Sliding Pressure Analysis

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by Gerald Weber
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Introduction

Commonwealth Edison Company is an investor owned utility based in Chicago, Illinois. We service approximately 3.2 million customers in Northern Illinois with about 22,500 MW's of available capacity. This capacity includes power from 12 nuclear units, 24 fossil units and 68 fast start peaking units.

Many of our power plants, especially those which are coal or oil fired, are cycling more than they did ten or twenty years ago. Many of the units are not operated at full load for long periods of time anymore. They are brought down to minimum load at night when power demand is low, then up to higher loads during the day when demand is high.

This paper will utilize the first and second laws of thermodynamics to show an optimum method of operating the control valves on a unit which is frequently cycled. Optimizing the first law means minimizing heat rate. Optimizing the 2nd law means minimizing entropy generation. If we want to minimize the entropy generation across the control valves we must minimize the pressure drop across them.

We must keep in mind however that optimizing the 2nd law efficiency does not mean we optimized the system. The proof that we have truly optimized the system is minimizing the fuel input at the same load. This correlates to immediate dollar savings to the utility. This is measured by 1st law efficiency or heat rate. Therefore we will use the second law as a tool to help us optimize the heat rate. An analytical method of reducing the pressure drop across the valves by lowering or sliding the inlet pressure will be discussed. The same analysis will also be performed by PEPSE to find an optimum method of control valve operation. The PEPSE input parameters for sliding pressure will also be discussed.

Analysis

Before we can discuss the PEPSE application of sliding pressure we need to understand why it is needed. This understanding centers around the second law. Begin by applying the fundamental equations across a valve which has steam flowing through it:

(a)-------------|><|----------------- (b)

Assume: Steady state

Negligible heat loss (adiabatic)

Negligible body forces

Velocity only in x direction

The continuity equation shows that $\dot{m}a = \dot{m}b$ and the energy equation yields $h_a = h_b$ [7]

where: $\dot{m}$ = mass flow rate

h = enthalpy

Using the canonical relations: $$\frac{dh}{dx} = \frac{Tds}{dx} + \frac{1}{\rho}\frac{dp}{dx} = 0 \qquad (1)$$ [7]

and the second law: Entropy Generation = $$\dot{S}gen = \frac{ds}{dx} \qquad (2)$$

We can substitute (2) into (1) to obtain:

$$0 = T\dot{S}gen + \frac{1}{\rho}\frac{dp}{dx} \rightarrow \boxed{\dot{S}gen = \frac{-1}{\rho T}\frac{dp}{dx}} \qquad (3)$$

This is a very important relation because it shows that the entropy generation across the valve is directly proportional to the pressure drop across it.

This equation was checked using PEPSE and was found to agree exactly with the analytical values.

Apply the Bernoulli equation across the valve:

$$\frac{P_1}{\rho_1} + \frac{v_1^2}{2} = \frac{P_2}{\rho_2} + \frac{v_2^2}{2} + \frac{kv_1^2}{2} \rightarrow \frac{P_2}{\rho_2} - \frac{P_1}{\rho_1} = \frac{-kv_1^2}{2} \qquad [6]$$

$$\frac{P_2\rho_1}{\rho_2\rho_1} - \frac{P_1\rho_2}{\rho_2\rho_1} = \frac{-kv_1^2}{2} \rightarrow \frac{P_2\rho_1 - P_1\rho_2}{\rho_2\rho_1} = \frac{-kv_1^2}{2}$$

$$P_2\rho_1 - P_1\rho_2 = \frac{-kv_1^2}{2}\rho_2\rho_1 \rightarrow P_2 - \frac{P_1\rho_2}{\rho_1} = \frac{-kv_1^2}{2}\rho_2$$

$$(P_2 - P_1) + P_1 - \frac{P_1\rho_2}{\rho_1} = \frac{-kv_1^2\rho_2}{2}$$

$$-dp = \frac{kv_1^2\rho_2}{2} + P_1\left(1 - \frac{\rho_2}{\rho_1}\right) \qquad (4)$$

This is another important relation because it shows that the pressure drop across the valve is directly related to the k value of the valve as well as the inlet pressure.

Substituting (4) into (3)

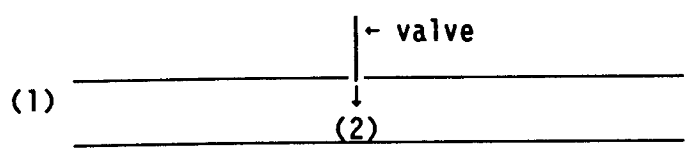
$$\dot{S}gen = \frac{1}{\rho_{avg}T_{avg}}\left(\frac{kv_1^2\rho_2}{2} + P_1\left(1 - \frac{\rho_2}{\rho_1}\right)\right) \qquad (5)$$

Where $\rho_{avg} = \frac{\rho_1 + \rho_2}{2}$ $\qquad T_{avg} = \frac{T_1 + T_2}{2}$

This relation which combines the two preceding relations shows that entropy generation across the valves is directly proportional to the k factor for the valve and the inlet pressure. The k factor depends on valve opening. It increases as the valve is closed.

Now we know what causes the entropy generation across the valves. We must reduce the k value of the valve as much as possible which implies keeping the valve as far open as possible. We can also reduce the inlet pressure to the valves. This will further reduce the entropy generation. If both of these processes can be performed to their maximum together, we have found an optimum point at which to operate as far as the second law is concerned.

The pressure reduction process centers on the analysis of flow through a valve. If we consider the energy equation from point (1) which is the valve inlet, to point (2), which is the minimum point of flow constriction: [4]



$$\cancel{\frac{V_1^2}{2g}} + P_1 v_1 + C_v T_1 = \frac{V_2^2}{2g} + P_2 v_2 + C_v T_2 \rightarrow \frac{V_2^2}{2g} = P_1 v_1 - P_2 v_2 + C_v (T_1 - T_2)$$

negl.

Assuming Ideal gas behavior:

$$\frac{V_2^2}{2g} = P_1v_1 - P_2v_2 + \frac{C_v}{R}(P_1v_1 - P_2v_2)$$

Since: $R = C_p - C_v$ and $\frac{C_p}{Cv} = \delta$

$$\frac{V_2^2}{2g} = P_1v_1 - P_2v_2 + \frac{P_1v_1 - P_2v_2}{\delta - 1}$$

$$\frac{V_2^2}{2g} = P_1v_1 \left[1 + \frac{1}{\delta-1} \right] - P_2v_2 \left[1 + \frac{1}{\delta-1} \right] \qquad (8)$$

Using pv^{δ} = constant for an isentropic adiabatic expansion:

$$P_2V_2 = P_1v_1 \left(\frac{P_2}{P_1}\right)^{\frac{\delta-1}{\delta}} \qquad (9)$$

Substituting:

$$V_2 = \left[2_g (P_1v_1) \frac{\delta}{\delta-1} \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\delta-1}{\delta}} \right] \right]^{\frac{1}{2}} \qquad (10)$$

From continuity:

$$\dot{m} = \rho VA = \frac{VA}{v} \qquad v_2 = v_1 \left(\frac{P_1}{P_2} \right)^{\frac{1}{\delta}} \qquad (11)$$

Substituting (11) into (10):

The isentropic flow rate is given by

$$\dot{m}_s = A \left[\frac{2g\delta}{\delta-1} \left(\frac{P_1}{v_1}\right) \left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\delta}} - \left(\frac{P_2}{P_1}\right)^{\frac{\delta+1}{\delta}} \right] \right]^{\frac{1}{2}}$$

The actual flow rate is given by $\dot{m}s \times Cq$, where Cq accounts for the irreversibility. Therefore:

$$\dot{m} = C_q A_n \left[\frac{2g\delta}{\delta-1} \, \frac{P_1}{v_1} \left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\delta}} - \left(\frac{P_2}{P_1}\right)^{\frac{\delta+1}{\delta}} \right] \right]^{\frac{1}{2}} \qquad (12)$$

If valve position stays constant $\frac{P_2}{P_1}$, δ, and A stay constant. Therefore for constant valve position the flow rate is given by:

$$\dot{m} = C \left[\frac{P_1}{v_1} \right]^{\frac{1}{2}} \quad [3] \quad [4] \qquad (13)$$

$$\text{where } C = Cq\, A \left[\frac{2g\delta}{\delta-1} \left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\delta}} - \left(\frac{P_2}{P_1}\right)^{\frac{\delta+1}{\delta}} \right] \right]^{\frac{1}{2}} \qquad \text{(from equation 12)}$$

This relation is very important because it relates the mass flow rate through the valve with its inlet pressure and specific volume relative to a constant which is common to a fixed valve position. Therefore if C is calculated at one pressure and mass flow rate, it is constant for all pressures and flow rates provided that the valve position stays fixed. This allows the mass flow rate to reduce as the pressure is reduced.

Example: Consider the following initial conditions

$\dot{m} = 5.77 \times 10^6 \ \frac{lb}{hr}$ $\quad P_1 = 2400$ psia $\quad T_1 = 1000$ °F $\quad h = 1460.9$ Btu/lbm

$V_1 = .3214 \ \frac{Ft^3}{lbm}$ $\quad \rho_1 = 3.111 \ \frac{lbm}{Ft^3}$ [1]

Let's calculate these values as we gradually reduce the pressure from 2000 psia to 1400 psia.

First we must calculate C:

$$C = \frac{\dot{m}}{\left[\frac{P1}{v1}\right]^{\frac{1}{2}}} = \frac{5.77 \times 10^6}{\left[\frac{2400 \times 144}{.3214}\right]^{\frac{1}{2}}} = 5564.32$$

This value of C stays constant as the pressure is reduced.

Now lets reduce the pressure to 2300 psia while holding the temperature fixed at 1000 °F. At these values:

h = 1464.2 v = .3372 [1]

$$\dot{m} = C\left[\frac{P1}{v_1}\right]^{\frac{1}{2}} = 5564.32\left[\frac{(2300)(144)}{.3372}\right]^{\frac{1}{2}} = 5.51 \times 10^6 \frac{\text{lbm}}{\text{hr}}$$

The rest of the values in the pressure reducing process are easily calculated and are shown in the table below:

TABLE 1

Sliding Pressure Calculations

Press. (psia)	Temp. (°F)	Enthalpy Btu/lbm)	Specific Volume (ft^3/lbm)	Mass Flow Rate (lbm/hr)
2400	1000	1460.9	.3214	5.77×10^6
2300	1000	1464.2	.3372	5.51×10^6
2200	1000	1467.6	.3545	5.26×10^6
2100	1000	1470.9	.3734	5.01×10^6
2000	1000	1474.1	.3942	4.76×10^6
1900	1000	1477.4	.4171	4.51×10^6

From the table we see that as the pressure has been reduced 500 psia, or 20%, the mass flow rate has been reduced 1.26×10^6 lb/hr or 22%. Therefore instead of closing the valve by 20% to reduce the mass flow, we have reduced the inlet pressure to the valve while holding its position constant.

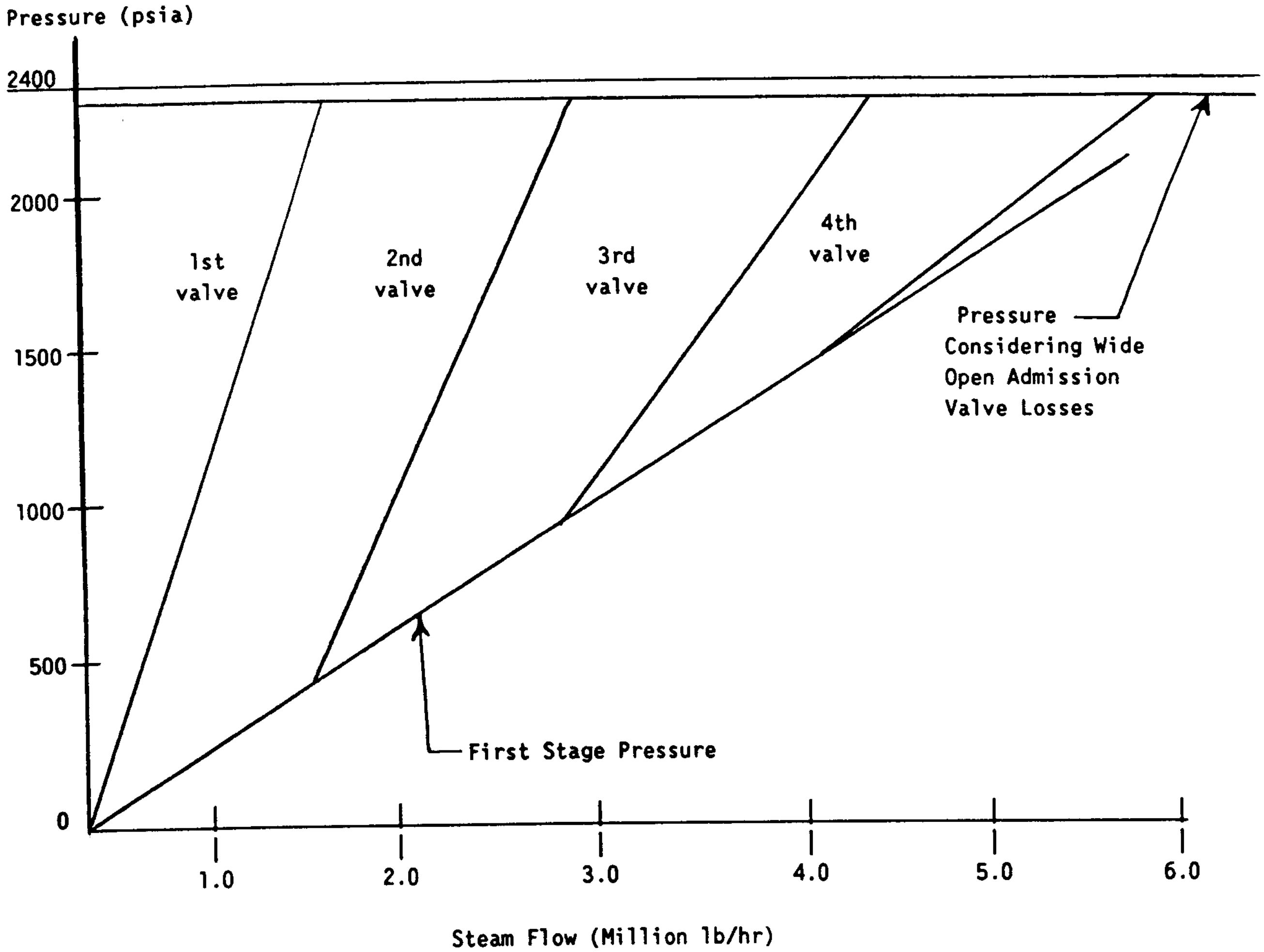
The mass flow rate calculated by PEPSE was found to agree with the analytical value.

Table 1 is a simple one valve system which served as an acceptable way to illustrate the calculation process of sliding pressure. However in reality the system is more complicated because there are often anywhere from 4 to 8 control valves in a modern steam unit governing system.

To analyze this system accurately we must consider the pressure drop across each individual valve as a function of first stage pressure. First stage pressure increases with throttle flow as approximated in Figure 3.

Figure 3

First Stage Pressure Vs. Steam Flow [5]

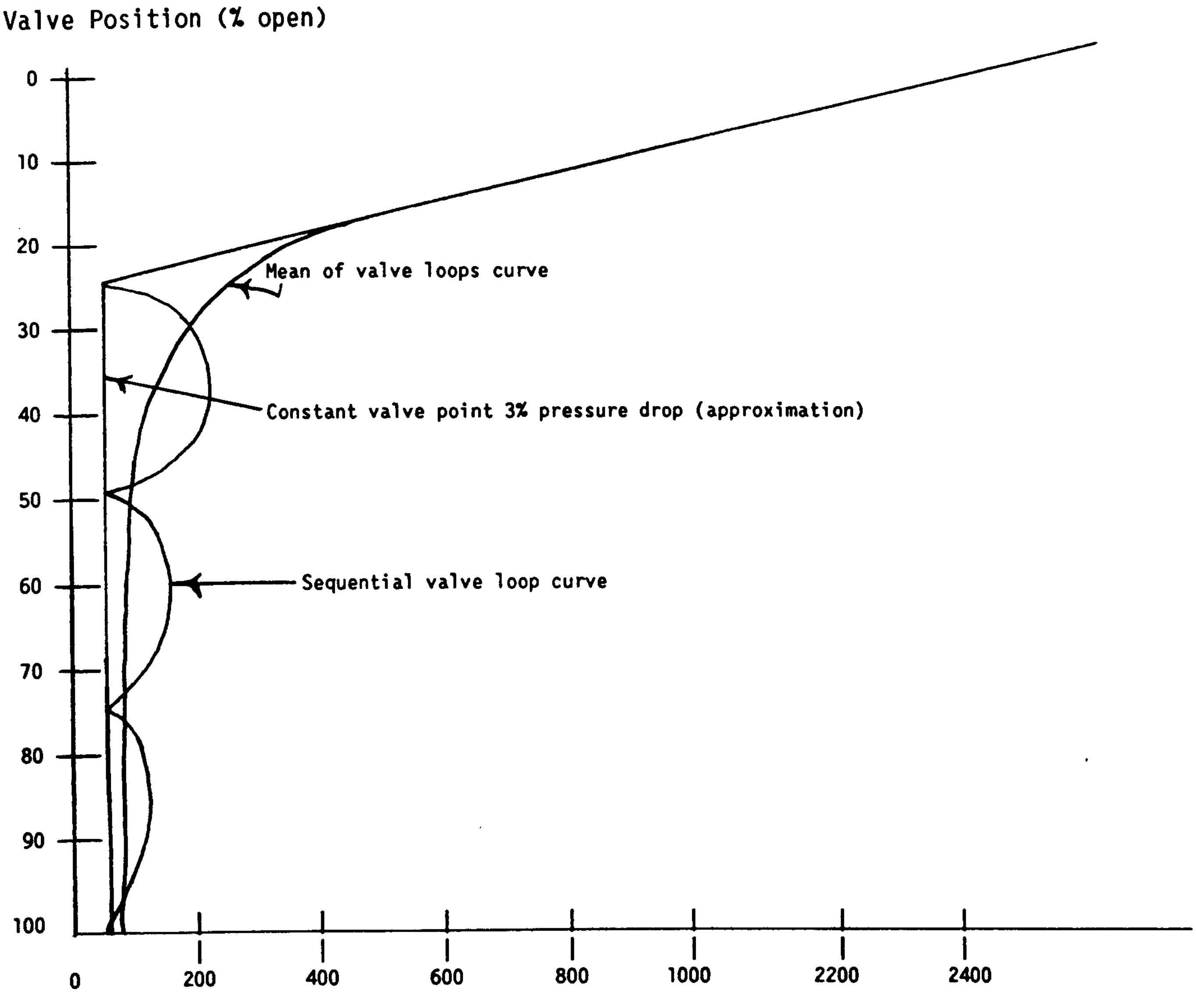


Note: Graph assumes choked valves and no valve overlap.

As we see in Figure 3, which was obtained by PEPSE, the first stage turbine pressure increases linearly with steam flow. This illustrates that we must consider each valve separately since we cannot assume a "generic" single valve pressure drop curve. The minimum pressure at the valve outlet is first stage pressure. Thus, a smaller pressure drop exists for the higher valves than for the lower valves. Therefore each valve will have its own pressure drop curve associated with it.

Using the valve pressure drops in Figure 3 and approximating the valve system outlet pressure by weight averaging each individual valve outlet pressure, we can calculate a valve loop curve. It is constructed by closing one valve at a time in a four sequential valve system. The loops increase in size as valve position decreases because there is a greater pressure drop across them as first stage pressure decreases. The curve which best represents the average of all points is shown and called the mean of valve loops curve. When the valve position is kept constant at a valve point the pressure drop stays constant at 3 per cent assuming that they are choked at all valve points. This assumption is an approximation because actually there is a slightly greater pressure drop at lower valve points.

PRESSURE DROP CURVE FOR A FOUR VALVE SYSTEM



Note: This is an approximation curve based on weight averaging choked valve pressure drops.

Figure 5

PEPSE Analysis

To slide the pressure using PEPSE, use Special Option Number 1. This option uses equation 13 to calculate the mass flow rate and valve position. In using Special Option 1 there are 6 important inputs:

1. Whether or not the mean of valve loops curve is considered. Mean of valve loops is shown in Figure 5. If mean of valve loops is considered there will be an HP Section efficiency loss. This incorporates the G. E. procedures for fossil turbines. PEPSE considers the pressure drop across the control valves to be 3 percent until only one valve is open. Once the last valve begins to close, the pressure drop across it is taken into account with an iterative equation. This is slightly different than the simpler theoretical analysis discussed previously in that not all the valve losses are considered as valve pressure drop. When sliding pressure or operating at any valve point, mean of valve loops should not be used.

2. The minimum throttle flow ratio. This input tells PEPSE when to start considering more than a 3% pressure drop across the control valves. A pressure drop greater than 3% occurs below the last valve point because the throttling effect starts there and increases until all valves are closed. If parallel valve operation is chosen, this input equals 1.0 because there will be immediate throttling losses.

3. The number of control valves. This input works with the minimum first stage flow coefficient to calculate the control valve pressure drop.

4. The valve point the unit is sliding from. There are as many valve points as there are sequential control valves. The lowest point at which the unit can operate for an extended period of time depends on the boiler.

5. The Boiler Feed Pump outlet pressure. This pressure must be reduced by the amount of the slide in pressure to consider the pumping power saved.

6. The Boiler Feed Pump Type. If the pump is motor, steam, or shaft driven. This will become a factor in the aux power saved by sliding.

Over the years, there has been extensive research done by Westinghouse and other turbine manufacturers on deciding what valve point to start sliding pressure instead of throttling. When sliding pressure, the HP turbine section efficiency and pressure ratio stay relatively constant because the valve position does not change. This constrains the HP turbine outlet conditions to a pressure and temperature and corresponding enthalpy. This could cause the HP turbine enthalpy drop to be less with sliding pressure than with sequential valve operation. This will increase the mass flow rate for the sliding pressure case to achieve the same load which will require more heat to be supplied by the boiler and correspond to a heat rate increase. Therefore even though the second law efficiency is optimized, in some cases the first law is not.

This could occur at the highest valve points in units operating with sequential control valves. However, as the load is reduced the sequential operation throttling losses increase and cause an HP turbine section efficiency reduction. The lower efficiency will reduce the HP section enthalpy drop just as sliding pressure does. Therefore, a point exists where the unit heat rate with sliding pressure equals the unit heat rate with sequential valve operation. At loads below this point, it is more beneficial to slide pressure. This point is found by running PEPSE for both cases as the load is reduced. On an eight valve machine Westinghouse advocates sliding pressure below the 50% admission point (See references).

Another benefit in favor of sliding pressure is that since the HP turbine is more efficient, the cold reheat temperature is greater than with sequential operation. This will reduce the heat addition necessary in the reheater and lower the heat rate. The lower reheat addition when sliding is especially important if the unit cannot make design hot reheat temperature at low loads. Sliding pressure will bring it closer to design.

The three examples that follow illustrate this discussion. The examples compare sliding pressure operation and sequential valve operation at three valve positions. These three positions correspond to maximum frictional losses for sequential operation. This occurs when a valve is 1/2 open.

Examples

1. Consider the 850 MW unit discussed previously. This unit has a 4 sequential control valve system, steam driven boiler feed pumps, and a 2400 psia boiler feed pump outlet pressure. It is desired to lower the unit generation to 750 MW. At this point the 4th control valve will be nearly 1/2 open for sequential valve operation. Using PEPSE, the pressure was reduced to 2119 psia to correspond to 750 MW. The following table compares the results for sliding pressure vs. sequential valve operation.

EXAMPLE 1

Sliding Pressure From Fourth Valve Point

	Sequential Valve	Sliding Pressure
Generation (MW)	750	750
Mass Flow Rate (lb/hr)	4.98×10^6	5.03×10^6
G. S. Efficiency	75.1	83.1
H. P. Efficiency	86.6	86.9
H. P. Section Efficiency	84.7	86.8
G. S. Press. Ratio	1.48	1.27
H. P. Press. Ratio	3.18	3.24
H. P. Section Press. Ratio	4.72	4.11
G. S. ΔH (Btu/lb)	40.4	27.6
H. P. ΔH (Btu/lb)	118.2	124.9
Total H. P. ΔH (Btu/lb)	158.6	152.5
Cold RH Temp (°F)	604.9	631.7
Hot RH Temp (°F)	1000.0	1000.0
Reheat ΔH (Btu/lb)	219.5	203.5
BFPT Power (MW)	8.9	8.2
ΔP Across Valves (Psia)	73	64
Gross Heat Rate (Btu/kwh)	8052	8086

Notice that even though sequential valve operation H. P. section efficiency is about 2% lower and the BFP Turbines consume about 1 MW more power, and 16 Btu/lb less heat is added by reheater than for the sliding pressure case, this is not enough to make this operation undesirable at this valve position. The sliding pressure total HP turbine enthalpy drop is 6 Btu/lb lower than for sequential valve operation. This requires a greater steam mass flow rate and causes a 34 Btu/kwh higher heat rate for the same load.

Examples

2. The same unit in Example 1 is now operating at the third valve point at 620 MW. It is desired to reduce generation to 520 MW which is near the 1/2 open position of the third control valve. Using PEPSE, the pressure was reduced to 1998 psia to correspond to 520 MW at the third valve point. The following table compares the results for sliding pressure vs. sequential valve operation.

EXAMPLE 2

Sliding Pressure From Third Valve Point

	Sequential Valve	Sliding Pressure
Generation (MW)	520	520
Mass Flow Rate (lb/hr)	3.43×10^6	3.39×10^6
G. S. Efficiency (%)	56.7	67.4
H. P. Efficiency (%)	87.0	87.4
H. P. Section Efficiency	76.9	82.2
G. S. Press. Ratio	2.20	1.81
H. P. Press. Ratio	3.08	3.13
H. P. Section Press. Ratio	6.77	5.65
G. S. ΔH (Btu/lb)	58.5	54.6
H. P. ΔH (Btu/lb)	112.6	118.0
Total H P ΔH (Btu/lb)	171.1	172.6
Cold RH Temp (°F)	561.7	582.6
Hot RH Temp (°F)	1000.0	1000.0
Reheat ΔH (Btu/16)	236.1	223.8
BFPT Power (MW)	6.4	4.8
ΔP Across Valves (Psia)	73	60
Gross Heat Rate (Btu/kwh)	8361	8311

The results indicate that at this valve position it would be more beneficial to slide pressure by 50 Btu/kwh. The frictional losses, extra BFP turbine power and reheat heat addition required, are enough to make sequential valve operation undesirable at this position. Notice the 5.3% HP Section efficiency difference and the 12.3 Btu/lb reheater heat addition difference between the 2 cases.

Examples

3. The 850 MW unit now is running at the second valve point at 400 MW and it is desired to drop load to 300 MW. The 300 MW corresponds to the second valve 1/2 open position for sequential valve operation. Using PEPSE the pressure was reduced to 1792 psia to correspond to the 300 MW generation at the 2nd valve point. The following table compares the sliding pressure results with sequential valve operation at this position.

EXAMPLE 3

Sliding Pressure From Second Valve Point

	Sequential Valve	Sliding Pressure
Generation (MW)	300	300
Mass Flow Rate (lb/hr)	2.05×10^6	2.02×10^6
G. S. Efficiency (%)	43.7	52.7
H. P. Efficiency (%)	87.4	87.7
H. P. Section Efficiency (%)	67.5	74.1
G. S. Press. Ratio	3.72	2.74
H. P. Press. Ratio	3.01	3.06
H. P. Section Press. Ratio	11.18	8.41
G. S. ΔH (Btu/lb)	71.0	70.3
H. P. ΔH (Btu/lb)	109.1	115.2
Total H P ΔH (Btu/lb)	180.1	185.5
Cold RH Temp. (°F)	522.9	550.0
Hot RH Temp. (°F)	1000.0	1000.0
Reheater ΔH (Btu/lb)	248.7	233.8
BFPT Power (MW)	3.23	2.51
ΔP Across Valves (Psia)	73	54
Gross Heat Rate (Btu/kwh)	9083	9008

This results indicates a stronger need to slide pressure than example 2 does. It is more beneficial to slide in this example by 75 Btu/kwh. This is a reinforcement of the Westinghouse theory of sliding pressure below the 50% admission point on a sequential valve unit.

Conclusion

This paper has shown the analytical aspects of sliding pressure as well as how to analyze it using PEPSE. It has illustrated the use of special option Number 1 as well as the important input parameters to consider while using it. The paper has also shown that sliding pressure is not always a good idea. It may be better to throttle at high valve points depending on the unit design. A PEPSE analysis will determine the point at which to initiate sliding.

An important question when considering sliding pressure is: How low can the pressure be reduced? The answer to this question is unit specific. Some units were designed to slide pressure while others were not. Other units can only slide pressure a certain amount. This will determine along with a PEPSE analysis, the best valve point to initiate sliding. It is best to consult with both the boiler and turbine manufacturers before considering sliding pressure operation.

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