Approaching Free Electricity: How the Real World Differs from Thermodynamic Models

Sean Casten
Turbosteam Corporation
Thomas O'Brien
Integrated Process Solutions

ABSTRACT

As distributed generation moves from concept to reality, industrial and institutional steam customers are realizing that they can often use steam turbine/generator technology in their existing steam networks to make "opportunistic" electric power and significantly reduce their annual energy costs. Historically, turbines operating in this "backpressure" mode have been assumed to have a net fuel-to-electric efficiency that is approximately equivalent to the on-site steam boiler, because every unit of energy removed as electricity must be replaced with a unit of heat. Under this assumption, backpressure turbine/generators easily become the most efficient power generation technology ever invented. While this logic is thermodynamically true, it often breaks down in practice, where the nuances of steam plant operation make it possible to generate electricity at an even higher efficiency with backpressure turbine/generators. In some cases, the power produced by backpressure turbine/generators is actually free.

INTRODUCTION

In industrial and institutional steam systems, backpressure turbine generators can produce electricity with no added fuel, and often produce electricity with well in excess of 100% fuel efficiency. This does not repeal the laws of physics, or rival cold fusion for the scam of the year, but occurs because these systems reject hot condensate to the sewer. As a result, the net impact of utilizing additional enthalpy from the steam in a backpressure turbine generator is simply to lower the temperature of rejected condensate—making the resulting electricity fuel-free. The full explanation follows, but first the 20,000-foot view.

Most steam systems send out relatively high-pressure steam to minimize distribution pipe size, and then use pressure-reducing valves (PRVs) at the points of use. These PRVs lower steam pressure, but not enthalpy, so the heat exchangers receive superheated steam. If a backpressure turbine is substituted for a PRV, it will lower the steam pressure and the temperature. The typical backpressure turbine generator converts roughly 96% of the enthalpy removed to electricity. Because heat exchangers are sized for maximum expected load, under most conditions, the heat exchanger is oversized and will produce the required hot water without any added steam. Instead, it further cools the condensate relative to when supplied with superheated steam. When all condensate is returned to the boiler, the enthalpy shortfall of the cooler condensate must be made up with more fuel, reducing the efficiency of the backpressure turbine to about 82% fuel to electricity. However, most district steam systems and many industrial steam systems do not return condensate, so the enthalpy loss simply cools the water and lowers the need for cooling condensate before dumping in the sewer. In the no condensate return situation, the backpressure turbine generator produces electricity with no added fuel, and reduces the need for atemperation water. In the partial condensate return systems, some fuel will be added, but may achieve apparent fuel-to-electric efficiencies of several hundred percent.

IMPLICATIONS

Steam users served by most district steam systems (and selected use points in industries that are concerned with contamination and thus dump condensate in the sewer) can extract some of the presently wasted enthalpy and produce fuel-free electricity. In the past, the analysis of these systems has assumed the enthalpy that is removed would need to be made up by purchasing added steam, but this is (in most cases) not necessary.

This presents a great opportunity to cities like New York, Philadelphia, Boston, Detroit, Denver and San Francisco, to name a few, to buttress their congested urban electric transmission systems with fuel-free power. It also makes installation of hybrid backpressure turbine/absorption chiller systems in these cities significantly more economical than has been assumed. In addition, process industries should reexamine past studies of backpressure turbine opportunities with a corrected fuel cost analysis. If the steam is not being returned, out of concern for possible contamination, then there is a hidden opportunity to cut costs by extracting electricity from presently wasted enthalpy.

Finally, these installations produce no added pollution, either of regulated pollutants or carbon dioxide, thereby presenting a compelling argument to those who believe that the environment is the enemy of the economy. Every backpressure turbine/generator installation represents a pollution control device with a positive rate of return.

THE DETAILED VIEW OF FREE ELECTRICITY FROM BACKPRESSURE TURBINES

The marginal cost of electricity generated by any power generation technology, independent of capital cost recovery and operations and maintenance is simply the cost of fuel divided by the power generation efficiency. Thermodynamics tells us that efficiencies are always less than 100% (e.g., you never get as much useful energy out as you put in), leading us to conclude that the only way to get free electricity is to find free fuel. However, because fuel usually has a price, it becomes economically imperative to use a generator with the highest possible generation efficiency.

When installed into existing steam systems, backpressure turbine/generators are the most efficient form of power generation ever invented. To a first approximation, the efficiency is equivalent to the boiler efficiency multiplied by the generator efficiency, reflecting the fact that any enthalpy removed from the steam as turbine-shaft power displaces a unit of heat to the process—and any shaft power that is converted into electric power will have additional losses in the generator. At modern boiler and generator efficiencies, this equates to between 75 and 85% fuel-to-electric efficiencies.

It is important to recognize that this does not imply a violation of the second law of thermodynamics, which (according to Carnot) teaches that a backpressure turbine/generator should be unable to achieve efficiencies higher than 20% given the temperatures available in most steam systems. Indeed, if one measures the total fuel added to the boiler and compares this to the total electricity extracted in the turbine/generator, we can verify that the second law has not been violated. However, this analysis is a purely academic exercise because it fails to recognize that at a practical level—that is, at the level that impacts economics and the environment—efficiency must be calculated at the margin. At this level, one asks the question: "How much more fuel do I have to burn to generate a kWh?" To answer this question, it is the marginal—rather than the total—fuel consumption that matters. So long as the boiler is going to operate anyway to provide process thermal loads, the marginal fuel consumption is only that required to make up for enthalpy removed as electricity—and thus the power generation efficiency comes to approximate the efficiency of the steam boiler.

But is that as good as you can do? In some cases, the answer is no—you can actually do much better. The logic for backpressure turbine/generator is contingent on the existence of a pre-existing steam plant and thermal load to achieve the efficiencies outlined above. In many cases, the way in which that steam plant is operated can make the effective efficiency of power generation higher still. The balance of this document explains how.

PART I: WHY DO YOU HAVE TO PAY FOR HEAT?

Let's review a backpressure turbine/generator installation as compared to a pressure reduction valve (PRV) that it replaces. The following example shows the conventional comparison that is made, in which the backpressure turbine/generator is designed for a slightly larger mass flow than that of the PRV.

Example 1: Conventional Backpressure Turbine/Generator Analysis

A backpressure turbine/generator is designed to replace a PRV that reduces 30,000 lb/hr of steam from 220 psig (saturated) down to 40 psig. What is the impact on steam flow and/or process condensate temperature, if the process is currently designed to condense—but not to subcool—the steam, which returns to the boiler at 38 psig?

First, let's calculate the enthalpy requirements of the process ($Q_{process}$), which are simply the mass flow (m), multiplied by the specific enthalpy (h) drop across the process, or:

$$Q_{process} = m \times (h_{PRV} - h_{condensate})$$

The PRV does not remove any enthalpy from the steam, and therefore, will have an enthalpy that is equivalent to the enthalpy of 220 psig, saturated steam (h_{PRV} = 1,200.4 Btu/lb). The condensate will have an enthalpy equivalent to 38-psig water, at its boiling temperature ($h_{condensate}$ = 253.6 Btu/lb). Substituting, we thus calculate that the process requires:

$$Q_{process} = 30{,}000 \text{ lb/hr} \times (1{,}200.4 - 253.6) = 28.404 \times 10^6 \text{ Btu/hr}$$

At a 51% isentropic turbine efficiency, the 40-psig exhaust steam from the turbine would have a specific enthalpy, $h_{turbine}$ = 1,142.3 Btu/lb. Substituting this into the above formula and solving for the mass flow at a constant $Q_{process}$, we can calculate that the mass flow requirements at constant process condensate conditions are:

$$m_{new} = Q_{process}/(h_{turbine} - h_{condensate})$$
$$m_{new} = 28.404 \times 10^6 \text{ Btu/hr}/(1{,}142.3 - 253.6 \text{ Btu/lb})$$
$$m_{new} = 31{,}961 \text{ lb/hr}$$

These calculations are shown graphically in the following two figures.

Note in this analysis that the net effect of the turbine/generator is to increase the steam flow through the system. The reason for the increased flow at first seems to be fairly obvious—we can't disrupt the process (which in this example requires 28.4 × 10^6 Btu/hr of heat), and we can't violate the first law of thermodynamics, which says that energy removed as shaft power must no longer be present as heat. Therefore, we increase the mass flow to make up for the lost enthalpy—right?

Well, sometimes. Notice that there is an implicit assumption in this calculation that the condensate returned from the process must remain at a fixed condition (in this case, 254 Btu/lb). In some cases, this is a legitimate concern for example, in any instance where a reduction in ΔT across the heat exchanger would violate a "pinch" and compromise the heat flux to a process.

However, good engineering practice mandates that in most cases, heat exchangers are slightly overdesigned with built-in safety factors such that if the temperature of the condensate falls slightly (e.g., if the condensate is slightly sub-cooled at the tail-end of the hot section), there is still sufficient heat transfer. Furthermore, any heat exchanger designed to handle widely varying heat loads—for example, most space heating loads, which show tremendous seasonal variations in heat duties—are designed for a worst-case condition that at worst occurs just once per year.

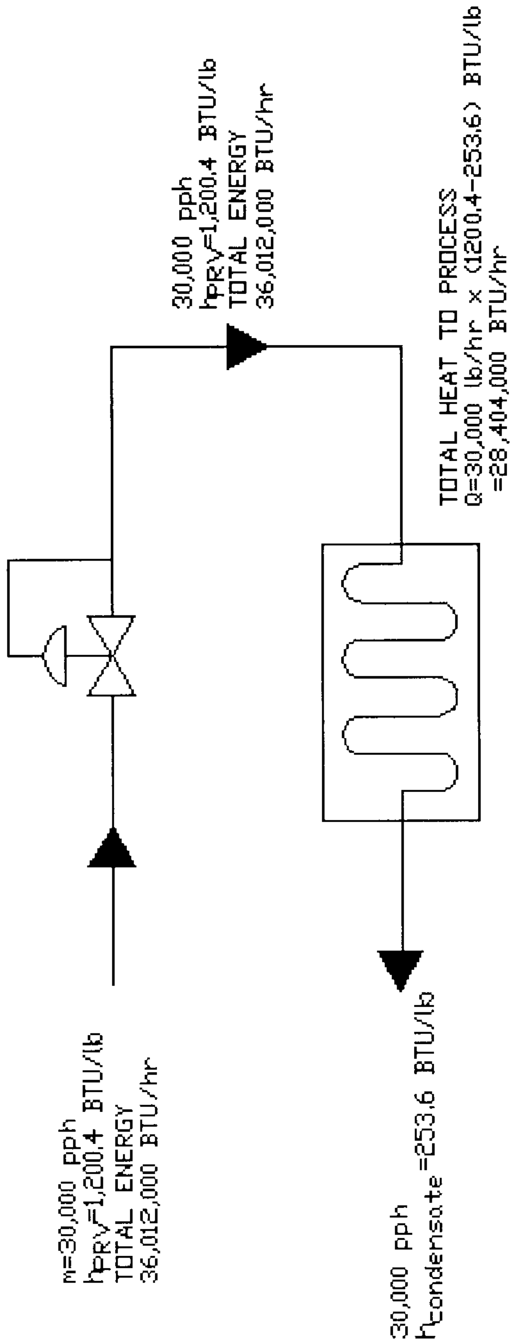

Figure 1. Steam Energy Flows with a Pressure Reducing Valve

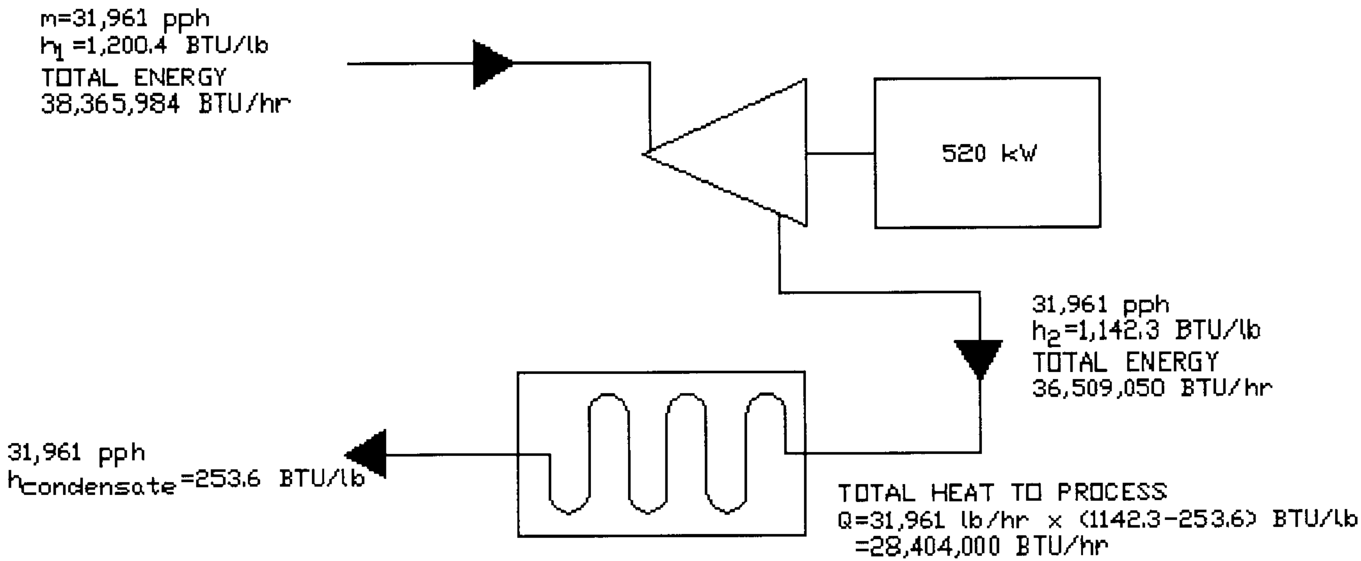

Figure 2. System Energy Flows with a Backpressure Steam Turbine/Generator

In both of these situations, the impact of the turbine/generator installation is not to require a greater steam flow, but simply to result in a lower condensate return temperature.* We can illustrate the magnitude of this difference by way of example.

Example 2: Backpressure Turbine/Generator Analysis Including Real-World Heat Exchanger Operation

Consider the same process specifications identified in Example 1 above, but assume that the steam system is not limited by condensate return temperature. What does this imply about condensate return conditions?

In this case, we can simply solve for $h_{condensate}$ at a constant mass flow and $Q_{process}$:

$$h_{condensate} = h_{turbine} - Q_{process}/m$$
$$h_{condensate} = 1{,}142.3 \text{ Btu/lb} - 28.404 \times 10^6 \text{ Btu/hr}/30{,}000 \text{ lb/hr}$$
$$h_{condensate} = 195.5 \text{ Btu/lb}$$

At 38 psig, this implies that the water returning to the boiler from the process has a temperature of 227°F—or approximately 57°F of subcooling. Energy flows after turbine/generator installation in this non-condensate limited case are shown in the following figure. Compare the energy flows in this figure to those in the previous example to see how these considerations of condensate requirements effect system operation.

So can steam plants tolerate this degree of additional sub-cooling in their heat exchangers? It depends, but in many cases, the answer is yes, because of safety factors built into the equipment. Note that this does not yet have any impact on the cost of power generation; if the condensate is cooler than it was before, then the boiler must work harder, and power generation efficiency is still the product of boiler × generator efficiencies. However, this potentially bi-modal impact has important ramifications, as we shall shortly demonstrate.

---

*Note that this depends upon the manner in which the heat exchanger is controlled. If the heat exchanger is controlled based on a condensate temperature set-point, the net impact will still be to increase steam flow. However, if the control is based on a temperature set-point on the cold-side exit, falling condensate temperature will not (initially) impact steam flow.

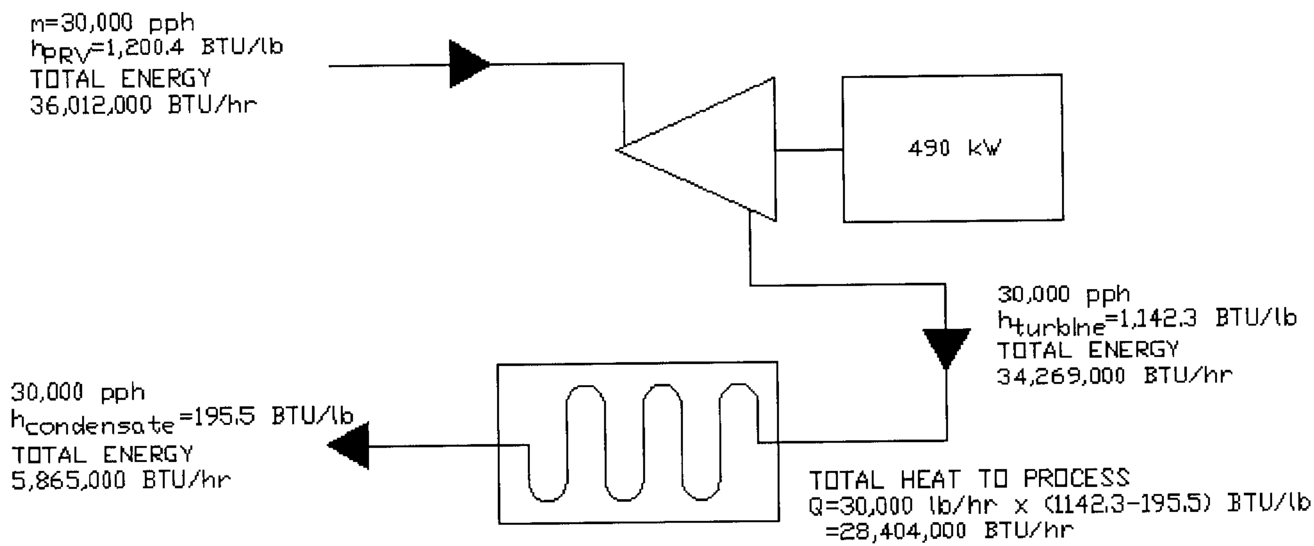

Figure 3. Energy Flows After Backpressure Turbine/Generator Installation (Non-Condensate Limited Case)

PART II: HOW MUCH DO YOU PAY FOR HEAT?

So now we know that there are two different ways that a backpressure turbine/generator can "penalize" a steam cycle, both of which lead to increased boiler duties. It is, therefore, worth asking the question—how much does it cost to run a boiler a little bit harder?

It is virtually impossible to obtain boiler performance curves as a function of rated load for actual, working boilers—steam meters are usually either not installed or inaccurate, and fuel meters often lack sufficiently instantaneous data to allow one to correlate instantaneous steam production with instantaneous fuel consumption. Nonetheless, we can make several broad generalizations that apply to all boilers.

A boiler operating at very low load is expending much—if not most—of its energy heating up the thermal mass (walls, pipes, etc.) of the boiler rather than raising steam. However, as load increases, marginal fuel Btus are increasingly being used just to raise steam.

Most operating boilers have fixed-speed air fans. This means that a boiler at low flow is consuming a large amount of excess air—and paying an efficiency penalty to heat up that excess air—while a boiler at higher flow is increasingly using its marginal fuel Btus just to raise steam.

As a result, boiler efficiency tends to increase with load. The following figure is a simplification of this relationship, the shape of which can be applied generically to boiler design.

Using this curve, suppose that we are considering a backpressure turbine/generator installation in which the boiler is operating at 63% rated load and 82.5% average efficiency before installation (the X on the curve) and 69% of rated load after turbine generator installation. At this modest increase in boiler capacity, the average boiler efficiency increases to 83%. However, this efficiency applies to all the steam produced by the boiler—even the pounds produced in the low-efficiency, low load conditions at start-up. The efficiency with which our 6% marginal steam was generated is thus:

$$(69 - 63)/(69/83 - 63/82.5) = 88.6\%$$

In other words, our power generation efficiency is 6% higher than we would have projected from our simple assessment using av-

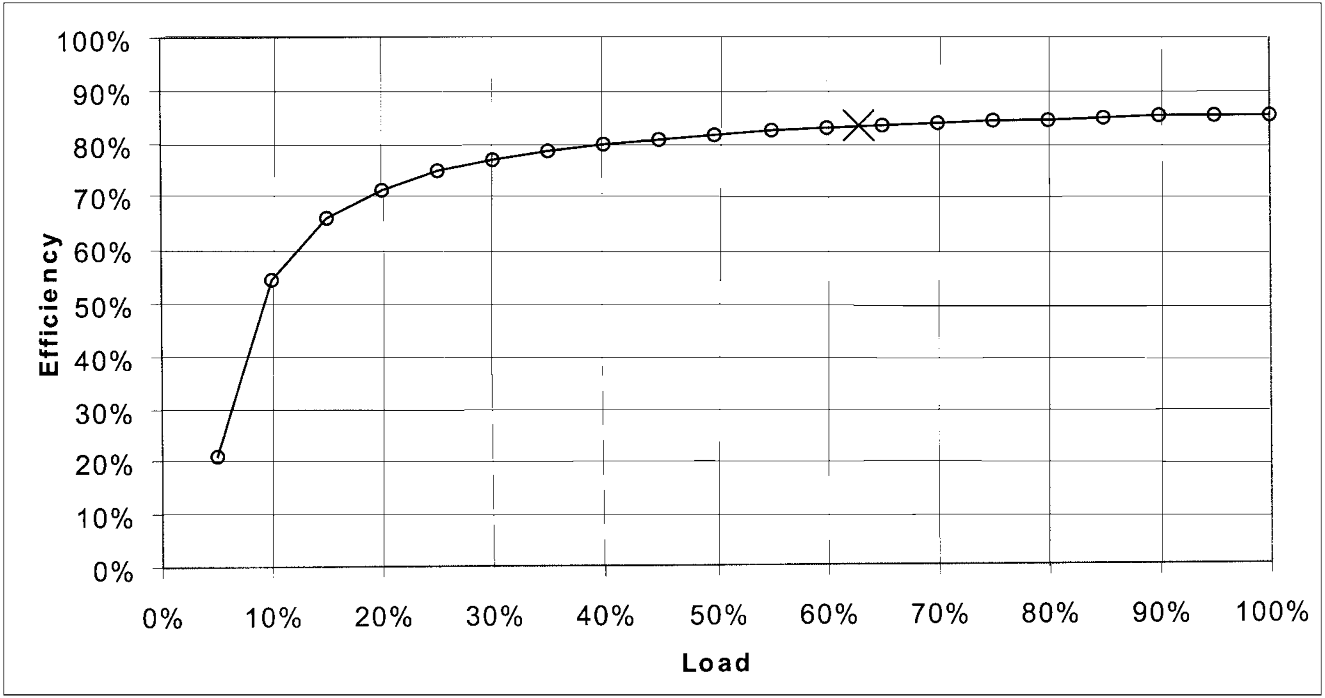

Figure 4. Boiler Efficiency Curve, as a Function of Load

erage boiler efficiencies.* This is an important distinction, because this is the efficiency impact that a backpressure turbine/generator user will actually observe after installation if they are tracking boiler fuel costs and/or emissions impacts.

This obviously doesn't make the power free, but it does mean that the power will be cheaper than we might have otherwise assumed. The degree to which this phenomenon is important ultimately depends on both the shape of a particular boiler's load curve and on the current (pre-turbine/generator installation) load condition. It is also extremely difficult to quantify. But it is real.

PART III: WHAT IMPLICIT ASSUMPTIONS HAVE WE MADE THUS FAR?

We've now realized that our assumptions of fixed condensate temperature are not always relevant, and further, that our assumptions of constant boiler efficiency are also not quite accurate. What other assumptions are we implicitly making in our analysis?

There is one remaining critical assumption—and as soon as we take it away, we actually have the potential to generate free power from a backpressure turbine/generator, under some conditions. It's in the condensate.

When we fixed our condensate temperature from the process, we raised our steam production, and the cost of power was obvious. However, once we include condensate sub-cooling, it becomes apparent that we have unduly penalized ourselves through the implicit assumption that all condensate returns to the boiler. In an efficiency-optimized world, everyone would return 100% of their condensate back to their boilers. It saves money on boiler treatment chemicals, and it also reduces boiler fuel costs because the condensate that returns to the boiler still contains some residual heat. However, the real world isn't efficiency optimized. In many cases, steam plants simply don't get the capital dollars that go to other processes. Seals rupture, pipes corrode and slow leaks gradually develop through the condensate system, leading to less

*Note that we would come to a similar conclusion if we assumed that we did not have to generate additional steam, but rather had to add a slightly larger amount of heat to the condensate. Because we would still be measuring a marginal boiler efficiency, the logic remains the same.

than 100% of the condensate being returned to the boiler. (In completely closed steam loops, a "tight" steam plant might only return between 70 and 90% of its condensate back to the boiler.)

However, even in the best steam plants, there are often cases where condensate is intentionally not returned to the process. For example:

Any processes that inject steam directly into their processes will not make any effort to recover the resulting condensate because it becomes contaminated with process chemicals and would require unacceptable water treatment costs to bring it back to boiler-purity requirements. This is found in many chemical and food operations, some hot-water generators and in every lumber kiln.

Many processes that use steam to indirectly heat caustic materials (e.g., much of the chemicals industry) intentionally "dump" their condensate so as to minimize the risk that these chemicals will accumulate in their boiler water system.

Ultimately, the universality of <100% condensate recovery is demonstrated by the fact that every boiler plant has a make-up water line, adding cold water back into the boiler to make up for the condensate lost in the process.

PART IV: SO IF WE FACTOR ALL THIS IN, WHAT HAPPENS TO GENERATION EFFICIENCY?

Now things get interesting. On the boiler side, we clearly realize a higher efficiency at the margin than we do over an average load. However, the big story is on the condensate side.

If the process design allows us to cool the condensate further than we were able to with a PRV, and if that condensate is not being returned to the boiler—the power is free. The penalty imposed by the turbine/generator in the form of lower temperature condensate now truly doesn't matter because the heat of that condensate is not being returned to the boiler. We need to be very clear that these are both "if"s, and need to be recognized as such. They do not apply to every circumstance, but they aren't completely atypical either.

In actual practice, one finds that the various factors play a limited role throughout. For example:

A process usually will be able to tolerate a slight reduction in the temperature of its exhaust condensate down to the heat exchanger pinch point, but cannot always tolerate a reduction all the way down to the temperatures implied by a backpressure turbine/generator operating with no additional steam flow.

A condensate-return system might return some—but not all—of the condensate from a given process back to the boiler.

While extreme conditions—complete temperature independence and 0% condensate return—do occur from time to time, this "in-between" state is the most common. As a result, most backpressure turbine/generator installations will approach—but not quite reach—free power.

Example 3: Backpressure Turbine/Generator Analysis in Real Steam Plants

Refer back to the turbine/generator and process specifications used in Example 2, but assume that the process requires a minimum condensate return temperature of 250°F, and that 60% of the condensate is returned to the boiler. If fuel costs $6 per million Btu (based on the fuel's lower heating value, LHV), what is the impact of these constraints on the cost of power? How does this differ from the cost of power as calculated from the more conservative—but less accurate—boiler × generator efficiency algorithm?

First, we can calculate that at 250°F, 38-psig water has an enthalpy of 218.6 Btu/lb. Our process still requires 28.404 × 10^6 Btu/hr, and our enthalpy at the turbine exhaust is still 1,142.3 Btu/lb, so we can re-solve for m_{new} as:

$$m_{new} = 28.404/(1{,}142.3 - 218.6)$$
$$m_{new} = 30{,}750 \text{ lb/hr}$$

Compared to the 31,961 lb/hr we required without condensate return, this implies that we now require only:

$$(30{,}750 \text{ lb/hr} - 30{,}000 \text{ lb/hr})/(31{,}961 \text{ lb/hr} - 30{,}000 \text{ lb/hr})$$

or 38% as much steam production—and hence fuel consumption—in the boiler to satisfy the process thermal loads. Of the remaining 62%, only 60% is actually recovered as condensate, implying that we are only penalized for 60% × 62% = 37.2% of our total thermal penalty as a result of lower condensate return temperatures at the boiler. Thus, we must pay for (38% + 37.2%) = 75.2% of the heat that we remove from the steam. The remaining 24.8% is free!

The enthalpy removed by the turbine as torque is simply the enthalpy drop of the steam, multiplied by the mass flow rate. Incorporating a conversion factor and a 95% efficient generator, we can calculate the turbine power output to be:

$$30{,}750 \text{ lb/hr} \times (1{,}200.4 - 1{,}142.3 \text{ Btu/lb}) \times$$
$$1 \text{ kWh}/3413 \text{ Btu} \times 96\% = 502 \text{ kWe}$$

At 100% condensate recovery and/or a lack of tolerance for lower temperatures in the process, our efficiency would equal boiler × generator efficiency. Assuming an 80% efficient boiler, we can calculate the marginal fuel consumption as:

Fuel consumption = Power produced/Efficiency × Marginal heat penalty

"Marginal heat penalty" is the number we've already calculated—75.2%. Thus:

$$\text{Fuel consumption} = 502 \text{ kW}/(80\% \times 96\%) \times 75.2\% = 492 \text{ kW}_{\text{fuel}}$$

From this calculation, the effective power generation efficiency is (502/492) = 102%! It looks like we're violating thermodynamics, but we're not—we're just not penalizing ourselves for heat that would have otherwise been thrown away.*

Thus, at $6 per million Btu fuel, our cost of power is:

$$(\$6/10^6 \text{ Btu})/102\% \times 3413 \text{ Btu/kWh} = 2 \text{ cents/kWh}$$

By comparison, the more conservative calculation would have resulted in:

$$(\$6/10^6 \text{ Btu})/(80\% \times 95\%) \times 3413 \text{ Btu/kWh} = 2.7 \text{ cents/kWh}$$

Over a 7,000-hour operating year, we've just increased the savings—which will be calculated by someone considering the economics of a

---

*Note that this is still assuming average boiler efficiency—the numbers are even better if we include marginal boiler calculations. We have left them out of this analysis only because in practice, they are extraordinarily difficult numbers to come by.

backpressure steam turbine/generator—by:

0.7 cents/kWh × 497 kW × 7,000 hours/year = $24,353

Notice that we haven't changed the fundamental economics of power generation—we've just done a more accurate job of calculating it. In other words, the savings calculated by this method are closer to the savings that will actually be realized post-installation.

The following two figures illustrate this before and after condition graphically. Note that in these figures, we have shown the entire steam plant rather than simply the pressure reduction station and process.

PART V: GENERAL RULES OF THUMB

1. As observed in the previous section, this logic does not apply to every scenario. However, we can apply some broad "rules of thumb":

The efficiency enhancements implied by this math are only present when there is less than 100% condensate return. In a completely closed steam network, nothing is free, with the possible exception of…

2. In geographically large steam plants, where lengthy and/or poorly insulated condensate return lines upstream of the boiler imply that the net impact of lower temperature condensate at the process may be negligible, the lower temperatures correspond with lower radiative heat losses throughout the condensate return network.

3. The potential for lower temperature condensate from processes is enhanced in those processes where a PRV is supplying superheated steam to process heat exchangers (e.g., there is either no desuperheater, or a very short pipe run between the PRV and the process heat exchangers). The relatively poor heat transfer coefficient of superheated steam means that such processes will invariably be able to extract greater amounts of heat from saturated steam flows, such as are present at the exhaust of a backpressure steam turbine/generator.

4. For similar reasons, processes that are clearly using unnecessarily high pressure steam are more likely to be amenable to condensate cooling, since they probably already have larger-than-necessary DTs in their heat exchangers. (For example, facilities using >30-psig steam for space heating, or > 50-psig steam for lumber drying.)

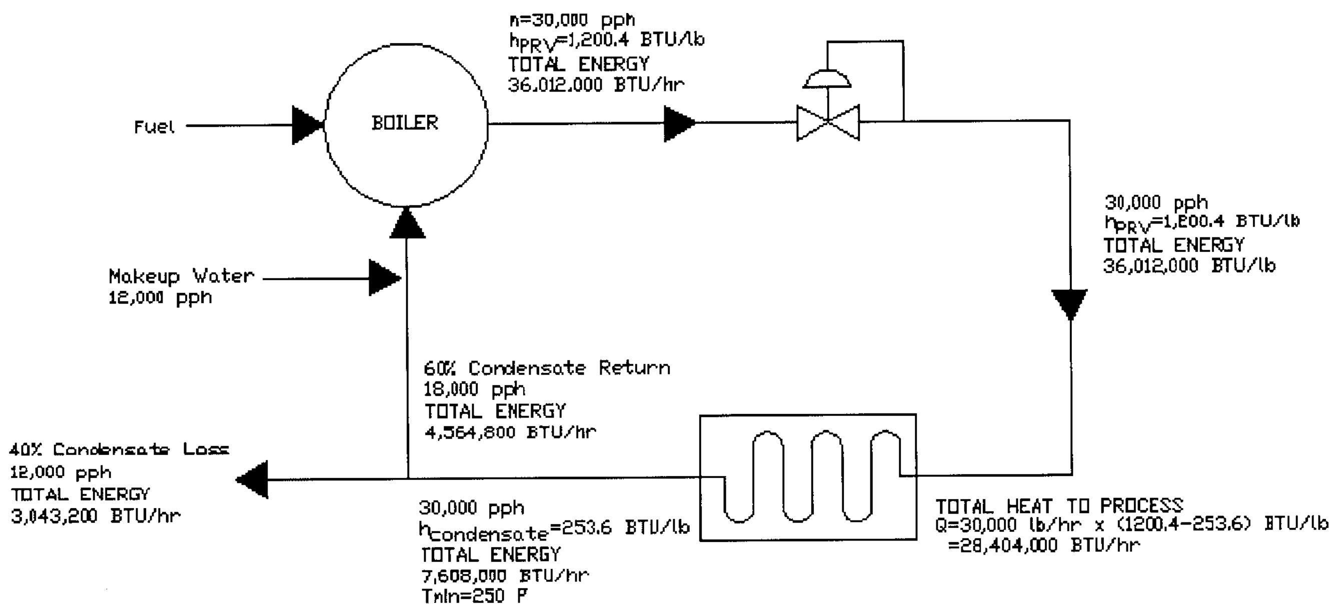

Figure 5. Steam Plant With <100% Condensate Recovery Before Turbine/Generator Installation

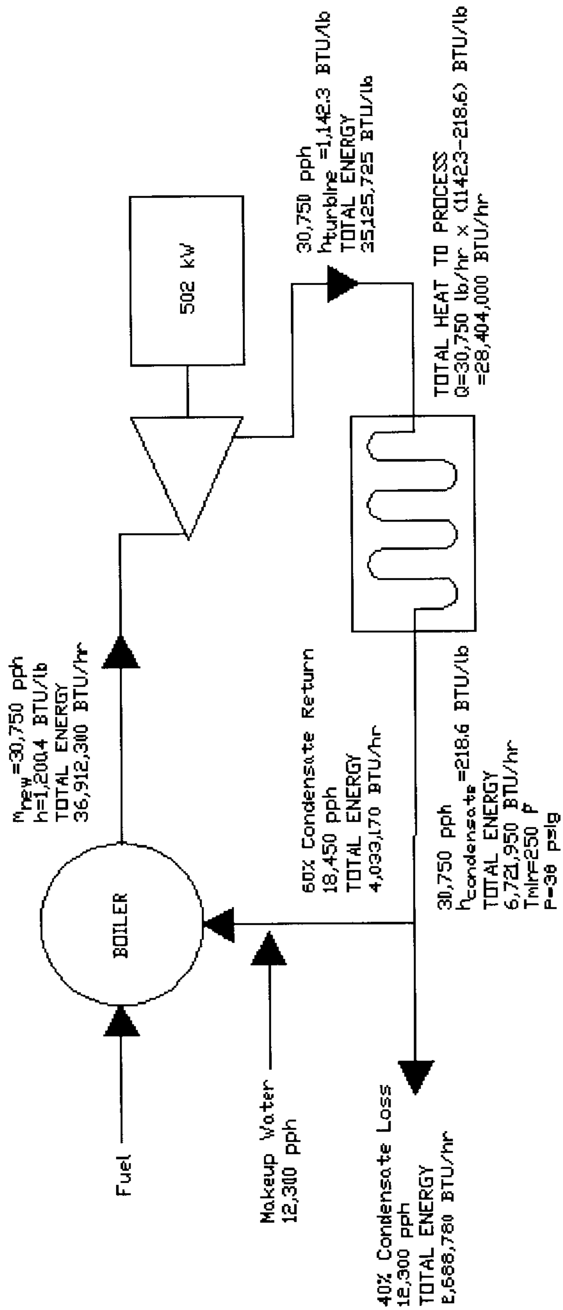

Figure 6. Steam Plant with <100% Condensate Recovery After Turbine/Generator Installation

ABOUT THE AUTHORS

Sean Casten is the president and CEO of Turbosteam Corporation, a designer, packager, installer and financier of steam turbine/generators. Since 1986, Turbosteam has placed 154 systems into service in 18 countries worldwide. Prior to joining Turbosteam in 2000, Sean spent three years at the consulting firm Arthur D. Little, where he focused on the emerging distributed power generation sector. Sean has a B.A. in molecular biology, an M.S. in biochemical engineering and a Masters of Engineering Management. Sean Casten may be contacted via e-mail at *scasten@turbosteam.com.*

Tom O'Brien is president of Integrated Process Solutions, an international consulting firm specializing in energy solutions. The company's focus is on both supply and demand solutions for its clients. Tom has over 25 years of experience, starting his career with E.I. Dupont, Quantum Chemical, and Kerr McGee before entering into consulting. Tom has B.S. degree in chemical engineering and an MBA. Tom O'Brien may be contacted via e-mail at *thomas.p.obrien@worldnet.att.net*.