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PROCEEDINGS OF
2004 ASME POWER

COMPONENTS, PLANT SYSTEMS, AND DESIGN ENGINEERING

OPERATIONS, MAINTENANCE, RELIABILITY, AVAILABILITY, AND MAINTAINABILITY

COMBINED CYCLES, COMBUSTION TURBINES, STEAM TURBINES, AND GENERATORS

FUELS, COMBUSTION, AND EMISSION ISSUES

RENEWABLE AND ADVANCED ENERGY SYSTEMS

MANAGEMENT ISSUES

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TUBE FAILURE DURING STARTUP IN A STEAM SURFACE CONDENSER INSTALLED IN A COMBINED CYCLE PLANT OPERATING IN COLD CLIMATE

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ABSTRACT

Combined cycle plants in cold climates experience low circulating water inlet temperatures during winter months. Low circulating water inlet temperatures combined with partial bypass steam flow to the condenser results in extremely low condenser pressures and high steam velocities. Improper design, control & operation of desuperheating valve and improper drainage of bypass header lines can lead to pockets of wet steam in the bypass steam. High steam velocities combined with wet steam pockets of varying quality can cause flow-induced vibration and tube failures. This paper examines the performance of a condenser in bypass mode for varying condenser pressures, bypass steam flow rates, support plate spacing, and moisture pockets with varying quality. Actual and critical steam velocities are calculated. Condenser operating points prone to flow-induced vibration and associated tube failures are predicted. Recommendations on safeguards to eliminate flow induced vibration and resulting tube failures are discussed.

INTRODUCTION

The power industry has witnessed the commissioning of numerous large combined cycle plants in the last five to ten years. In a combined cycle plant, the steam surface condenser has to condense the turbine exhaust steam (normal operation) as well as the bypass steam (steam turbine bypass operation). In bypass operation, with only the gas turbines in operation, high-pressure steam from the HRSG is attemperated in a pressure reducing/desuperheating valve and then admitted into the condenser. The total bypass steam flow can be as high as 150-200 % of the design turbine exhaust flow and the duration of bypass operation can vary from a few hours to a few weeks.

Combined cycle plants are put into commercial operation in the beginning of summer. Startup, testing, and debugging activities are usually scheduled in the beginning of the year during winter months. During "DLN" tuning (Dry Low NOX tuning), the gas turbines are operated individually at partial loads leading to partial bypass steam flow to the condenser. Often times the gas turbines are shown down for hours for tuning purposes. In certain instances, circulating water pumps and cooling tower fans are often operated at full capacity even though gas turbines are shut down and bypass steam is not admitted into the condenser. The concept "colder the water, lower the condenser pressure" applies when the steam turbine is in operation. This concept does not apply to bypass operation. In cold climates the circulating water inlet temperatures can approach freezing temperatures. Condenser pressures substantially lower than 1.0 inch HgA have been encountered. Oversized evacuation packages, air tight condenser, clean tubes, excessive safety margin in design can lead to a further decrease in the condenser pressure. Low circulating water inlet temperatures combined with partial bypass steam flow to the condenser can lead to low condenser pressure and high steam velocities, flow-induced vibration and tube failures.

Improper operation of the desuperheating valve can further aggravate the problem. Improper design/operation of the desuperheating valve, improper drainage of the steam header between the desuperheating valve and the condenser can lead to pockets of moisture with varying quality. Low condenser pressures and high steam velocities combined with pockets of wet steam can cause severe damage to condenser tubes.

Numerous combined cycle plants operating in northeast USA have reported tube failures during winter startups. Tube failures have been reported during DLN tuning, partial, and full bypass operation. In certain instances, tube failures have been reported within hours of bypass operation. The tube failures have disrupted the startup schedule and delayed the commissioning of the combined cycle plant.

MECHANISM OF TUBE FAILURE

The tube bundles in a steam surface condenser are susceptible to flow induced vibration from four different sources. These are fluid-elastic instability, vortex shedding, turbulent buffeting, and acoustic resonance. Fluid elastic instability, vortex shedding, and turbulent buffeting are associated with large amplitude of vibration. Vortex shedding, turbulent buffeting, except in rare cases, does not initiate large tube vibration amplitudes in a steam surface condenser. Acoustic resonance results in a loud acoustic noise. Acoustic resonance seldom coincides with the tube vibration itself. As a result the effects of vortex shedding, turbulent buffeting and acoustic resonance are not considered.

With fluid-elastic excitation, the maximum tube displacement increases as the fluid velocity is increased until the critical velocity is reached. Increasing fluid velocity past the critical velocity causes an exponential growth in the tube displacement and subsequent tube failure. Numerous researchers [4-6] have presented formulations to

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calculate the critical velocity. Connor's method is the most widely accepted formulation to evaluate the critical cross flow velocity. In the present analysis the Connor's method will be adopted for calculating the critical cross flow velocity.

NATURE OF ACTUAL TUBE FAILURE

Almost all of the reported tube failures have occurred during start up bypass mode when the steam turbine was not in operation. The tube failures were located randomly along the length and width of the condenser. Tube failures were not clustered in one area. In almost all of the cases, the failure resulted in a clean shear of tube at the center of the unsupported span. On numerous occasions, a cluster of tubes surrounding the failed tube were dented or damaged. The damage to surrounding tubes is attributed to the "whirling effect" of the failed tube.

A PRACTICAL EXAMPLE

The mechanism of tube failure is analyzed using a practical example. It is assumed that the bypass headers and the steam dome are designed such that the bypass steam is evenly distributed over the entire tube bundle. The actual cross flow over the tube bundle is dependent on tube layout, tube pitch, length and width of the shell. To simplify the analysis the cross sectional area is assigned a fixed value.

The condenser pressure is dependent on circulating water flow rate, inlet temperature, tube geometry, cleanliness factor, and a number of other design and flow characteristics. To limit the number of independent variables and simplify the analysis, the condenser pressure is assumed to vary between 0.5 inches HgA to 3.5 inches HgA. The support plate spacing is assumed to vary between 30 to 48 inches. The bypass steam flow is assumed to vary between 250,000 lbs/hr to 1,500,000 lbs/hr. The variables considered in the analysis are as follows:

Dump steam flow, lbs/hr:	250,000 - 1,500,000
Inlet bypass steam enthalpy, Btu/lb:	1200
Flow area at the topmost tube, ft^2:	100
Tube outer diameter, inches:	1.0
Tube thickness, inches:	0.028
Tube material:	304 SS
Support Plate spacing, inches:	30 to 48
Condenser Pressures examined, inches HgA:	0.5 to 3.5

CALCULATIONS

Actual and critical cross flow velocities are calculated for a non-homogeneous mixture of bypass steam containing pockets of wet steam of varying quality. The "actual" steam velocity within the condenser is calculated from the specified bypass steam flow rate, enthalpy, condenser pressure, and assumed cross-sectional area. The velocity of bypass steam is assumed to be uniform over the entire tube bundle. The critical cross-flow velocity is calculated for the "wet steam pocket" with 25%, 50%, 75%, and 100% quality. An additional "superheated" case is considered wherein the bypass steam is dry and does not contain any wet steam. The critical velocity is calculated using the Connor's method. The calculations for the critical cross flow velocity are lengthy and therefore not reproduced in the present paper. The calculations are presented in detail in reference 3 & 4.

The actual velocity, critical cross-flow velocity and the ratio of actual/critical cross flow velocities are calculated for each "wet steam pocket" quality for condensers pressures ranging from 0.5 to 3.5 inches HgA, unsupported tube spans ranging from 30 to 48 inches, and bypass steam flow rate ranging from 250,000 lbs/hr to 1,500,000 lbs/hr. The ratios of Vactual/Vcritical are plotted as a function of condenser pressure, steam flow rate, and unsupported tube span.

RESULTS

The actual cross flow velocity must be lower than the critical cross-flow velocity. To account for uncertainties in empirical formulas, calculations, assumptions, material properties, physical dimensions, and steam flow conditions; it is recommended that the actual cross-flow velocity should not exceed 50% of the critical cross-flow velocity.

A picture of failed tubes is included in Figure 1. The tube is sheared at the center of the unsupported span between two support plates. The failure occurred in bypass mode operation during a winter startup.

Variation of Vactual/Vcritical with condenser pressure for support plate spacing of 36 inch and flow rate of 250,000 lbs/hr, 750,000 lbs/hr, 1,000,000 lbs/hr and 1,500,000 lbs/hr are included in Figures 2, 3, 4 & 5 respectively. Tube failure can be avoided if the condenser is operated such that Vactual/Vcritical is less than 0.5. Tube failures can be expected as the velocity ratio approaches or exceeds 1.0.

For bypass steam flow rate of 250,000 lbs/hr the velocity ratio remains below 0.5 for all operating cases. Low steam flow results in a very low velocity as it is assumed that the steam is evenly distributed over the entire cross-section of the tube bundle. For bypass steam flow rate of 750,000 lbs/hr the velocity ratio exceeds 0.5 when the condenser pressure starts to drop below 1.5 inches HgA. For bypass steam flow rate of 1,000,000 lbs/hr the velocity ratio exceeds 0.5 for condenser pressure below 2.0 inches HgA. For bypass steam flow rate of 1,500,000 lbs/hr the velocity ratio exceeds 0.5 for the entire range of condenser pressures. The velocity ratio is lower than 0.5 for the 100% quality and superheated condition cases.

Figures 5 and 6 illustrate the variation of Vactual/Vcritical for varying bypass steam for a support plate spacing of 36 inches and condenser pressure of 1.0 and 2.0 inches HgA respectively. Higher steam flow rate combined with lower quality of steam in moisture pockets lead to higher velocity ratio. The velocity ratio increases with decreasing condenser pressure.

The effect of support plate spacing on the velocity ratio for bypass steam flow rate of 1,500,000 lbs/hr for condenser pressure of 1.0 and 2.0 inches HgA is included in Figures 8 & 9. As expected, velocity ratio increases with increasing support plate spacing. The velocity ratio increases as the condenser pressure decreases.

RECOMMENDATIONS

Condenser tubes are susceptible damage from flow-induced vibration at lower condenser pressures and from bypass steam containing pockets of wet steam of varying quality. Maintaining the condenser pressure in bypass mode equal to or above 2.0 inches HgA can decrease the incidence of tube failures. Higher condenser pressures can be achieved by increasing the circulating water inlet temperature

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(by varying the number of cooling tower fans in operation), reducing or recirculating the circulating water flow. Precautions must be taken to eliminate wet steam pockets in bypass steam. Performance of the desuperheating valve during maximum and minimum flow rate cases, the control logic, header lengths, and placement of temperature & pressure sensors on the bypass header must be carefully examined to ensure that wet steam is not admitted into the condenser. The bypass header between the desuperheating valve and the steam surface condenser must be drained using a drain pot/gage glass assembly to ensure and confirm proper drainage.

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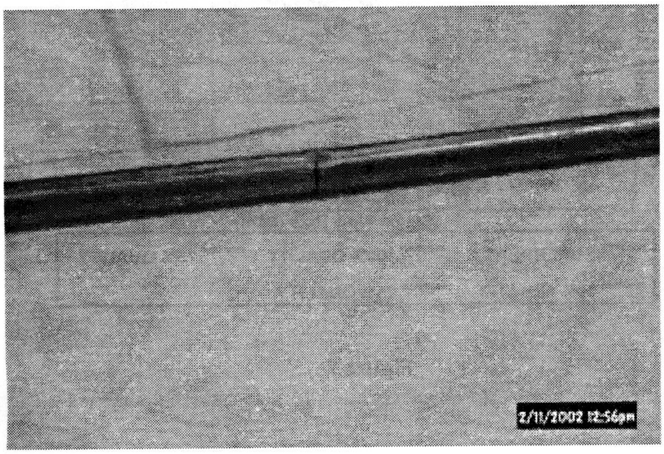

Figure 1: Stainless Steel Tube Sheared at Midspan

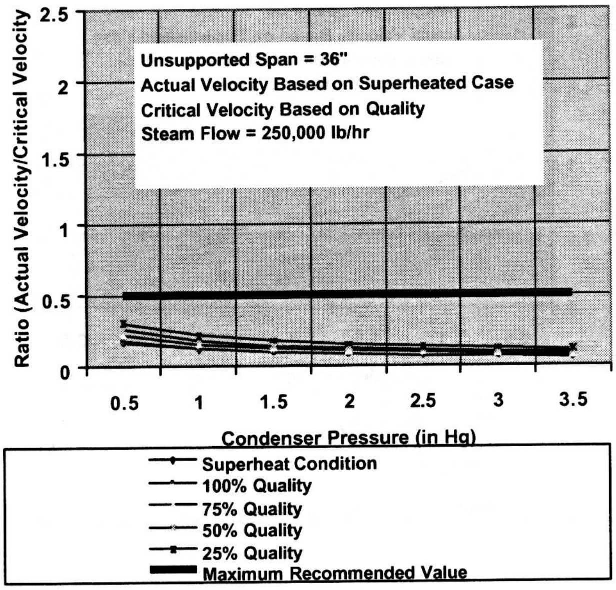

Figure 2

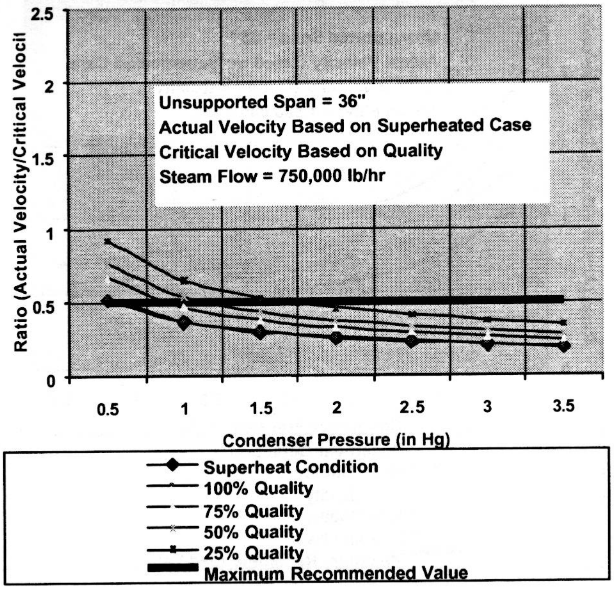

Figure 3

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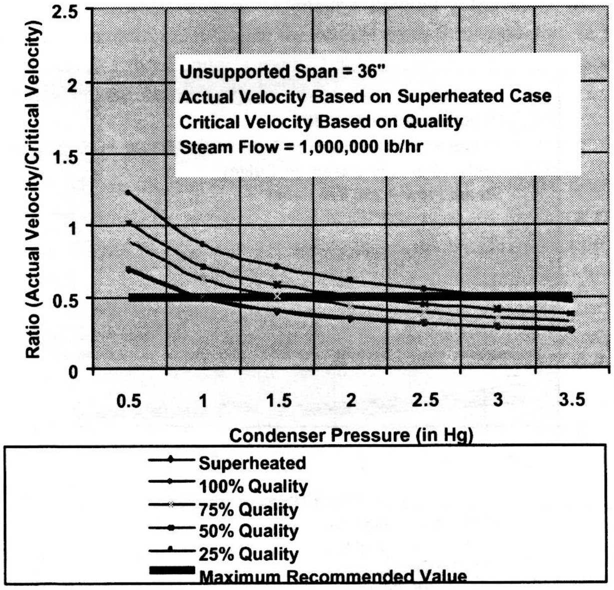

Figure 4

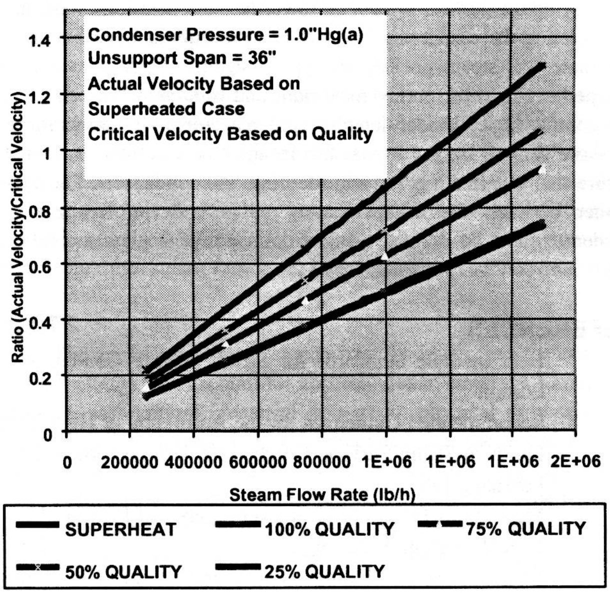

Figure 6

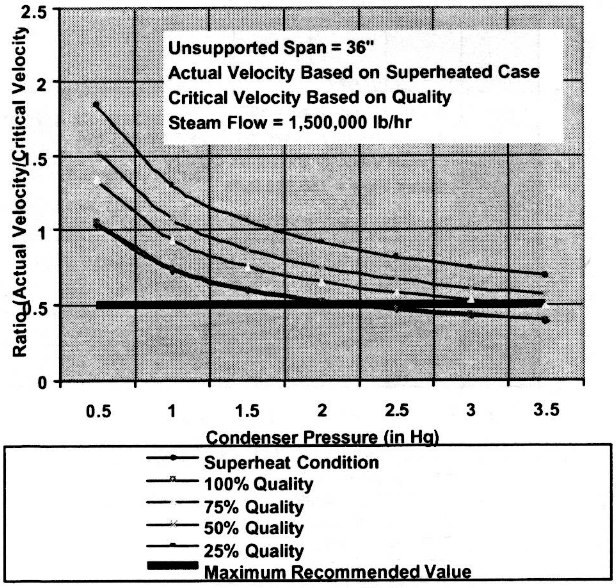

Figure 5

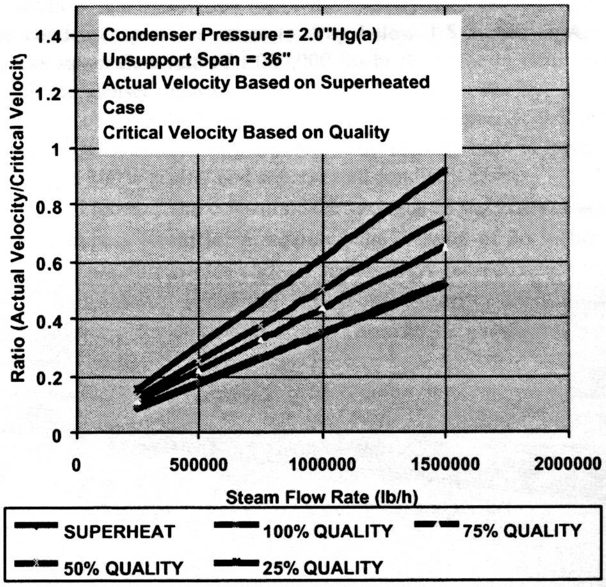

Figure 7

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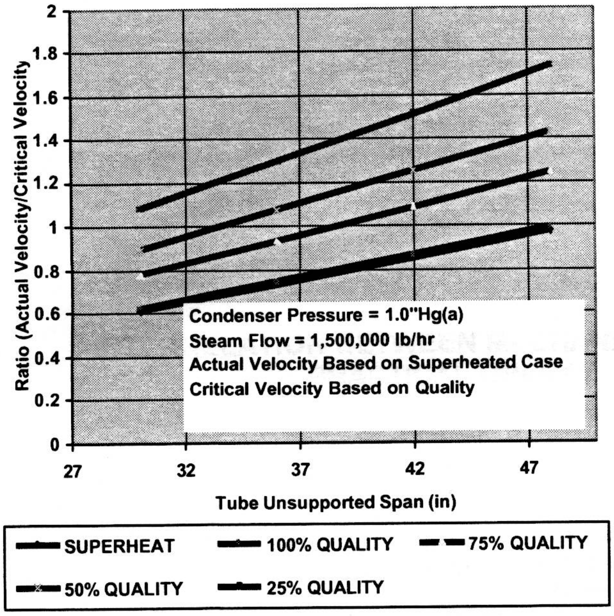

Figure 8

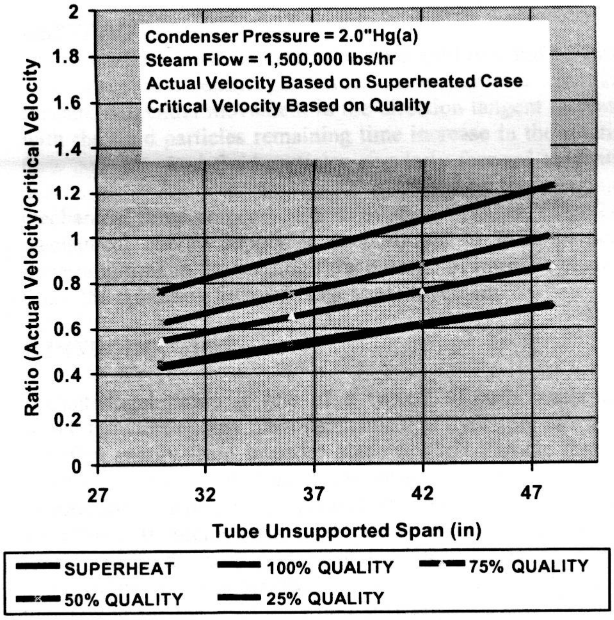

Figure 9

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