TECHNICAL PUBLICATION

Constant and Sliding Pressure Options for New Supercritical Plants

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ABSTRACT

The abundant domestic energy resource of coal will certainly be used for both replacement and additional power generation in the USA. As a primary component of on-going efforts to reduce the environmental impact from coal combustion, the USA industry is renewing its interest in new, high-efficiency plant designs. Supercritical steam conditions (greater than 3208 psia pressure) are employed to achieve plant efficiencies of 40% (HHV) and beyond. Operation at such pressures — without distinct phase differences between liquid and vapor — requires certain steam generator design features, most notably in the furnace circuitry and components.

Within this category of steam generators, the design is also very much influenced by the intended pressure operating mode. This paper takes a close look at the potential costly differences between sliding pressure and constant pressure steam generator designs. Beyond the obvious differences in component and constructional design features, we consider broader implications including overall furnace sizing differences and materials options. The related costs represent additional hurdles for sliding pressure to justify over time via any operational advantages. Part-load operating efficiency — previously considered to be an advantage of sliding pressure — is evaluated in light of modern designs and the difference in operating costs between the modes is shown to be minimal over a wide range of loads. Further, the value of rapid and sustained load-following capability — perhaps the primary remaining advantage of sliding pressure — will be dependent on regional grid realities and particular owners' needs. These conditions are considered to vary widely and are hard to predict in definite terms, so it is proposed that some combination of both sliding pressure and constant pressure supercritical designs could be appropriate in many owners' generating fleets for the future. Plant developers should recognize the steam generator design implications in their strategic planning and specification of new plants to arrive at the most effective portfolio of generating capability for the USA market environment.

Plant Development Trends and Perspective

Recent moves toward deregulation have led to a competitive need to reduce operating costs. Although coal remains relatively inexpensive in the USA, fuel still represents the primary operating cost of coal-fired utility plants. Further, fuel reduction through efficiency improvement has a direct impact on air and solids emissions, which increasingly represent operating costs through regulatory measures and emission trading credit values. One of the emission species currently of public concern and discussion is carbon dioxide (CO_2)[1] , which could be assigned a significant operating value in the future and further drive the moves toward higher efficiency. CO_2 emissions credits are already being traded in the European market to promote better use of resources and slow the volatization of carbon into the atmosphere, and similar practice is anticipated in the USA.

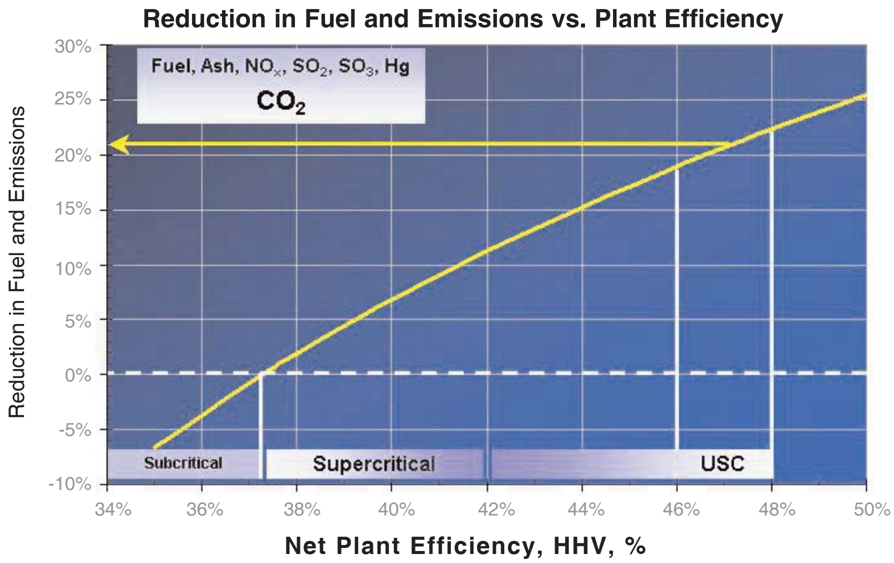


Figure 1. Reduction in Fuel and Emissions vs. Plant Efficiency

Figure 1 illustrates the reciprocal relationship between advances in efficiency and reductions in fuel consumption and emissions on a per-megawatt (MW) basis. The steam cycle conditions of a conventional, fossil-fired steam plant have a primary influence on the potential plant efficiency. The existing base of subcritical units in the USA is fairly old — primarily of 1950's-1980's vintage — and has a typical efficiency of less than 35% (HHV basis). Meanwhile, a modern, state-of-the-art subcritical plant could have an efficiency of 36-37.5% and serves as the baseline reference for consideration of advanced technologies in this chart. The existing supercritical plants in the USA are also quite old with modest steam conditions and efficiency in the 36-38% range, while new plants today can have an efficiency of 38-42%, leading to fuel savings of 2-11% relative to a modern subcritical plant. While the economic pressures driving further advances in efficiency will be gradual, there are currently worldwide efforts to develop material options to achieve UltraSuperCritical (USC) goal conditions (5000 psig, 1400/1400°F steam), which would yield an efficiency of 46-48% and significant reduction (20-22%) of all fuel and emissions rates on a per MW basis.

As the above statement implies, the advancement of steam conditions is not made out of a sense of adventure or of national pride (generally), but is rather a result of the particular economic realities in each region of the world. Up until now, the advancements in plant efficiency overseas have been driven primarily by high and rising fuel cost. Figure 2 indicates that after the establishment of typical 3700 psig / 1000/1000°F supercritical steam conditions, Japan was consistently using 1050°F throughout the 1980's, and Europe followed suit in the 1990's. Both regions are now well established with 1100°F steam, with some recent experience at 1150°F, all in the interest of reducing fuel consumption and waste heat. The experience in these two regions differs, however, when steam pressure is considered, as in Figure 3. In general, some matching of pressure and temperature is anticipated as steam conditions are elevated, due to limitations of moisture and steam quality at the turbine last stage. Despite using the same 1100°F steam temperature for advanced plants, Europe — especially Germany and Denmark — tends toward high pressure (3800-4300 psig) and Japan favors uniquely low pressure (3500 psig). The European plants at the very high pressure levels include both single and double reheat units.

As indicated in the two trend charts, however, most near-term supercritical developments planned in the USA are for modest advances in steam temperature (1050°F typical) with no significant deviation from the traditional 3750 psig pressure level. This reminds us that the pace and schedule of the economic drivers are different in each market. Further, the limitations to advanced steam conditions are set by the pressure parts materials available for use and approved by the governing Codes in each region. As a result of these materials and Code limitations, and with an eye to the future, research is being conducted worldwide to develop, demonstrate, and approve advanced materials to permit further progression of steam conditions (Figure 4).

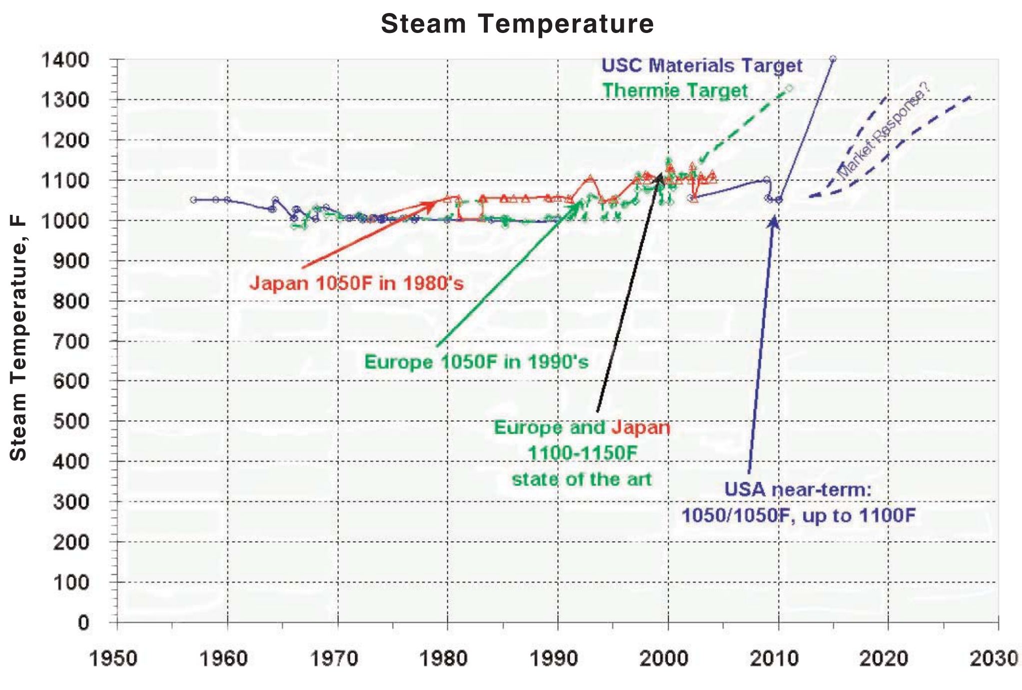


Figure 2. Historical Perspective: Steam Temperature of Advanced Plants (Reheat shown)

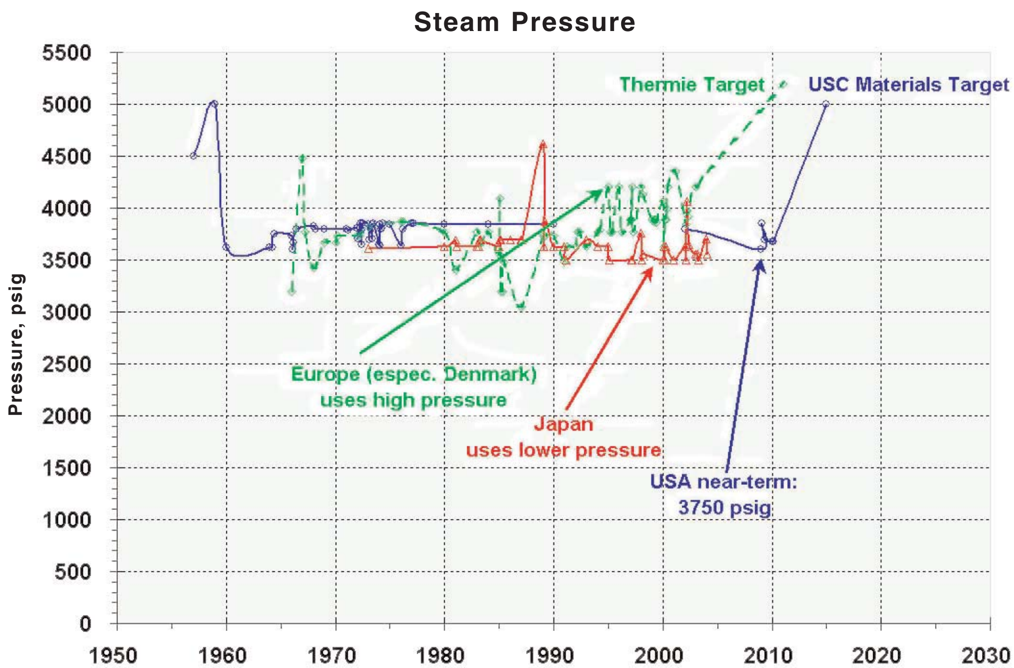


Figure 3. Historical Perspective: Steam Pressure of Advanced Plants

The Thermie / AD700 program in Europe, and the USC Materials Consortium in USA, have established target dates to provide materials options to permit design steam temperature of 1330°F and 1400°F, respectively, at elevated pressure (Figures 2 and 3). The indicated target time frames reflect when such materials are expected to be approved and available for use, but — as indicated above — the schedule of advancement in actual plant design steam conditions and the use of these new materials are expected to grow in response to local market pressures. While the fuel cost driver in Japan and Europe has been and will be steadily building from "natural" economic forces, the "man-made" driver of emissions regulatory changes in the USA (and now the rest of the world with Kyoto protocol) can be more sporadic, and indeed be difficult to predict or anticipate over the long term. Uncertainty stemming from future regulatory implications and New Source Review practices represents a significant hurdle for the planning of and investment in new large coal-fired plants and related infrastructure development.

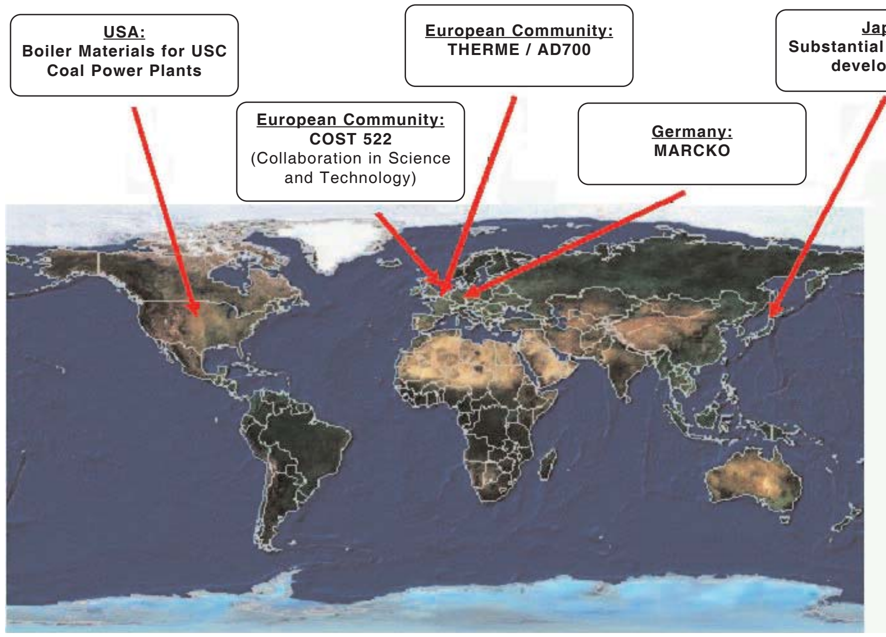


Figure 4. Worldwide Materials Research and Development

Cycle Operating Modes Should Not Be Assumed From Other Markets

Most of these research programs include a conceptual design task to develop one or more steam generator design options to produce the target steam conditions. From such conceptual designs, the operating conditions of the various system components are roughly predicted and so the materials testing conditions for the various alloys are established. As the starting model for these concepts, all program efforts have assumed the market needs and operating modes that have driven the recently built plants in Europe and Japan. In such markets, even new, highly efficient, coal-fired plants are regularly shut down, started up, and load-cycled to a wide extent. To handle rapid and continual load ramping, the turbine temperature transients are minimized by operating under sliding pressure mode. This requires certain and drastic adaptations of the steam generator design which — for current steam conditions — are well worth the investment under the market realities of these regions (except that the implied low capacity factor means a longer payback period for the higher capital investment).

However, as steam conditions are increased toward the USC goal, and if sliding pressure mode is assumed, a major challenge is the significant rise in furnace outlet temperature and the potential need to fabricate the furnace out of exotic materials. The USA USC Materials program is trying to make system and pressure concessions in order to limit the furnace material to T23 or T92, while the Marcko program in Germany was considering that the furnace may even need to be made of nickel-based alloys — an expensive consequence. As we will see later, these concerns and costs are much reduced for constant pressure applications and so utilities are encouraged to carefully evaluate their future needs. In Riley's developments for design options, we find that constant pressure can alleviate not only this materials selection problem, but can result in a smaller furnace enclosure and considerable capital savings. This is in addition to the obvious design costs for rapid load cycling and once-through design including special limits on component thickness, design, and material, as well as operational penalties at full load such as greater pressure drop in once-through units.

Potential Need for Options

We suggest that it is incorrect to assume that the USA market needs a single and certain plant design optimized for other regional market realities. As a parallel, we take a look at the current plans for plant development in the USA (Figure 5). Despite the previously mentioned pressures for high efficiency, we see that both subcritical and supercritical plants are being planned — not just advanced-efficiency plants. Just like worldwide markets differ, even USA regional markets and the needs of individual utilities and independent power producers (IPP) vary and call for different solutions and diversity of assets. Similarly, even within the supercritical category, there are two distinct options for operating modes and designs and, as with any investment, up-front decisions must be made regarding the objectives and appropriate design of each plant to be built. Whereas the various USC materials development efforts focused on sliding pressure application only, Riley Power Inc. (a Babcock Power Inc. company) suggests and is developing options for utilities with variations in dispatch competition, fuels, and other needs, both near- and long-term.

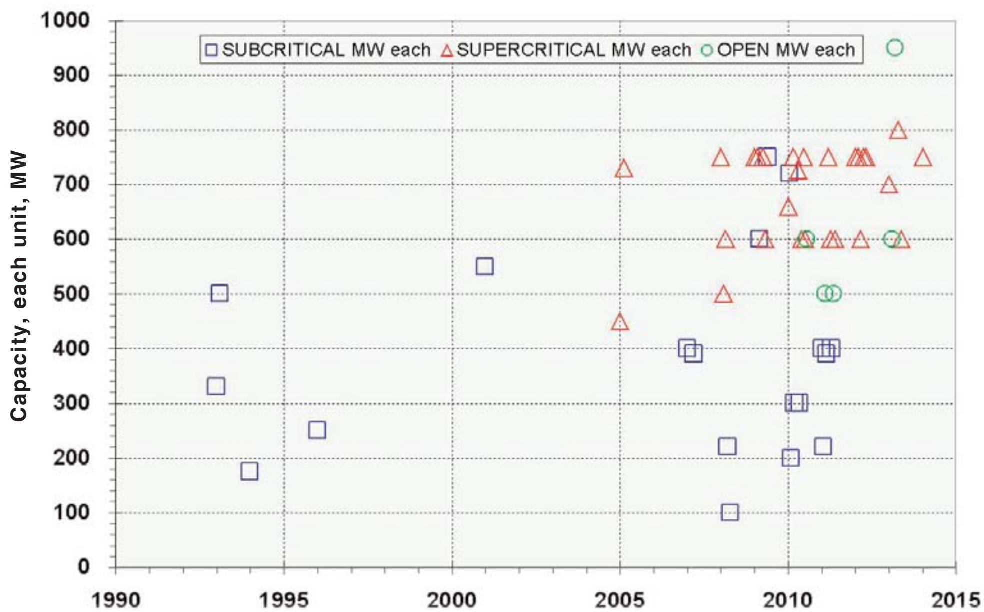


Figure 5. Announced Near-term Plant Developments

Each region is comprised of a certain distribution of generating technologies and fuels. With load dispatch (the potential for power sales / income) for each unit being granted on a basis mainly related to its specific cost-of-electricity, new units in each region are faced with a unique competitive environment, and could be base-loaded or cycled to various degrees or schedules. Some plants are inherently base-loaded — such as hydro and nuclear — while others with high fuel costs (natural gas and oil) are relegated to peaking duty (Figure 6).

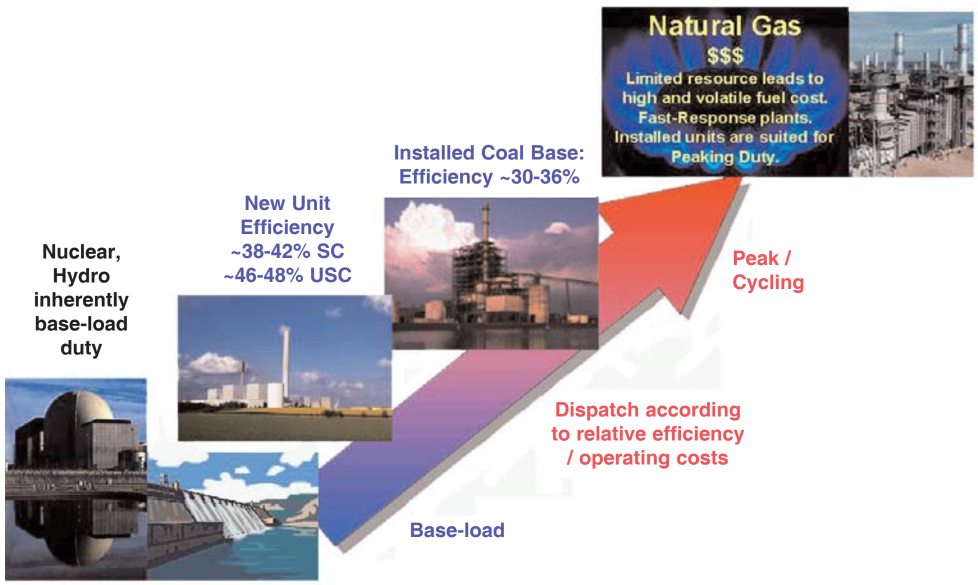


Figure 6. Technologies and Fuels in Each Regional Market

The need for capacity addition and replacement had been growing for some time, but with national energy and environmental policies in a state of flux, large plant developments were delayed. When the need for capacity became critical, natural gas combined cycle plants provided a feasible option due to their rapid deployment, reasonable fuel cost, and relatively low emissions. In the meantime, natural gas prices soared due to the unprecedented growth in demand, deregulation led to competitive load dispatch on individual unit operating costs and thereby de-coupled electricity rates from any one utility's operating costs, and economic growth had slowed. The result is that much of the new capacity in the form of natural gas combined cycle — despite its relatively high thermal efficiency — is in the back of the line for load dispatch due to its extremely low economic efficiency. Many of the units that were actually completed are now sitting unused or are used for only peak demand periods if cheaper coal-fired power can not be transferred from a neighboring grid region. This installed peaking capacity, as well as the overall reduced generating reserve, produce a different environment than that during the 1970's.

As explained for Figure 1, new supercritical coal plants have a healthy efficiency advantage over all of the existing USA coal fleet, and therefore should be fully dispatched before any other coal plant. Figure 7 illustrates an example regional plant dispatch curve and suggests that high efficiency coal plants will certainly be favored for base-loading in most regions of the USA. However, many of the existing generating units in the USA are of 1950-1960's vintage and must eventually be retired, though retirement schedules are continually being extended. Also, we recognize that there is the possibility of a resurgence in nuclear plants that would receive preference for base-loading. Over time, as a subject unit competes against other high-efficiency plants for load dispatch, it may move from the group of units required for “minimum grid load” to among those required for “average grid load”. On the other hand, some regions have significant and unused natural gas combined-cycle capacity as an available asset for peaking that will not be retired soon. So, we can predict that new coal plants will initially be fully loaded in most regions, and in some regions may eventually and progressively be relegated to 2-shift operation if and when other higher-efficiency units can better provide the consistent minimum load. In temperate climates such as for the USA, grid load reductions are first felt on a seasonal basis (Spring and Fall), then on weekends, and then nightly. Even upon nightly load reduction, it is most common to reduce load on the units required for average load rather than to repeatedly shutdown and startup these units. We suggest that it is very unlikely that the new generation of USA supercritical plants — or at least most of them — would be expected to peak (with frequent shutdowns) or continuously and widely load-cycle. If these conditions are not required for most of the new supercritical plants, there is significant capital savings to be had in specifying and designing plants for the more likely operating conditions, even if a significant amount of time is spent at reduced load. It is not necessary to count on full- or base-loading in order to justify these plant options since, as will be explained later, certain constant pressure plants achieve nearly the same heat rate as sliding pressure plants across the entire load range.

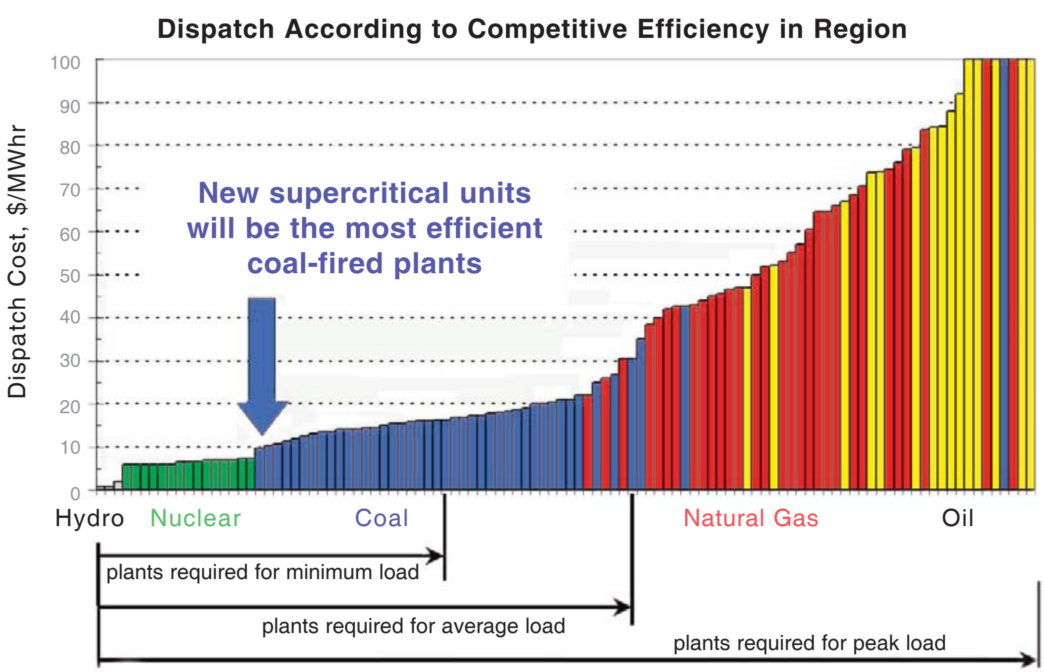


Figure 7. Example Regional Dispatch Curve

Constant and Sliding Pressure

Steam Generator Design Differences

Basic Operation and Constructional Differences

Considering the above points regarding USA market needs and comparative operating efficiency, we next identify significant design and capital cost differences which suggest that constant pressure should be considered for some, if not most, of the new USA supercritical plants. Some of the differences between sliding and constant pressure steam generators are widely known, and others are not. These lesser known differences can have a major impact on cost, so we will introduce them after briefly reviewing the basic constructional differences.

As a prelude to understand the reasons for the primary design differences, the basic operating pressure trends are indicated in Figure 8. Generally, constant pressure refers to the stable pressure of the steam generator and main steam line over the load range. The basic nature of a fixed, rotating turbine is to require less pressure as flow rate is reduced, and if the main steam pressure is limited to only that required for each load, this mode is referred to as pure sliding pressure. However, when we generally speak of "sliding pressure", we often mean modified sliding pressure as indicated, which has a certain amount of pressure throttling to provide a modest amount of fast response load reserve. A unit under constant pressure will have significant load reserve at any reduced load. (By opening the throttle valve or an admission valve, the pressure in the turbine and steam generator move toward equalization. The sudden reduction of pressure in the steam generator prompts an instantaneous expulsion of steam mass due to the increase in specific volume within the confines of the system, and provides a temporary load increase even before the fuel handling and firing system can be loaded to support any sustained higher load. Pure sliding pressure operation does not offer this kind of load response and is therefore generally not practiced.)

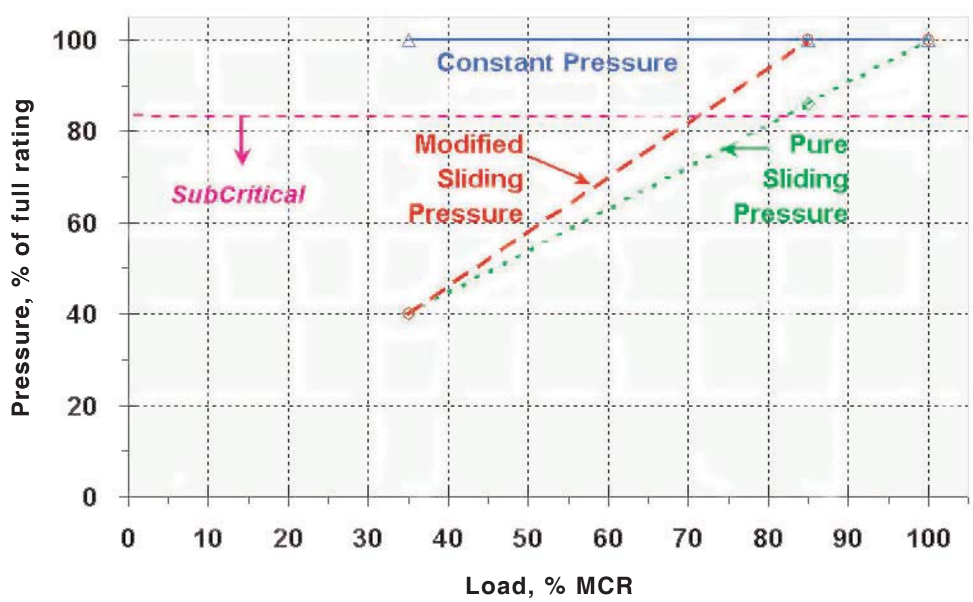


Figure 8. Pressure Vs. Load: Various Operating Modes

Sliding Pressure

Under sliding pressure operation, because the steam generator operates under both supercritical and subcritical conditions, the furnace must be designed according to both single- and 2-phase fluid challenges. The two pressure regimes and the wide variation in fluid specific volume make continual forced recirculation rather impractical and so a once-through design is appropriate, where flow rate through the furnace is directly proportional to load. Steam flow rate and velocity through the furnace tubes are critical for cooling the tubes and so — with flow proportional to load — low load represents a challenge for proper furnace tube cooling. Further, under sliding pressure, it is at low load that the fluid is subcritical or 2-phase and this condition poses specific challenges to heat transfer and tube cooling. Both departure from nucleate boiling (DNB) and steam dry-out carry the potential for elevated tube metal temperatures, and these conditions are mitigated or avoided, in part, by providing sufficient steam mass flow density at the subcritical, once-through, low loads. Designing for proper steam cooling effect at low loads produces very high steam mass flow density and pressure drop at full load in a once-through design, and so the specification of minimum once-through load should be done with careful consideration of the consequences at full load. Below the minimum design once-through flow rate, recirculation pumps are generally used to protect the furnace. In stark contrast, these 2-phase problems can be avoided entirely in a supercritical furnace maintained at constant pressure.

High steam mass flow density (lb/hr ft2) with low flow rates (lb/hr) at low load is provided in a once-through unit by use of a small flow area. Because the furnace perimeter has certain minimum limitations due to conventional firing configurations and slag control, the challenge of providing a small flow area to envelope a relatively large furnace enclosure requires special plumbing arrangements. But because sliding pressure operation involves two-phase fluid over most of the load range, multiple furnace passes with up-down-up flow direction become difficult to manage and a single upward flow progression is preferred.

The upward flow progression in a single pass is made with fewer tubes by laying the wall tubes down at a low inclination angle rather than hanging the tubes vertically. As shown in Figure 9, a given transverse dimension of a furnace wall normally covered by 9 vertical tubes and membrane fins can be covered by only 3 inclined tubes of the same tube and membrane size. However, to cover the same vertical dimension, the inclined tubes are much longer as they normally span each of the four walls at least once. Although the furnace cross-section remains rectangular, this inclined tube arrangement is often called a "spiral" design due to the overall progression of each tube upward and around the furnace. The tube inclination angle is typically 10-20 degrees from horizontal, and so the tube length is 3-5 times greater than the vertical distance gained.

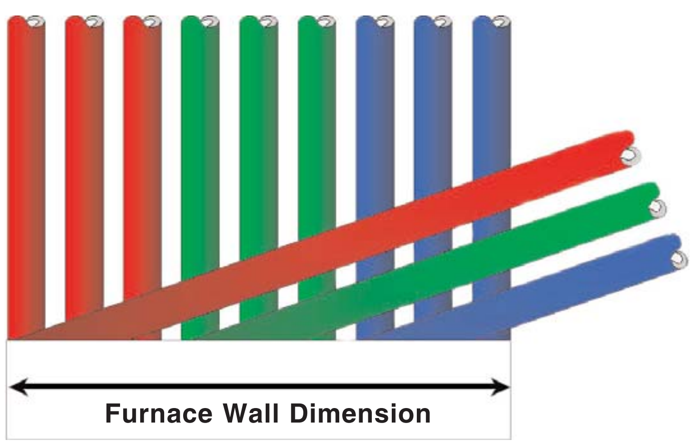


Figure 9. Reduction of Flow Area Via Inclined Tubing

Special internally-rifled tubing could permit acceptance of lower steam mass flow density and the use of vertical tubes, but the range of operating conditions under sliding pressure operation make such systems quite challenging with uncertain success.

Figure 10 includes an example of a sliding pressure design for PRB, with spiral arrangement in the high heat flux zone of the lower furnace. Although much experience and lessons learned have been gained with such a furnace wall design, it remains a complicated structure to design, fabricate, erect, and maintain. The expensive arrangement is terminated at a certain elevation in a lower heat flux zone, and a transition is made to vertical tubes in the upper furnace. The transition is commonly accomplished by a "ring" of forgings around the perimeter of the furnace, and an external "ring" mixing header. The walls comprised of inclined tubes are not self-supporting, and so an "exo-skeleton" support system is used, consisting of vertical support straps and load transfer by many welded lugs over the wall surfaces.

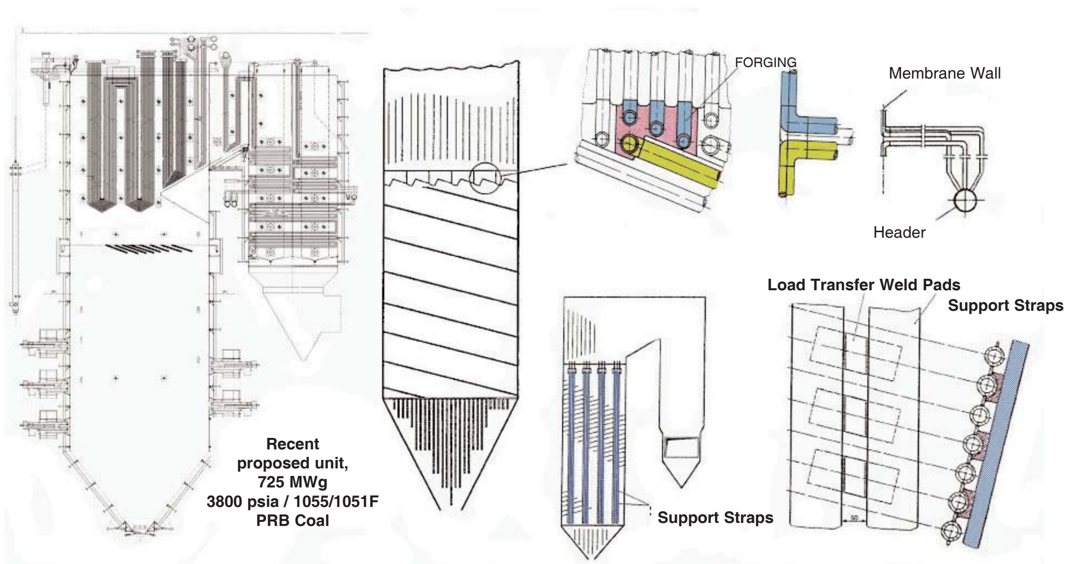


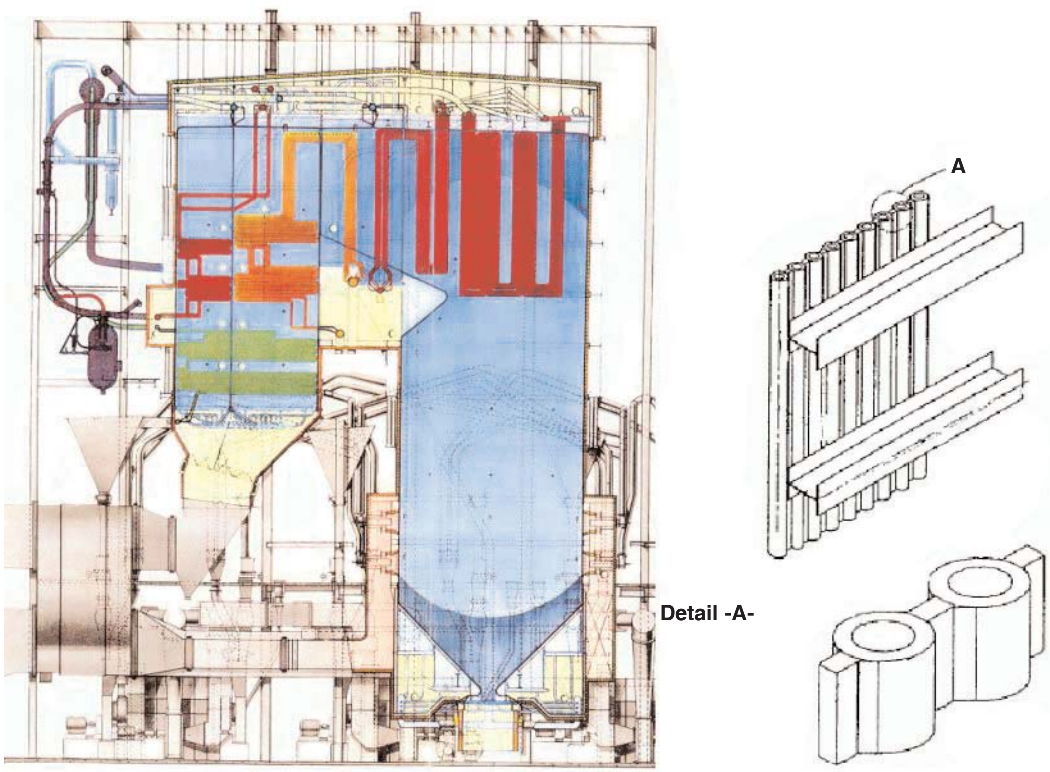
Figure 10. Sliding Pressure Once-Through Furnace Construction

Constant Pressure

The 2-phase heat transfer crises are not encountered in furnaces maintained at supercritical pressure and so constant pressure allows greater flexibility and use of conventional design. By employing furnace recirculation smoothly over the entire operating range, low load does not dictate furnace design and a furnace can be designed with:

* vertical, self-supporting, smooth-bore tubes,
* a single upward pass with same simple construction as a conventional drum unit, and
* no intermediate mixing or external piping is necessary.

Figure 11 shows the Riley Recirculating Supercritical unit with these features, in operation at SCE&G Wateree Station Units 1 & 2 since 1970.



400 MW Supercritical steam generator Riley Power Inc.

Figure 11. Riley Recirculating Supercritical Design for Constant Pressure Operation

Furnace Thermodynamic and Overall Sizing Differences

Beyond these obvious constructional differences, a sliding pressure furnace (evaporator system) must be sized to yield a greater outlet enthalpy, and so requires a greater heat duty and furnace size.

To illustrate this, the steam generator operating conditions and trends are compared on an enthalpy-pressure (H-P) steam diagram (Figure 12). This steam property diagram is used to trace the rising heat content (enthalpy) of the steam as it flows and loses pressure through the boiler.

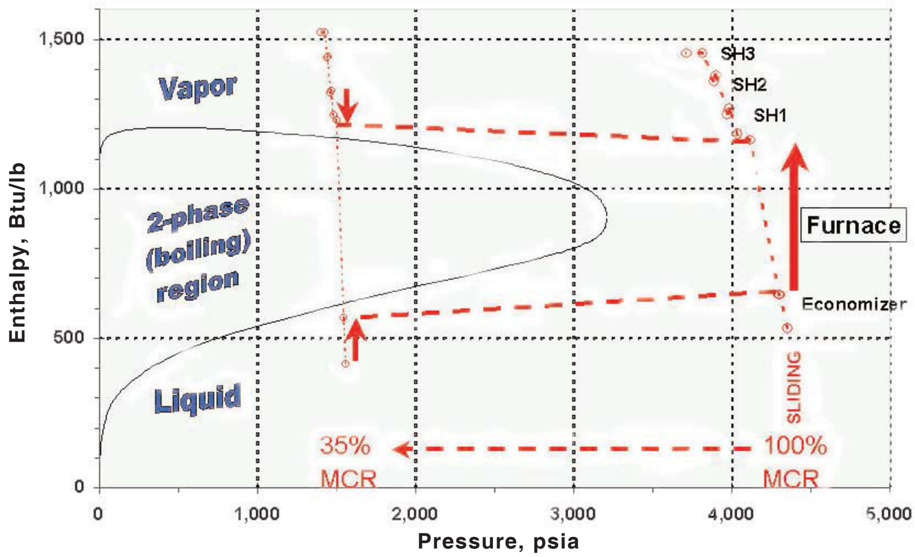


Figure 12. Steam Diagram, Sliding Pressure Operating Trend

As Figure 8 had indicated, sliding pressure as load is reduced results in the furnace being operated in the subcritical, 2-phase boiling region, typically at loads below 70-75% MCR. The nearly horizontal dashed lines in the chart above indicate the trend of furnace inlet and outlet conditions over the load range. To accommodate the 2-phase boiling condition of steam, there are specific steamside conditions that must be fulfilled at the minimum once-through load, and so it is sometimes low load — rather than full load — that determines the furnace or evaporator system heat duty and size.

* The economizer size is limited to prevent steaming in the economizer;

* The furnace size must be sufficient to produce dry steam in once-through mode to prevent introduction of liquid water into superheaters.

The furnace sized for a certain minimum once-through load produces the indicated conditions at full load, including the total heating duty (large red arrow) and the furnace outlet enthalpy and temperature. So, not only does the selection of minimum once-through load have consequences on the steam flow area and full load pressure drop, it also drives the overall furnace size and operating steam and metal temperatures. It is interesting to note that the sliding pressure furnace is essentially sized like one would size the evaporator system for a 1500 psig industrial unit (Figure 12). Often these medium-pressure industrial units employ a boiler bank or convective evaporator section to supplement the boiling heat duty while limiting the furnace size. This has been done in at least one instance for a supercritical Benson boiler, but is relatively rare due to its complication and expense. In summary, in addition to the usual gas-side furnace sizing criteria for emissions and slag control, the sliding pressure furnace must have a definite minimum size for steam heat duty.

In contrast, constant pressure units can stay in the supercritical, single-phase region and thereby have no such waterside sizing criterion. Figure 13 shows in blue the operating conditions of the constant pressure, Riley Recirculating unit over the same load range. The usual gas-side furnace sizing criteria that apply to any operating pressure unit — such as firing arrangement requirements, residence time and burnout, emissions considerations, and exit gas temperature limits for slagging and fouling control — will dictate. Depending on the particular fuel and fire-side conditions, the constant pressure furnace could be sized as indicated (large blue arrow). Note that while the sliding pressure furnace must be sized like an industrial boiler, the constant pressure furnace can be sized as one would expect for a high-pressure subcritical, natural circulation unit. This becomes quite apparent by review of Figure 14.

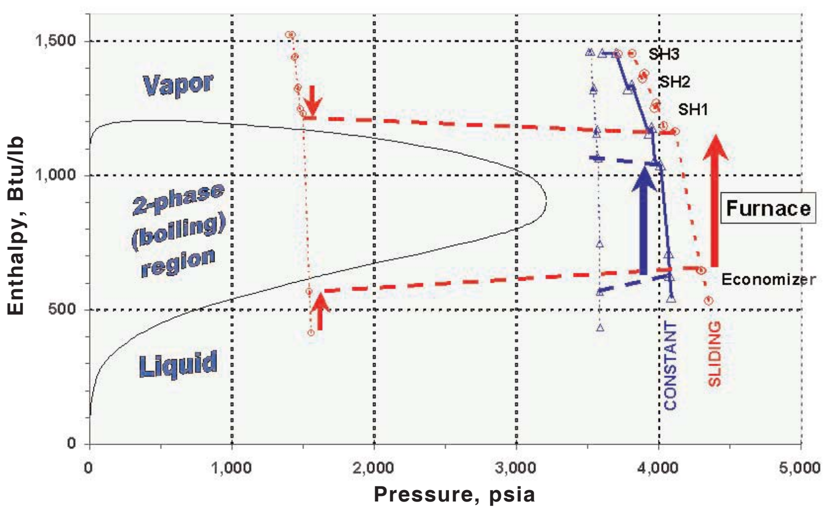


Figure 13. Steam Diagram, Constant and Sliding Pressure Operating Trends

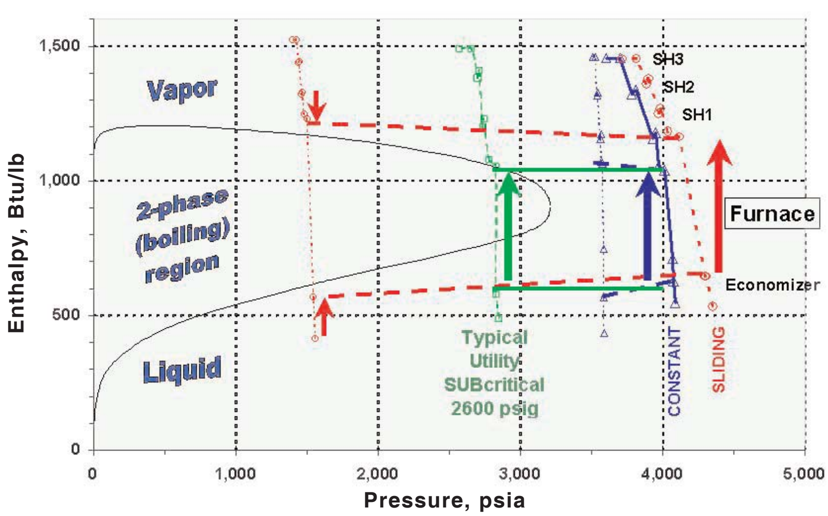


Figure 14. Steam Diagram, Relative Furnace Heating Duty

But unlike natural circulation units, the supercritical unit remains flexible in its performance since it does not have a fixed evaporator (furnace) end point. Evaporative and superheat duty can be shifted between furnace and convective surfaces in response to changes in fuel, slagging, or other conditions. This feature is not limited to Benson, Sulzer, or Once-Through designs, and the constant pressure design retains this flexibility at all loads, whereas a sliding pressure unit has less flexibility as pressure is reduced and margin above saturation (2-phase boiling) decreases.

Nearly as important as this size difference, the furnace outlet temperature of the constant pressure unit can be significantly less than that from the sliding pressure unit. Further, the thermodynamics of steam are such that, at the greater outlet enthalpy level required for the sliding pressure unit, temperature is much more sensitive to differences in enthalpy between furnace tubes. This increased sensitivity is partly mitigated by the heat absorption equalizing effect of the spiral tube arrangement around the sliding pressure furnace.

These are especially important points for extension to UltraSuperCritical conditions, where it is found that sliding pressure designs will have very high furnace outlet temperatures (approaching 1000-1100°F) and may require advanced alloys for the furnace walls. The potential reduction in furnace outlet design temperature with constant pressure becomes greater as the final steam conditions are advanced, and furnace materials options and margins are enhanced.

To visualize the implications of the steam-side heat duty differences, Figure 15 shows a constant pressure furnace — again in blue — that has been designed according to the universal gas-side criteria, and results in a furnace outlet steam enthalpy of about 1050 Btu/lb (760°F). The sliding pressure furnace in red is about 20% larger in order to yield the required outlet enthalpy of 1150 Btu/lb (790-800°F). Since the larger furnace is effectively accomplishing some of the superheat duty, the radiant superheater can be reduced accordingly, but the net cost increase is positive. Further, a particular advantage of the Riley Recirculating supercritical design is that it does not require intermediate furnace mixing, not only saving the associated piping costs, but also permitting a close-coupled backpass and eliminating the tunnel section otherwise required.

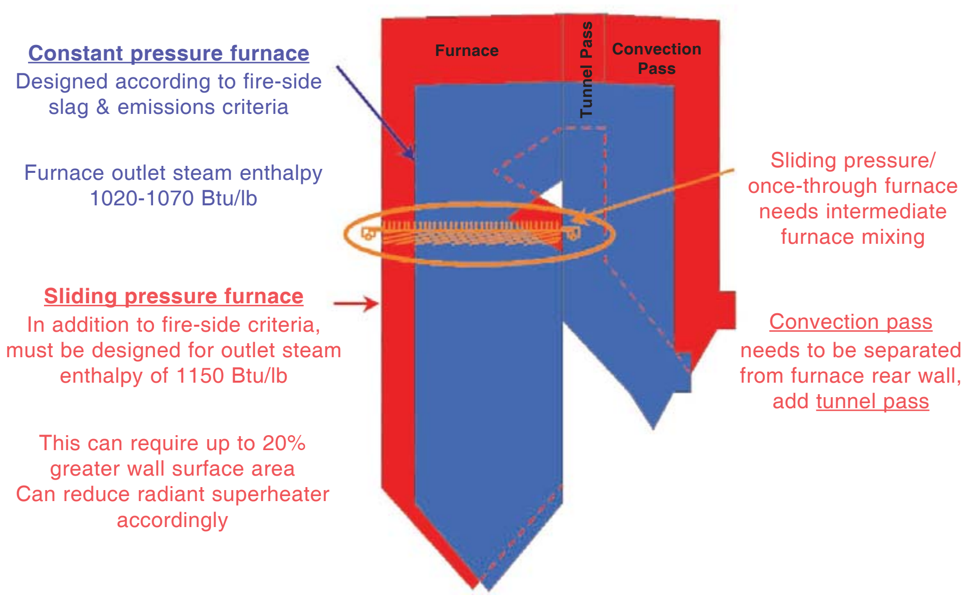


Figure 15. Furnace Sizing Difference Due to Furnace Heat Duty Requirements

The primary differences in furnace construction and size are estimated to result in 4-5% greater overall boiler cost for sliding pressure designs. For a 650 MW unit, this differential is about $6-7 million including materials and erection. This cost differential is due to only the tube circuitry, intimate support, erection, and overall furnace size differences, and for now we have ignored further differences in tube materials, tunnel pass elimination, and steel, building, or foundation differences, all of which lead to even greater costs for a typical sliding pressure design.

Operational Differences

A key remaining question is: Is it worth it? Can the additional capital investment in a sliding pressure plant be recovered by operating cost advantages in the USA market? Noting that differences in low-load heat rate and uncertainty of new plant dispatch loading are often cited as the primary attraction for sliding pressure decisions, we next evaluate heat rate trends for the various operating modes and later calculate the implied operating costs for comparison.

Heat rate comparison data was obtained from two major turbine manufacturers involved in the recent advanced plants being built in other regions of the world. As Riley contacts and collects more comparative data, our evaluations will be revised, but so far the information from the sources agrees reasonably well. Figure 16 is a plot of some of this comparative data, showing the difference in turbine cycle heat rate compared to pure sliding pressure, for various turbine / cycle control schemes. The data considers that the boiler feedpump is turbine-driven and so its effect is included in the cycle heat rates. One can see the significant heat rate penalty as normally understood for a throttle-controlled constant pressure steam cycle (blue dashed line), but throttle-control of pressure upstream of the turbine is not considered viable for modern utility plants and there is another option already in common use. The trend for nozzle-control constant pressure (blue solid line) yields an operating efficiency nearly equivalent to sliding pressure operation. Nozzle-control for constant pressure refers to the multiple admission valves operated in partial-arc admission mode to the HP turbine. The shape of the nozzle-control constant pressure curve indicates that at the valve- (best-) points (a 4-valve inlet example is shown), the corresponding valves are fully open with negligible throttling loss and the efficiency is equivalent to pure sliding pressure at various loads. The red dashed line represents throttle-control sliding pressure, or modified sliding pressure as previously explained. Here, the initial throttling loss of modified sliding pressure is apparent down to 90% MCR, and then pressure is allowed to slide at lower loads with no change to throttle valve position. The red solid line labeled as nozzle-control sliding pressure lies at zero differential, and can be considered to represent pure sliding pressure mode.

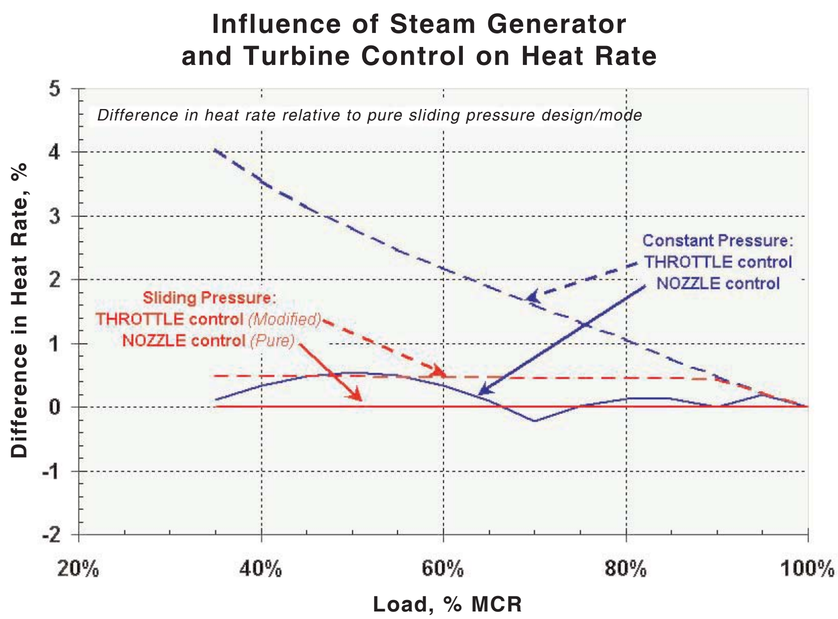


Figure 16. Differential Heat Rate Curves for Various Turbine Control Methods

While the nozzle-control differential is directly indicated to be quite small, minor differences in heat rate can indeed involve large sums of money at the scale of utility power generation. To convert these differentials into the real-world context of net plant heat rates, the differentials were overlaid onto sliding pressure turbine heat rate trends (examples from three independent turbine suppliers), and then combined with steam generator and auxiliary power efficiency trends to yield net plant efficiency trends vs. load. These heat rate trends, shown in Figure 17, can then form the basis of operating cost calculations, which can be compared among the various pressure control types.

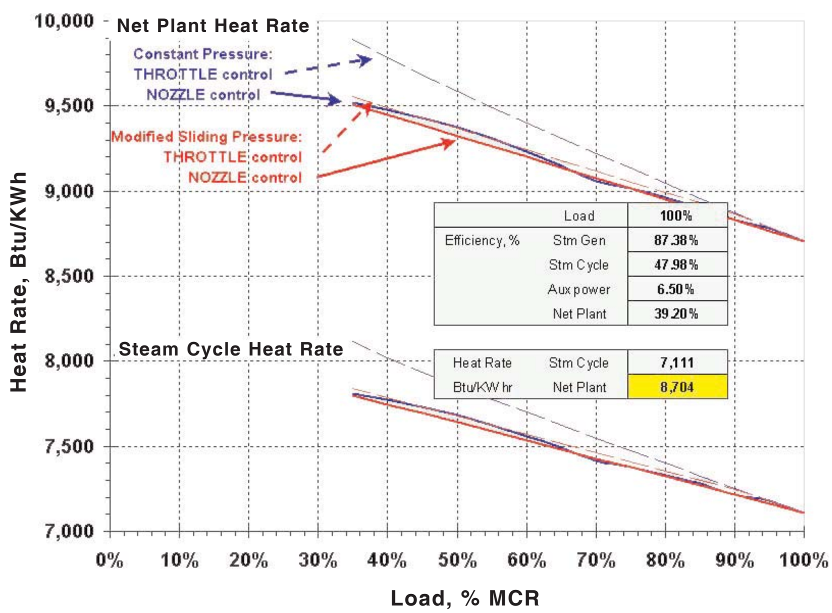


Figure 17. Heat Rate Trends Vs. Load

Economic Analysis

While the various plant types can be designed for the same full-load efficiency, what if the load must be eventually, and then consistently, reduced? Based on the previous analysis of the future USA market and regional dispatch situation, the worst-case, long-term situation of daily load cycling (2-shift; not continual cycling) was considered to compare the operating costs for the various modes. Our primary interest is in comparing modern, nozzle-control constant pressure to sliding pressure operating costs, but we also perform the calculations for throttle-control for reference.

Figure 18 summarizes the primary variables in an analysis of plant operating costs. Variables outside of the plant "box" are market rates or prices of fuel, reagents, and byproducts that can be assigned a regional market rate. Carbon dioxide (CO_2) emissions can be assigned a value in the model (in the form of a tax or capture-related costs), but the value was set to zero in the results presented here. Items within the plant "box" are the plant specifics including unit size and operations that drive the scale of revenue involved. Fuel type includes the specific heating value and chemical composition to permit the reagent and emissions calculations, and of course influences the price paid per heating value unit.

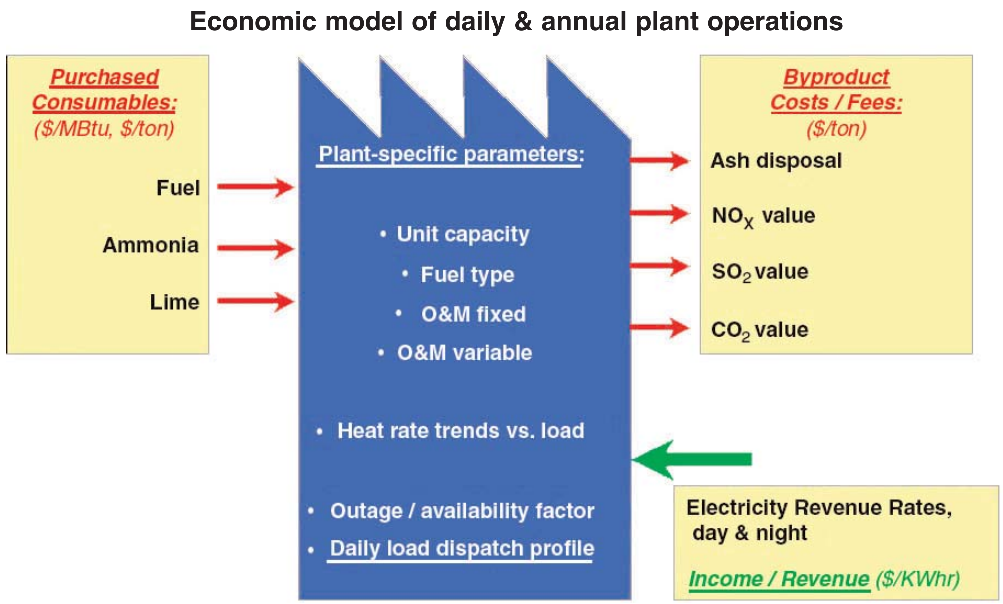


Figure 18. Plant Operating Economics Model

In the model, the daily load profile can be any combination of loads and times. As previously stated, a primary unknown in each regional competitive market is the load dispatch profile for a particular plant, and so the worst-case of daily load 2-shifting was considered. To gain a more complete picture, a wide range of possible two-shift load patterns was evaluated: each including 12 hours at full load and 12 hours at various reduced loads to simulate reduced night load dispatch (Figure 19).

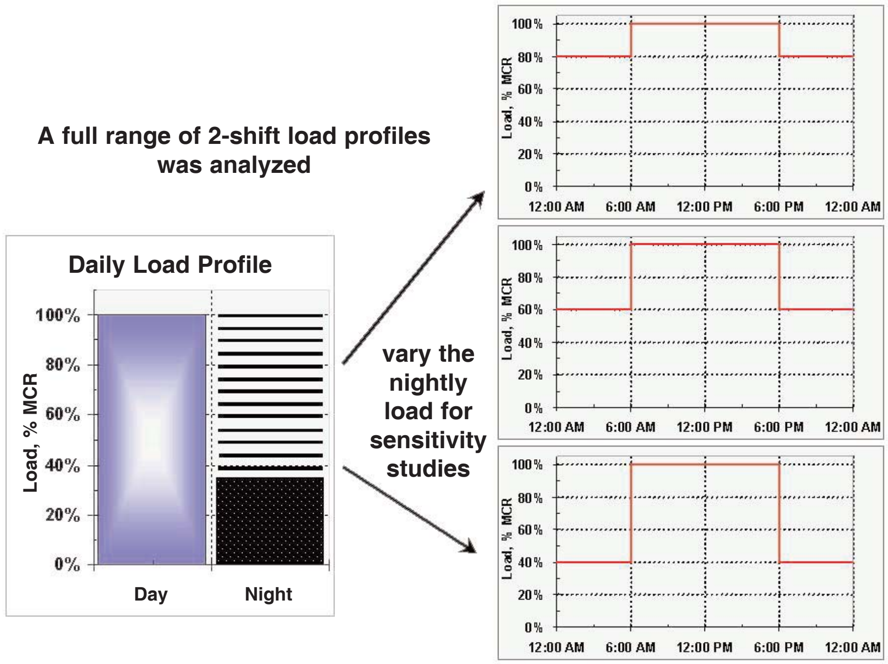


Figure 19. Range of Daily 2-Shift Load Profiles Evaluated

The economic model calculations were initially set-up for a particular plant design, which includes its design operating mode and associated heat rate trend over the load range. A daily load profile was specified, and the operating costs were calculated over the course of a typical year. The same model and conditions were repeated, but then with substitution of a different plant design operating mode with its own particular heat rate trend over the load range. The operating costs of the four plant designs / pressure operating modes were similarly calculated and were then compared for the same assumed daily load profile and all other conditions.

These calculations and comparisons were then repeated for the range of assumed night loads (35-100% load, in 5% increments) for sensitivity studies. Further, the entire analysis was repeated for a range of fuel prices and types.

The example results presented next are for a 650 MWn size plant, firing PRB coal, and reflect the following rates:

Capacity	MW net	650

Load / Usage		
Day	% MCR	100%
Night	% MCR	35-100%
Night, % of 24 hrs	%	50%
Availability / Outage	%	95%

Operating Costs		
O&M Fixed	$/KW capacity	$20
O&M Variable	$/MWhr	$3.50

Fuel		
Description	-	PRB
HHV	Btu/lb	8,500
SO2 removal	%	95%
NOx removal	%	90%

Fuel / Emissions Prices & Fees		
Fuel price	$/MBtu	$1.00-$1.40
Ash disposal	$/ton	$25
CO2 tax	$/ton	omit
SO2 credit	$/ton	$550
NOx credit	$/ton	$2,200
Lime price	$/ton	$60
Ammonia price	$/ton	$225

Though the differences are not readily apparent in the bar charts, Figure 20 basically illustrates the process and calculations described above, for one particular fuel and price.

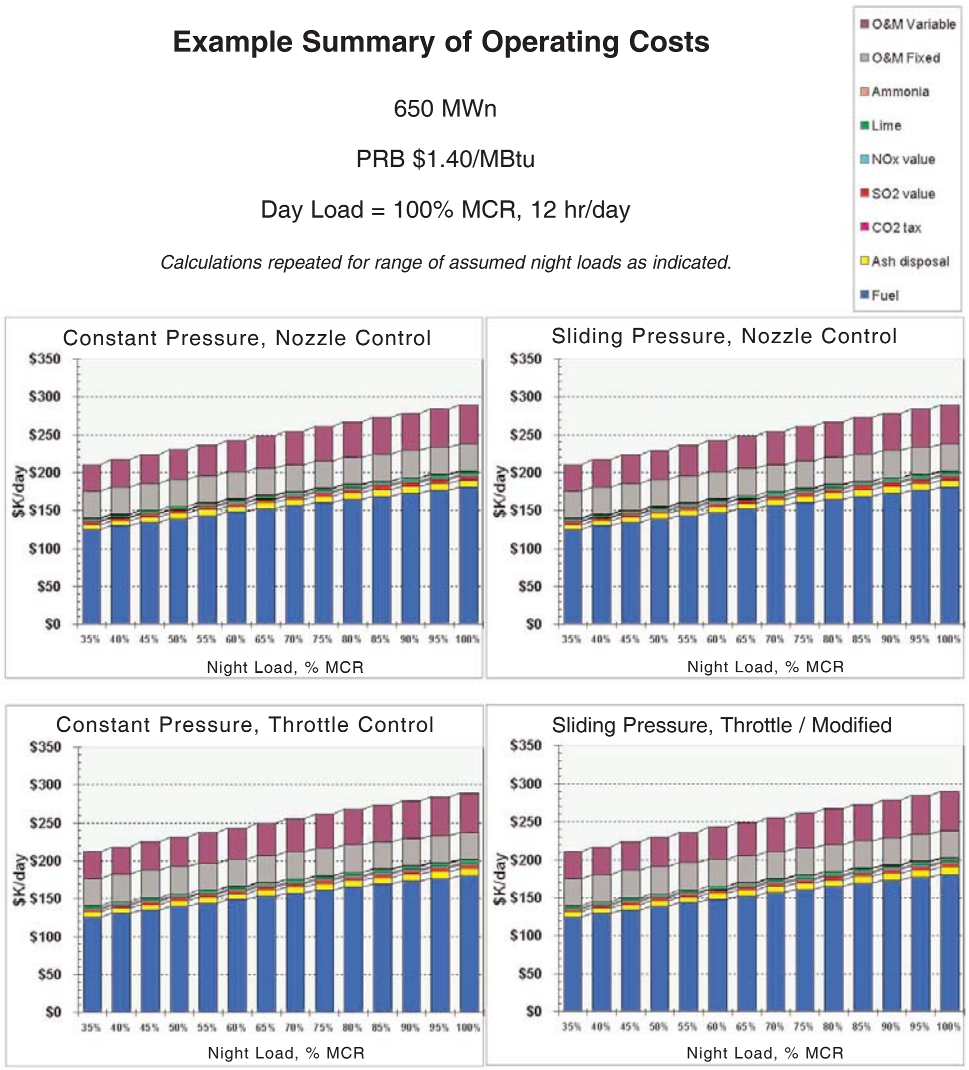


Figure 20. Calculation of Daily Operating Costs for Constant and Sliding Pressure Plants

The relatively small differences in operating costs are better realized by calculating and plotting the difference in annual operating costs between pairs of plant design operating modes (constant vs. sliding pressure with nozzle control, and constant vs. sliding pressure with throttle control) as a function of the assumed nightly load profiles. The results are shown in Figure 21, where positive values indicate a relative savings for sliding pressure operation over the course of a typical year.

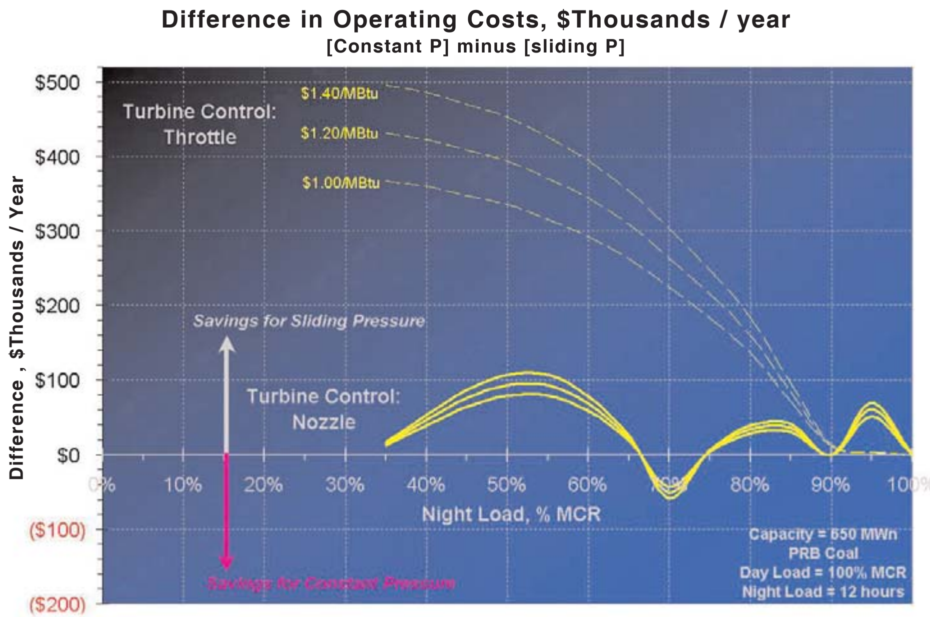


Figure 21. Difference in Annual Operating Costs for Range of Night Loads and Fuel Prices

As expected, the basic shapes of the differential heat rate curves are reflected in the differential operating costs when plotted as a function of the assumed night load. For the nozzle-control comparison (solid lines), due to negligible heat rate differences at the valve best points, there are several loads at which the operating costs are exactly the same (zero difference). If night load is anywhere near 100%, 90%, 75%, 65%, or 35%, the constant pressure unit is just as efficient as a sliding pressure unit. The efficiency and number of best loads is a function of the number of turbine admission valves used in sequential mode.

Although throttle-control for constant pressure will likely not be used in the future, the calculated results are also shown. This indicates the more widely believed result that constant pressure efficiency is progressively worse as load is reduced, and you see here a significant difference in operating costs as the nightly low load is reduced and as fuel price is increased. (The three solid lines for nozzle-control also correspond to the three fuel prices indicated.)

To illustrate a single potential situation for the 650 MWn example firing PRB at $1.40/MBtu, if the night load was set consistently at 60% MCR, a plant designed and operated with pure sliding pressure mode could realize a savings in operating costs of $80,000 over the course of a year, compared to a plant designed and operated with nozzle-control constant pressure.

With multiple valve best points and loads, the owner of a nozzle-control constant pressure plant might be able to shift limited amounts of load to or from the particular unit so that heat rate and operating costs would be minimized and nearly equal to those from a sliding pressure unit. Assuming that even this freedom is not available, we consider further a situation that over the lifetime of the unit, the nightly load is in fact nearly random or evenly distributed between 35-80% MCR. In this case, the sliding pressure unit operating costs would be only $34,000 — $46,000 / year less. This is equivalent to a present value of about $420,000 — $570,000 based on a simple 20-year evaluation life at 5%.

Under the conditions considered reasonable and discussed in this paper, the present value of operating cost savings with sliding pressure is not enough to cover the additional $6-7 million capital investment for the sliding pressure steam generator. Meanwhile, the sliding pressure turbine cost savings is reportedly estimated to be on the order of $0.5 million, and would be partly offset by any additional feedwater heater and steam generator costs to handle sliding pressure and any associated load and pressure cycling. Figure 22 further illustrates the significant gap between capital cost investment and present value of future operating cost savings with sliding pressure.

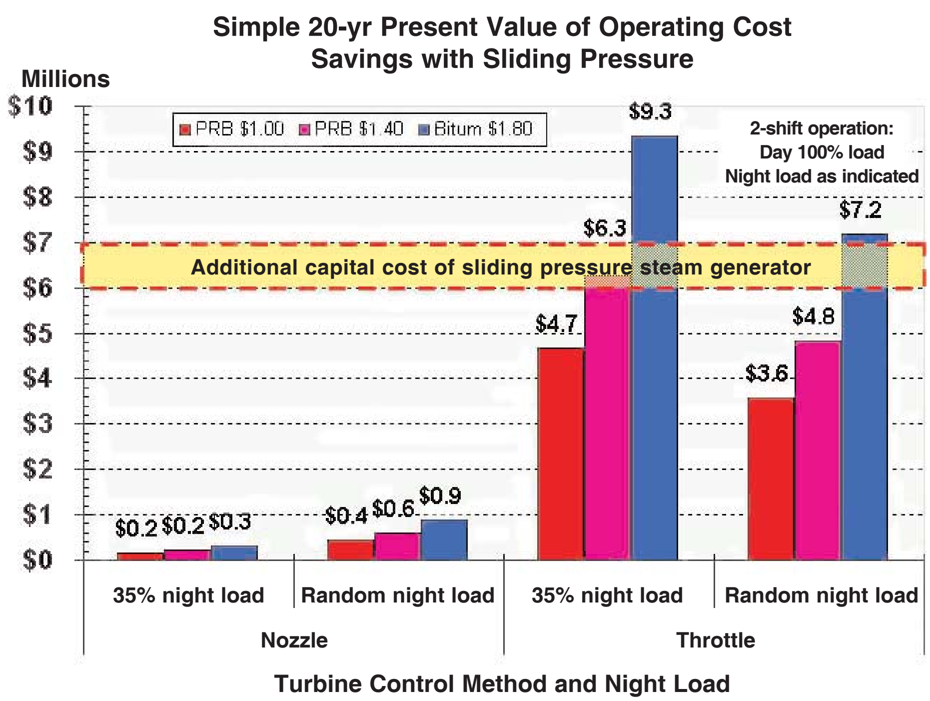


Figure 22. Comparison of Capital Investment and Operating Cost Savings

For completeness — and as a signal of future development efforts — it should be recognized that fast start-up and continual load cycling ability may be of value for a limited number of units in particular regions, though the value is relatively difficult to quantify. Regarding continual load cycling, we believe that the market and dispatch evaluation presented in this paper will apply to the bulk of the coal-fired capacity addition needs, and so continual load cycling of new coal units, beyond controlled nightly reductions, will be a smaller proportion to be strategically determined for each grid region. Regarding startup, we note that not all of the startup systems and features employed on modern units around the world are inherent or exclusively applicable to sliding pressure operation, and the expense of once-through sliding pressure steam generators need not be assumed to include many of the attractive, modern start-up features. The Riley Recirculating units in operation since 1970 already prove the successful application of recirculation to facilitate start-up of a constant pressure supercritical unit. For the future generation of coal-fired plants in the USA, Riley is developing and integrating modern startup features with appropriate plant designs for the range of expected domestic needs, both constant- and sliding- pressure applications.

Summary

The expected regional market conditions in the USA tend to favor a diversification of technologies and operating capabilities, but it seems clear that the most economically efficient generating units will be favored for load dispatch, and so much of the supercritical coal-fired additions will not be frequently shut-down or continually load-cycled.

With a significant 4-5% cost advantage in the steam generator, new nozzle-control constant pressure supercritical plants will be appropriate assets in each region since they operate with nearly the same heat rate as sliding pressure plants across the operating load range. The cost advantage is expected to grow as design steam conditions are gradually advanced toward UltraSuperCritical goal conditions in pursuit of greater efficiency and reduced emissions.

Mainly because all of the recent coal-fired plants have been built for market conditions in other regions of the world, modern start-up systems have been developed and applied to sliding pressure plants only. Recognizing that both sliding pressure and constant pressure plant options are appropriate for the USA market conditions, Riley Power is continuing its efforts to adapt and develop modern start-up and other features to both types of plants.

[1] It is the authors' opinion that thermal heat release of stored energy resources — whether nuclear, fossil, or renewable; both in the forms of "lost" energy and "useful" energy, at all stages of use — represents the direct and primary anthropogenic influence on the heating of our surroundings, and that increased solar heat retention due to elevated CO_2 in the atmosphere is perhaps a smaller secondary effect. If so, it is only reduction of energy consumption through improved economy and efficiency — and not byproduct capture — that will slow any human influence on global warming.