^{-6}$	1/R
Poisson's Ratio	ν	0.30	
Thermal Conductivity	k	18.0	Btu/h-ft-F
Density	ρ	490	lb/cuft
Heat Capacity	c	0.125-0.175	Btu/lb-R
Thermal Diffusivity	δ	0.20-0.25	ft^2/h

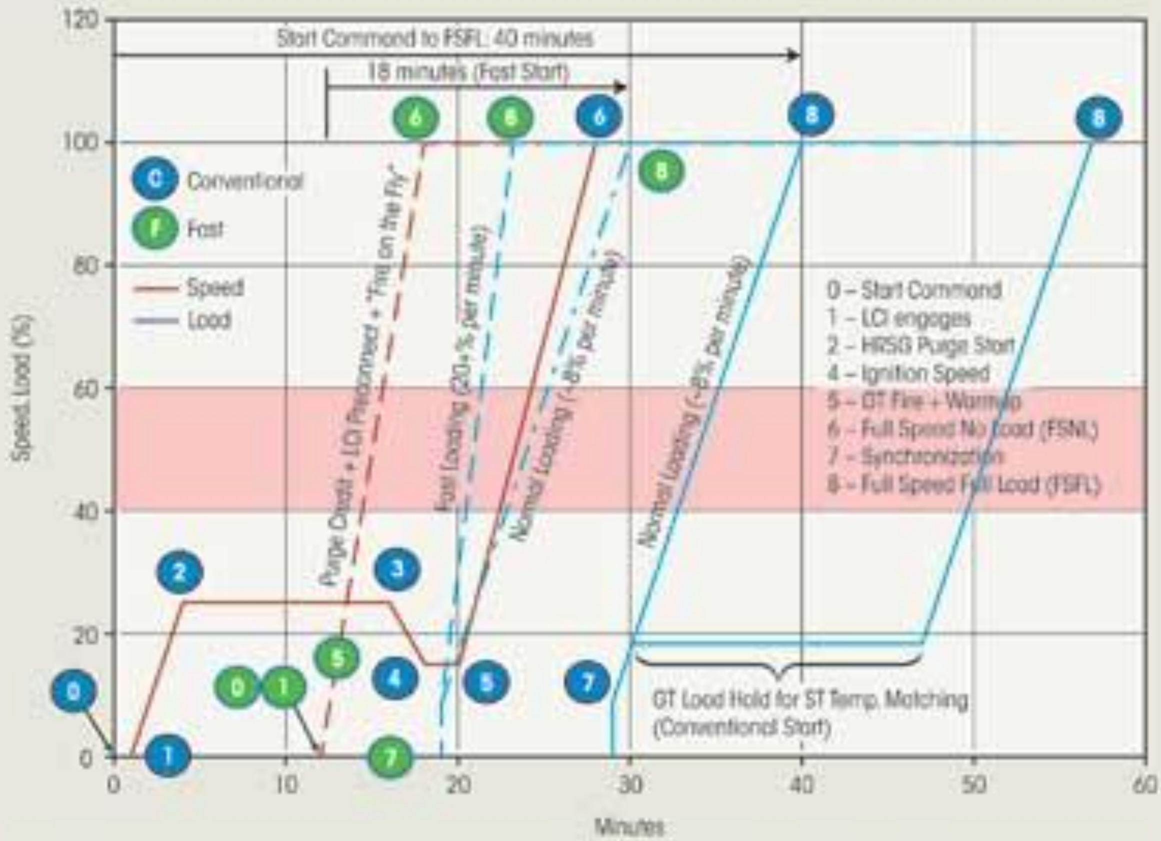
In principle, the solution is simple enough: thermal *decoupling* of GT and ST start processes. Thus, GT is started and rolled to full speed at no load (FSNL) at the maximum rate dictated by the size of static starter (Load Commutating Inverter, LCI), shaft torque limit, particular Dry Low NO_x (DLN) combustion system limits (e.g., availability of heated fuel gas, minimum fuel requirement by the lean blow-out margin, Wobbe index variation, etc.) among others. Following synchronization, GT is loaded as fast as possible first to its minimum emissions-compliant load (MECL) and then to its full load at full speed (FSFL).

GTCC start time definition hinges on when to start the chronometer. Unless specified unambiguously, one can never be sure when time t = 0 is and the difference can be significant. For a conventional start with HRSG purge and normal loading rate (i.e., no holds for HRSG warming) the difference between start command and ignition is 20 minutes (see Figure 1). Thus, the same start time (40 minutes to be exact) can be quoted as 20 minutes by someone who sets t = 0 at ignition. Today's fast start GTs with features like "purge credit", LCI pre-connect and "fire on the fly" can reach FSFL in 18 minutes or less from the start command (depending on the loading rate).

The rush to MECL is critical for reduction of startup emissions. The reason for that lies in the basic design philosophy of modern DLN combustors with fuel-air premixing, which are designed to run near the lean limit for low emissions. This is accomplished by piloted, multi-nozzle fuel injectors via sequential activation of fuel flow through individual nozzles (known as staging) to prevent lean blow-out and combustion dynamics while staying within the narrow equivalence ratio band to control NO_x and CO emissions. For older units MECL is 60%; for modern units the low load limit is around 50% (maybe 40% for most advanced systems). The exception to the rule is sequential combustion (reheat) GTs, which can turn off their second combustors to operate at 20% or lower load while emissions-compliant.

Two steps are instrumental in reducing GT start time: elimination of (i) HRSG purge sequence (by performing it right after shutdown in compliance with NFPA® 85) and (ii) hold time at low load with reduced exhaust energy (flow and temperature) to control HRSG steam production rate and steam temperatures (at the HP drum and HP superheater exit). Elimination of direct HRSG steam temperature control via GT load and exhaust energy is the "thermal decoupling", which is the key enabler of fast start. It can be accomplished via a bypass stack and modulated damper controlling the exhaust flow to the HRSG. A recently proposed technique is "air attemperation" of the GT exhaust gas flow via air injection into the transition duct. Ignoring the obvious but wasteful practice of "sky venting", the currently accepted method is a "cascaded" steam bypass system with terminal attemperators (TA). Steam generation and temperature-pressure ramp rates in HP drum are dictated by GT exhaust energy whereas final steam temperature control is accomplished by TAs. Until steam temperatures reach acceptable levels for admission into the ST, steam is bypassed via a route including the reheat superheater so that the latter is pressurized and "wet" (i.e., cooled by steam flow obviating the need for expensive alloys).

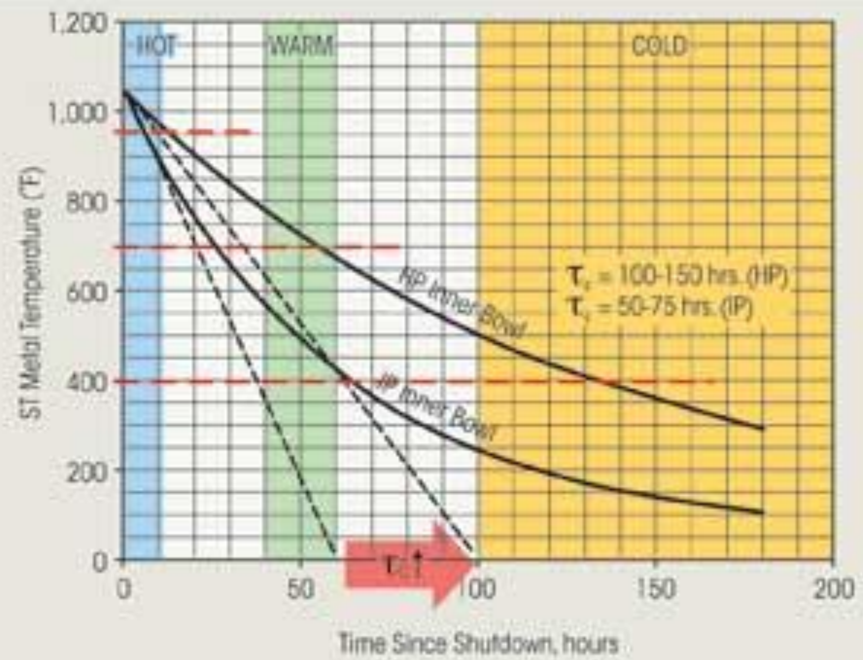
Gas Turbine Startup Diagram 1



Typical gas turbine startup diagram (conventional and fast versions). Conventional start with GT hold can take up to 50 minutes to reach MECL. Purge credit shaves off 12 minutes of that time. Eliminating GT hold saves another 15+ minutes. With fast start feature MECL point is passed in less than 10 minutes.

Steam FPT acceptable for admission into the ST is dictated by metal temperatures (primarily valves, casings or shells and the rotor). The critical component is the rotor, whose temperature cannot be measured directly and inferred by proxies (e.g., HP and IP inner bowl). ST metal temperature, Tm, is a direct function of unit downtime and ambient temperature as shown in Figure 2 (unless forced cooling is applied to start maintenance as soon as possible to minimize the downtime). The natural cooling time depicted in Figure 2 is represented by the exponential decay law.

Steam Turbine Cool-Down Profiles 2



Typical steam turbine cool-down profiles (as measured at HP and IP inner bowls). Shaded regions indicate typical time windows for "hot", "warm" and "cold" start classifications. Red dashed lines indicate average metal HP bowl temperature corresponding to the same.

$$\frac{T_m - T_{amb}}{T_{m,0} - T_{amb}} = e^{-\frac{t}{\tau_c}} \qquad \text{Eq. 1}$$

with a characteristic cooling time constant, τc, as a function of the ambient temperature, Tamb, and the starting value (denoted by subscript 0). This temperature is the main GTCC startup classification gauge instead of widely used but fuzzy terms such as "hot" or "warm", whose definitions vary from one source to another. Component Tm and, more precisely, its variation in a metal structure across a characteristic dimension, Lc, (e.g., diameter of ST rotor - 20-25 in. for modern GTCC units) along a characteristic dimension, x, is the key determinant of thermal stress via the following formula:

$$\sigma = E' \cdot \alpha \cdot \Delta T_m \qquad \text{Eq. 2}$$

where E' = E / (1-ν). For the ST rotor, ΔTm in Eq. 2 is the difference between rotor surface or bore and mean body (bulk) temperatures for surface and bore stresses, respectively. For a given steam temperature, T**stm**, bulk rotor body Tm varies according to the exponential decay law

$$\frac{T_m - T_{m,0}}{T_{stm} - T_{m,0}} = 1 - e^{-\frac{t}{\tau}} \qquad \text{Eq. 3}$$

with a characteristic time constant, τ, which is a function of rotor material (e.g., 1% CrMoV) and size cum geometry represented by Lc,

$$\tau = \frac{\rho \cdot L_c \cdot c}{h} \qquad \text{Eq. 4}$$

where h is the convective heat transfer coefficient (HTC) between steam and metal. Equations 1-4 tell the entire ST thermal stress management story in the concise language of mathematics. Thermal stress is determined by the temperature gradient in the rotor (essentially a cylinder) via Eq. 2; the latter is determined by the initial steam-metal ΔT (denominator of LHS of Eq. 3) with a time lag, which itself is dictated by HTC in Eq. 4. Everything hinges on the initial value of Tm, Tm,0, which is a function of the cooling period (Eq. 1). In physical terms, this translates into a mechanism to control steam FPT into the steam turbine at initial values sufficient (i) to roll the unit from turning gear (TG) speed to FSNL, (ii) to warm the ST rotor until steam-metal ΔT decreases to an acceptable level and (iii) to ramp them up at acceptable rates to their rated levels while ensuring that thermal stresses do not exceed prescribed limits.

Steam flow enters the picture via HTC in Eq. 4, which controls the rate of heat transfer between steam and the rotor surface as described by the heat flux balance at the steam-metal boundary (x = 0)

$$\dot{q} = h \cdot \left(T_{stm} - T_m\right) = k \cdot \left.\frac{dT_m}{dx}\right|_{x=0} \qquad \text{Eq. 5}$$

This equation introduces the dimensionless Biot number, Bi = h·Lc/k, which is a relative measure of the uniformity of temperature gradients inside a heated or cooled body. Determination of HTC is one of the most uncertainty-prone undertakings in transienTheat transfer problem in a complex geometry such as steam path flow. Its dependence on steam flow is based on the well-known Nusselt number correlation for heat transfer in internal flows, i.e., h ∝ . The heat transferred from steam to the rotor at the surface increases the rotor's bulk temperature according to Fourier's law

$$\frac{dT_m}{dx} = \frac{k}{\rho \cdot c} \cdot \frac{d^2T_m}{dx^2} \qquad \text{Eq. 6}$$

Equation 6 introduces the thermal diffusivity, δ = k/ρc, which quantifies the speed with which the temperature of a heated or cooled body changes. Typical values for the key parameters governing ST rotor thermal transients are given in Table 2.

Representative values of major parameters characterizing the transient heat transfer during steam turbine warm-up for typical steam flow, pressure and temperatures. 2

m/m_o	P	T	h	Bi	δ	τ
[-]	psia	F	Btu/h-ft²-F	[-]	ft²/h	min
1.0	120	700	116	7	0.26	37
1.0	120	1,050	100	6	0.21	54
1.0	1,200	700	958	56		5
1.0	1,200	1,050	701	41		8
0.2	120	700	32	2		135
0.2	120	1,050	28	2		196
0.2	1,200	700	264	15		16
0.2	1,200	1,050	193	11		28

For ferritic steels used in modern GTCC units, k and ρ do not show significant variation. Thus, δ is primarily a function of temperature and changes by about 25% between 700 and 1,050°F; i.e., rate of change of metal temperature is 25% faster at the higher temperature. The data in Table 2 can be summarized as follows: higher steam flow and/or pressure result in higher rates of heat transfer between steam and metal, which is quantified by higher Biot numbers and shorter time constants (i.e., faster heating or cooling). In conjunction with the data in Table 2, Eqs. 5 and 6 identify the two distinct phases in ST start with thermal stress control:

> (i) low flow and high steam-metal ΔT with low HTC until temperature gradients settle down (non-stationary phase or Phase I) and
> (ii) increasing steam FPT to load the unit with high HTC and nearly constant, low steam-metal ΔT (quasi-stationary phase or Phase II).

Equation 5 describes Phase I via its simplified solution for a cylindrical geometry given by [4]

$$\sigma_{max} = E' \cdot \alpha \cdot \left[\frac{Bi}{2.8+Bi+\sqrt{Bi}}\right] \cdot K_T \cdot \Delta T \quad \text{Eq. 7}$$

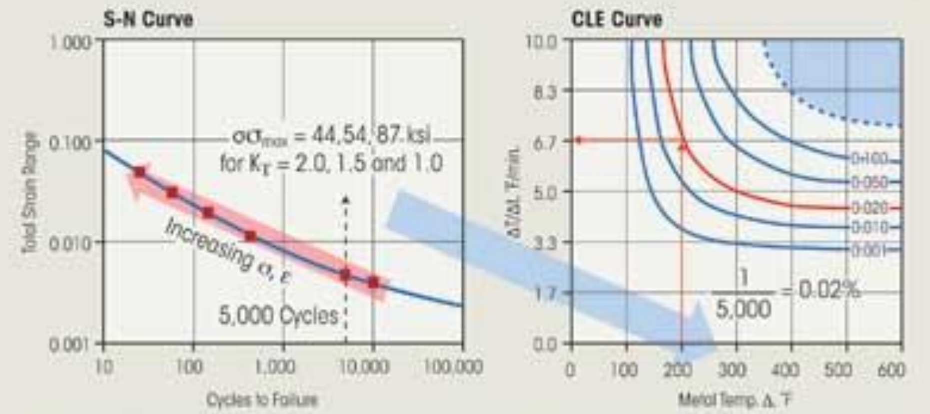
which gives the maximum thermal stress implied by a given step rise in T**stm** at time t = 0 (with a time lag characterized by the Biot number). Note that the base stress formula of Eq. 2 is amplified by a stress concentration factor KT, which accounts for the presence of geometric discontinuities on the rotor (which is not a perfect cylinder after all). Similarly, Eq. 6 describes Phase II via its simplified form given by

$$\frac{dT_{stm}}{dt} = \frac{\delta}{\phi_F \cdot E' \cdot \alpha \cdot L_c^2} \cdot \sigma_{max} \qquad \text{Eq. 8}$$

where ØF is the form factor (0.125 for a cylinder [4]). Equation 8 gives the allowable T_{stm} ramp rate for a given maximum allowable stress, σ_{max}, which is dependent on rotor material and typically lies in a range of 50-80 ksi. For the cited range, with the data in Table 2, Eq. 7 suggests that for low HTC (~100 Btu/h-ft^2-F or less) steam-metal ΔT can range from 200-300°F (high KT) to 500°F and higher (low KT). For high HTC (~650 Btu/h-ft^2-F), steam-metal ΔT can range from 100-200°F (high KT) to about 400°F (low KT). Similarly, using Eq. 8 with Table 2, it can be seen that allowable values for dT**stm**/dt range from 3-6°F to 8-10°F.

The allowable stress is not a precisely defined material property. (For ferritic steels used in ST rotor construction, 0.2% tensile yield strength lies between 70-90 ksi for temperatures 600-1,000°F.) It is derived from the S-N curves relating total strain to cycles to failure, which gives the fatigue life of the material in question (for LCF life of CrMoV alloy see Figure 3). Based on the relationship between stress and strain, ε, via the modulus of elasticity, σ = E´ • ε, this curve is used to determine σmax for a defined fatigue life. In practice, the relationship between σ and ΔT allows the translation of the S-N curve into *Cyclic Life Ependiture* (CLE) curves, which determine the allowable T_{stm} ramp rates (Figure 3). Depending on the rotor material, size and geometry and its temperature at start initiation, the range is limited to about 5 to 10°F per minute except for very hot "restarts" after a few hours of downtime.

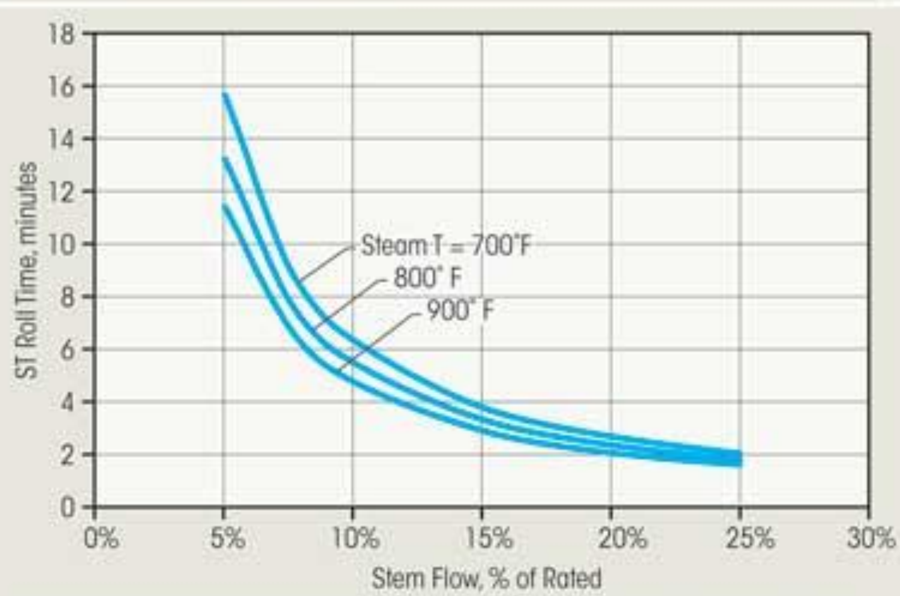
S-N and CLE Curves 3



Typical S-N and CLE curves for ST rotor LCF (CrMoV) [5]. Metal ΔT in CLE chart represents the total temperature change between initial and final states (beyond 600°F, curves are flat).

Steam turbines with cascaded steam bypass are typically started by admitting steam from the reheat superheater into the IP section. Admission steam FPT should be sufficient to overcome the rotational inertia (in lb-ft^2) of the entire ST and its generator, Irot, and accelerate it from TG speed (a few rpm) to FSNL (3,000 or 3,600 rpm). Based on available steam FPT and initial IP rotor temperature, using the relationship between ST power generation (expansion from IP inlet to the condenser), rotor torque and rate of change in angular speed, ω, the roll time can be estimated as 2 to 15 minutes (see Figure 4) via

Steam Turbine Roll Times 4

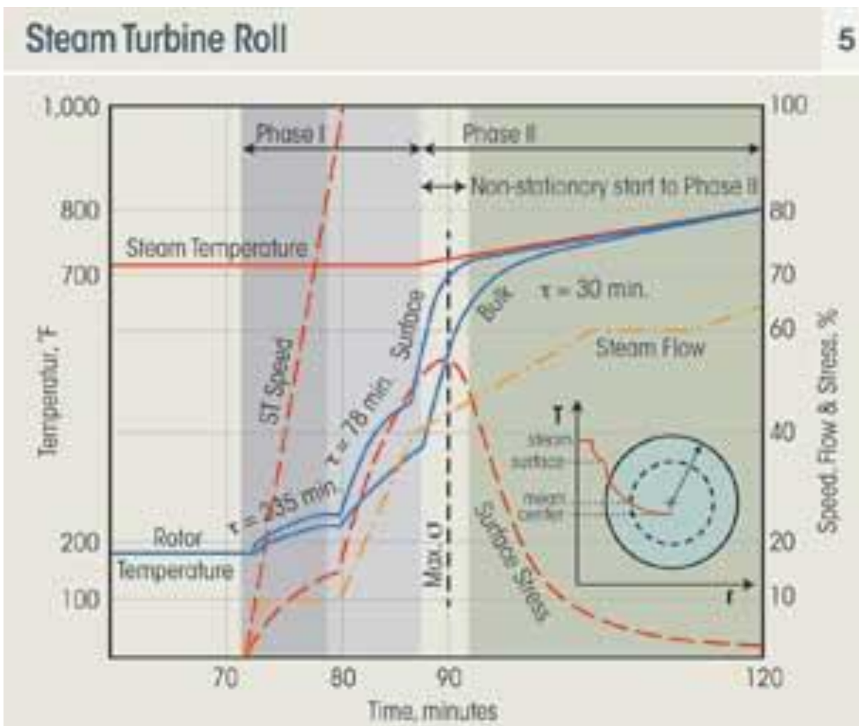


Steam turbine roll times for varying steam flows and temperatures. (Steam pressure 120 psia, rotational inertia 700 kp-ft2, rated IP turbine inlet flow 681.5 kpph.)

$$N(t) \approx \frac{3000}{\pi}\sqrt{\frac{5}{I_{rot}} \cdot \int_0^t \dot{m}_{stm} \cdot \eta \cdot \Delta h_{isen} \cdot dt} \qquad \text{Eq. 9}$$

where N is the rotor speed (rpm) and the argument of the integral on the RHS of Eq. 9 is the power (in Btu/s) generated by steam expanding between IP turbine inlet and condenser [6].

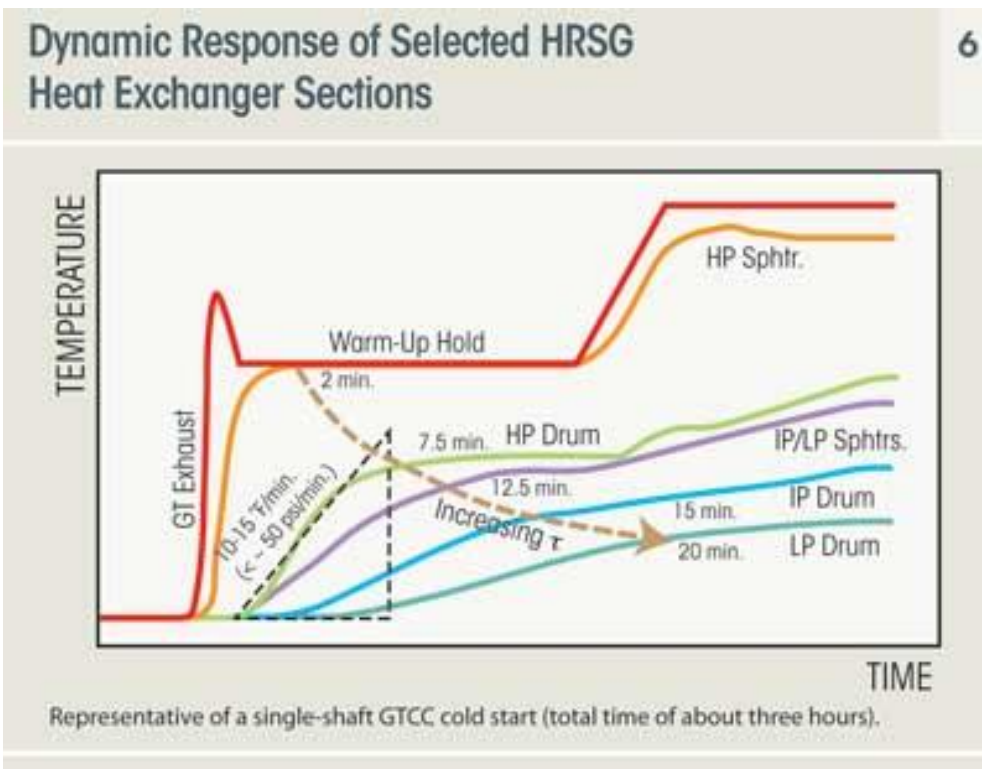
The chart in Figure 5 shows the first two hours of ST roll, warm-up and loading phases for an initial Tm of 180°F (about 5-6 days of downtime per Figure 2). Steam is admitted into the IP turbine at 715°F and 120 psia at a flow rate of 10% of its rated value at full load. This is sufficient for acceleration from TG to synchronization in 8 minutes (see Figure 4). Initial steam-metal ΔT is 500+°F but this is acceptable due to the low HTC (less than 30 Btu/h-ft^2-F per Table 2) and the ensuing low σmax from Eq. 7 (also very high τ > 200 minutes). Following synchronization, IP steam flow is ramped steadily to 40% to accelerate the warm-up process via increased HTC. Once the steam-metal ΔT (based on rotor surface temperature inferred via IP inner bowl thermocouple) reaches about 250°F, T**stm** is ramped (via TA control) at a rate defined by the CLE curve (about 3 to 4°F per minute for an acceptable life of 4 to 5,000 cycles from Figure 3).



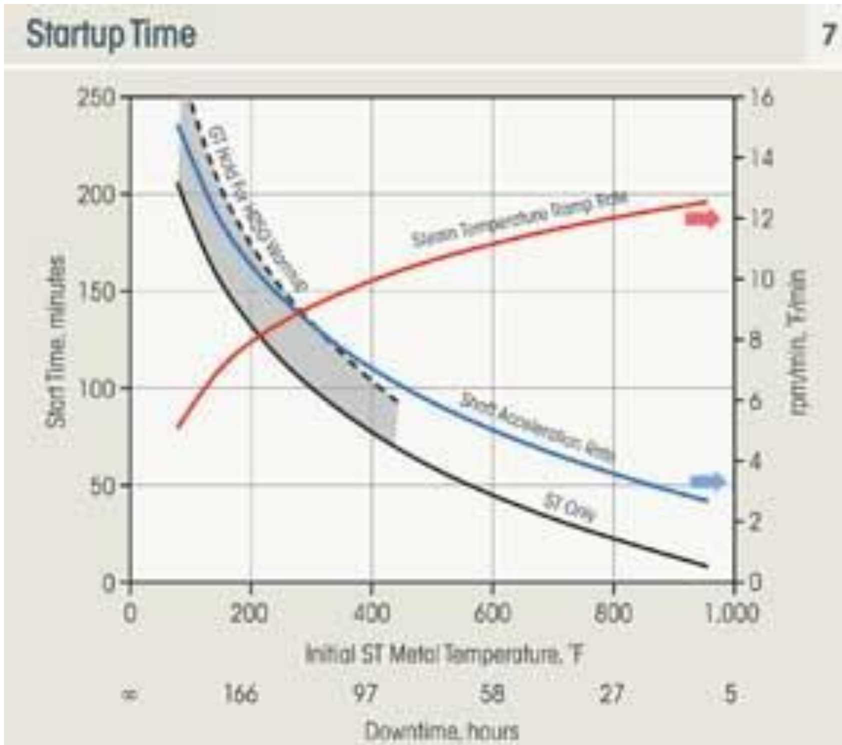
Typical ST roll via IP steam admission and the ensuing warm-up period [6]. Representative of a single-shaft GTCC cold start (total three hours). Note how the quasi-stationary Phase II is preceded by a short non-stationary period.

The other component subject to LCF damage due to cycling is the cylindrical HP drum of the HRSG (4-5 inches wall thickness). The limiting thermal stress is at the inner drum wall controlled by saturated steam p-T inside the drum. During startup, mechanical stress due to internal drum pressure and thermal stress due to thermal expansion are in opposite directions, while they are in the same direction during shutdown. Unlike the ST, which is thermally decoupled from the GT via TAs, HRSG sections are directly "under fire". They respond to GT exhaust temperature transients much faster than the ST rotor in direct proportion to their distance from the inlet (see Figure 6). Thermal stress calculations and material properties similar to those described above limit the p-T ramp rate inside the drum to 10-15 °F/min (about 50 psi/min max.) for units designed up to ~1,800 psig at ST throttle (~6-10% higher at the HP drum). Advanced steam cycles with 2,400 psig throttle and drum-type HRSGs (very thick walls) would push down the ramp rate to a few degrees per minute (see Eq. 8 for the relationship between dT_{stm}/dt and Lc). This can be alleviated to a certain degree by using stronger alloy steel (obviously more expensive) and/or designing the HRSG per EN-12952 rather than the ASME code, which results in thinner walls. One obvious solution is once-through design of the HP evaporator, which eliminates the thick-walled drum altogether but has its own drawbacks and caveats. A recent design approach proposes to replace the HP drum by a cylindrical, thin-walled knock-out vessel with external separator bottles and thus avoid the thermal stress problem in cold starts. According to HRSG OEMs, cold starts (T_{drum} < ~400°F) are 20 times more damaging than warm starts (T_{drum} < ~500°F) whereas hot starts (T_{drum} > 500°F) do not impact LCF life. In "hot" starts, HP and reheat superheaters subjected to very steep gas temperature ramps are critical in terms of HRSG life consumption. In this context, one should add that the desirability of purge credit is due to more than startup time reduction. It prevents excessive quenching of superheaters, which act as "supercoolers" during hot starts when subjected to relatively cold GT exhaust with detrimental impact on their fatigue life.

Natural p-T decay of the HP drum can be described by Eq. 1 with τ_c of 60 to 80 hours. It takes about 2-3 days for the pressure to decay to the atmospheric conditions. Bottling up the HRSG via stack dampers with insulation up to the damper, steam sparging (requires auxiliary boiler) or running the SCR ammonia vaporizer heaters help keep the HRSG warm and pressurized over limited duration shutdowns to enable GT starts with no low-load hold. Beyond about three days, however, this is increasingly impractical and even in plants designed for fast starts limited duration GT holds are needed to accomplish HP drum warm-up in two steps (somewhat similar to that shown in Figure 6).



Combining the elements discussed above and illustrated by the ST roll example in Figure 5, a representative ST start curve can be established as a function of the key controlling parameter, namely, ST metal temperature at the startup initiation (Figure 7). Appropriate GT start time per Figure 1 (from start command to the point when ST roll begins) should be added to that for total GTCC start time (e.g., 18 minutes for the fast start). The four-minute mile of fast start capability is roughly 30 minutes from a standstill (to be defined precisely) to combined cycle full load for a "hot" start (e.g., following an overnight shutdown). This is generally compared to a conventional hot start, which takes around one hour (see Figure 1). The underlying physics discussed herein briefly and summarized in Figure 7 hopefully makes it clear that this particular case is only one single point in a continuum of start scenarios driven mainly by the downtime preceding the pushing of the start button.



Startup time of a typical steam turbine in a modern GTCC with drum-type HRSG (from the start of ST roll to the point when all bypass valves are closed and all admission valves are fully open).

References

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