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Energy Diagnostic of Condensing Unit in a 160 MW Electric Power Plant

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Abstract

This paper presents the methodology and results of the energy diagnostic of a surface condenser operating in a 50 MW steam turbine of a 160 MW combined cycle. The equipment energy performance was evaluated from various performance indicators presented before and after the implementation of a maintenance program. The main cause was determinate that kept the pressure inside the condenser in a critical operating condition and generate a load loss production in the steam turbine around 3 MW. Additionally, the article presents cause and the methodology used for the determination of the initial reduction in the equipment performance respect to their design conditions. Finally, an additional criterion was carried out with the help of an analytical statistical model for determining the optimal frequency of cleaning under maximum load conditions for the power plant.

Keywords: Surface condenser, optimum cleaning frequency, energy performance, initial reduction performance

1 Introduction

In a thermoelectric combined cycle power plant, the condenser plays an important role maintaining the steam turbine's generation capacity. The vacuum pressure that exists inside the condenser is a variable that allows kept the steam temperature that leaves the turbine as low as possible, so there is a greater conversion of thermal energy to mechanic energy in the turbine. In this way, any pressure variation within this equipment directly affects the turbine generation capacity and consequently the total thermal efficiency of the cycle. Only a 10% decreases in the efficiency of this equipment reduces the load production of a turbine in 1 MW [1].

Moreover, the maintaining of the condenser's integrity is very importance for preventing major damages to others components of the system. For example, the corrosion in the condenser's pipe can lead to failures and leaks through the pipe walls, allowing the circulating water to mix with high purity condensate; so contaminants dissolver in the circulating water can cause serious damage to the boiler pipes and the steam turbine's materials.

In the literature we can find important reports of developments and research associated with the study of the surface condensers [2, 3]. For example, one of the most recent and complete published works [4] focuses mainly on the performance monitoring and optimization of the maintaining works; but this do not take into account other variables that eventually occur in the d

The main contribution of this work, aims the study of some of the most important factors that influence in the performance reduction of surface condenser, which were identified during the analysis of the operational conditions of these equipment. Specifically, we studied the effects of the steam passes and the configuration of the pipes inside the condenser, the condensation pressure and the power delivered by the steam turbine.

2 Methodology

From its first operation day in the year 1995, the surface condenser of the studied power plant, presented operational problems related to obtaining the design vacuum pressure. According to the power plant's normative parameters, this value is 2,44 in HgA; however, the minimum value reached in operation were 2,8 in HgA. This increase caused a loss of power in the steam turbine approximately 1.6 MW. During all this time, the behavior of this equipment has presented an operational variability, which will be object of analysis in this article.

Figure 1 shows the condenser vacuum behavior from its first startup day, including the initial design deviation.

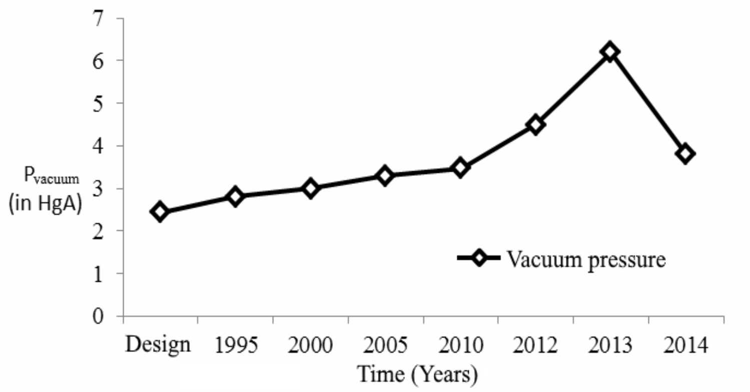

Figure 1. Increase of the vacuum pressure in the equipment

It can notice how the condenser's vacuum pressure has gradually increased over the years. However, from 2012 it is observed appraise a significant increase in the pressure maintained until mid-2013, which generated a load loss in the steam turbine around 3 MW. After a scheduled maintenance shutdown during which a general cleaning of the equipment pipes was made and where some condensate drain valves belonging to the piping system were changed, the condenser vacuum returned to its common performance values and the turbine recovered the normal load it had maintained until a few months ago. The methodology used to calculate the main cause of the pressure increase in 2013 and the origin of the initial deviation will be presented below.

2.1 System description and assumptions

Figure 2 shows a scheme of the power plant's condensation system with some interest components for analysis.

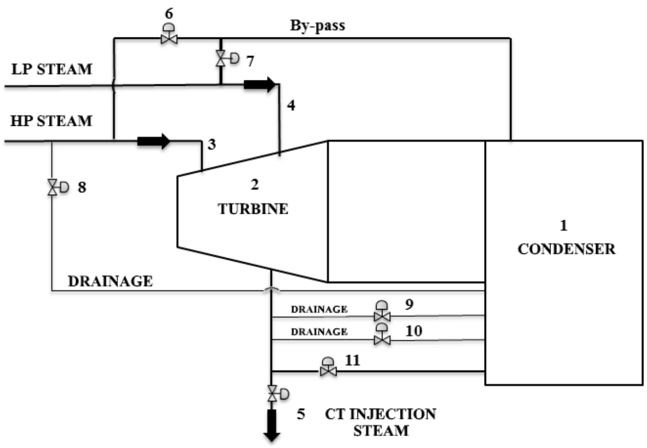

Figure 2. Scheme of steam-turbine system.

The surface condenser (1) is the equipment responsible for receiving the output steam from the turbine (2). The turbine is fed by the steam produced in a two-pressure heat recovery boiler which generates a high pressure steam and an induction or low pressure steam, which is injected into the turbine at points (3) and (4) respectively. The Stream 5, represents a outlet of steam in one of the stages of the turbine, which is sent to the combustion turbine for the control of polluting gases emissions. The system was also designed to supply vapor from the recovery boiler directly to the condenser when the process conditions so require, by means of two bypasses in the control valves shown in (6) and (7) points. In addition, figure 2 show the condensate drain valves of the high pressure pipes readjusted during the power plant shutdown, which are listed with digits (8-11).

Figure 3 shows a surface condenser's scheme where the main input and output streams are located. Through stream 1 the steam from the turbine enters the condenser and after its condensation on the pipes surface it accumulates in the hot pit, where it is them pumped out through stream 2. States 3 and 5 correspond to the inflow flows of circulating water, which run through the condenser in two steps, until the end of the output by states 4 and 5.

Within the surface condenser studied, there are some limitations associated with the instrumentation's calibration that prevent the internal pressure being measured accurately, so the condensate temperature is used as the saturation temperature, which is a very good approximation considering that no trace of air within equipment. The circulating water's flow entering the condenser is 38000 gpm, with a steam quality of 88%. The studies performed are based on an operational data log under a condition of operation of the maximum load system, while the flows that reach the condenser through its drains are not considered.

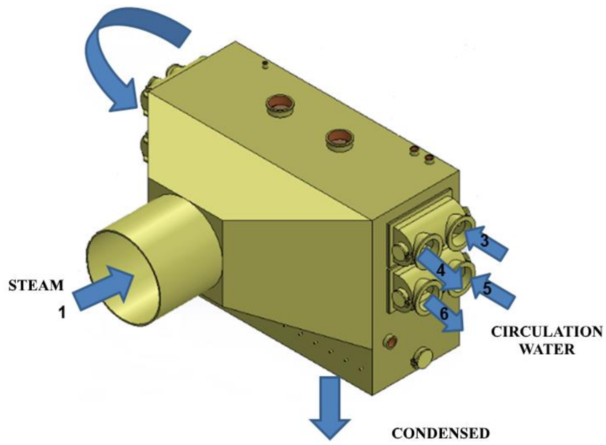

Figure 3. Schematic diagram of a surface condenser.

2.2 Current energy performance

The main cause of the surface condenser's low energy performance that was introduced in mid-2013 is determinate from a comparative analysis of the energy performance indicators of the condenser before and after a maintenance shutdown.

Additionally, the results are compared with the records of two maintenance events occurring during the same date, which correspond to the cleaning of the condenser pipes and the change of some condensate drain valves (Figure 2, valves 8-11). Among the performance indicators used are the exergetic efficiency concept [5], the methodology evaluation for condensers proposed by the Heat Exchangers Institute [6] and other basic parameters such as heat absorbed by water circulation, heat load and temperature increase [7].

2.3 Deviation from the initial energy performance

In order to determine the fundamental cause that generated the original operational deviation of the condenser with respect to the design conditions, visual inspections were made to the equipment, in addition to a complete monitoring of the operating conditions of the equipment from the first month of its implementation service. These analyzes were complemented by interviews with the operation and maintenance personnel of the power plant.

2.4 Optimal cleaning frequency

The optimum cleaning frequency, which minimizes the operation's total cost of the condenser, is calculated from a basic economic model [8] shown in equation (1)

$$Initial\ Cost = nC + n\,V \int_0^{1/n} f(t)\,dt, \tag{1}$$

where n is the number of cycles or clean frequency, C is the cleaning cost, V is the performance's loss by soiling cost and f(t) is the rate at which performance is lost by soiling. The value of $f(t)$ is obtained from the monitoring of the cleaning factor during 5 month of continuous operation, filtering data that are due to periods of maximum load.

3 Results and discussion

This section presents the results and the analysis of the energy evaluation of the surface capacitor before and after the maintenance stop, the results of a qualitative and quantitative study of the initial operational deviation of the equipment and the economic study for the determination of the optimal cleaning frequency.

3.1 Energy performance evaluation

The results of all evaluations made to the condenser before and after the maintenance stop are presented in Table 1.

Performance indicator	Before	After	Variation
Exergetic efficiency %	18.28	29.36	11.07
Cleaning Factor %	22.77	30.37	7.59
Condensed oxygen content	Equal	Equal	0
Pressure drop	15	15	0
Heat absorbed by circulating water (MBtu/h)	360.918	338.569	-6.19
Heat load (MBtu/h)	412.428	411.715	0.17
Temperature increase (°F)	19.006	17.834	-6.57

Table 1: *Result of condenser evaluations.*

When a condenser is effectively cleaned, this is reflected in the increase in the outlet temperature of the circulating water and in a decrease in the pressure drop [9]. The results shown in Table 1 show a different context, since the outlet temperature of the circulation water decreased and the pressure drop of the circulation water as it passed through the condenser remained practically constant. Figure 4 shows the final state of the surfaces of the pipes after the last cleaning with the "*Hydrojet*" system, which consists of the removal of deposits by means of the mechanical action of the water.

Figure 4. *State before and after the cleaning of the pipes.*

The results show that the deposits were not completely removed because the surface of the pipe has an inlay state and the selected cleaning system is not effective for the elimination of this kind of soiling. Thus, it can be concluded that the cleaning performed on the condenser pipes was ineffective and therefore had little significant influence on the improvement in the operating conditions reflected in the system.

On the other hand, if we analyze the temperature increase and heat absorbed by the circulating water, these values were higher before stopping, even though in theory the heat load received by the condenser was practically the same. This leads to the conclusion that before the shutdown, there was an additional heat load inside the condenser that kept the condenser in the overload state. This charge corresponds to the flows of superheated steam coming from the passes in the valves that were replaced. Although these types of flows are small compared to the total flow of steam entering the condenser through the turbine, their impact can be significant on the efficiency of the equipment. In figure 5, what happens when superheated steam enters the condenser through its drains is shown.

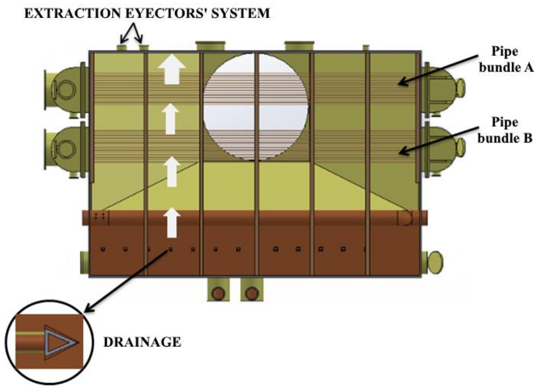

Figure 5. *Steam pass through condenser drains.*

As can be seen in Figure 5, when an overheated vapor flow exits through the drains within the condenser, it tends to rise directly towards the top of the condenser, without there being a mechanism that promotes its distribution on the pipe bundles. Due to this, in its almost vertical, the area available for heat transfer is very reduced considering the high level of energy that implies its overheating state; as a consequence, only part of the vapor condenses and the rest will accumulate in the upper part of the condenser, specifically affecting the performance of the upper pipe bundle (beam A). A proof of this is that before the stop the circulating water in the upper pipe bundled absorbed much more heat than it absorbs at present, where the steam passes have been eliminated. This explanation contrasts very well with the evidences found in the ejector system, in which a constant vapor output through the external vents of non-condensable gases was evidenced. This percentage of overload can be estimated from the performance curves of the condenser. Figure 6 shows such curves for the condenser in its state of operation before and after the maintenance stop; it also includes the curve of the equipment in its new or design state to obtain a better visualization of the current performance.

In figure 6 it can be seen that before the shutdown, the vacuum pressure recorded by the equipment at maximum load was 6.2 in HgA, then according to the current performance curve, where there are no steam passes, it can be said that before the stop the condenser was overloaded with approximately 40%. When a condenser is in a state of overload, the vacuum pressure increases abruptly from certain loads, as shown in Figure 6.

This is because in partial loads the effects of the condenser overload are not reflected because in this condition it is oversized in area. As the load increases, the excess area is reduced to the point that it is insufficient to maintain the rate of condensation and that is when the pressure rises sharply.

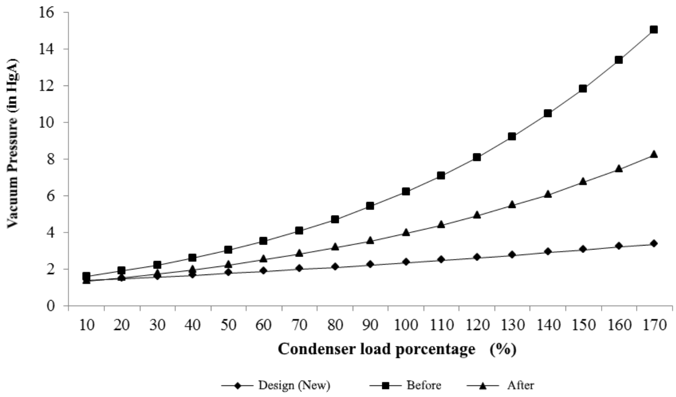

Figure 6. *Curves of condenser's performance.*

The continuous accumulation of superheated steam in the condenser increases the heat load on the system and increases the output back pressure of the turbine, impacting the generating capacity of the turbine. However, the improvement in the cleaning factor of the condenser has little correlation with the power recovered in the steam turbine; according to the results shown in Table 1, the increase in the cleaning factor of 7.6% should correspond to an increase in power of 0.6 MW and not to 3 MW as actually happened. This shows another important consequence of the steam passes and is related to the reduction that is made of the total steam flow that is injected to the turbine for the work's accomplishment.

3.2 Initial operational deviation of the equipment

Another important analysis that emerges from the previous results lies in the study of the individual performance shown by the two pipe bundles that make up the condenser. Historically these beams of pipes had always had different behaviors, being the pipe bundle superior the one of smaller efficiency. Prior to shutdown the upper beam performance improved noticeably, closely matching the performance of the lower pipe bundle; this improvement was associated with the amount of additional steam generated by the steam passes. At present these passes no longer exist and therefore the performance of the pipe bundle decreased because the steam does not reach the same proportion as the upper bundle as it did before the stop. This fact shows that the poor performance of the upper pipe bundle is due to the fact that it is not able to receive the steam flow for which it is designed; for this reason, it became necessary to study the internal configuration of the condenser. A comparison of the design planes of the condenser with respect to its actual configuration allowed to discover an initial assembly error of the equipment. A longitudinal cross-section of the condenser is shown in Figure 7 where the actual arrangement of the pipe bundles is appreciated.

It can be clearly seen in figure 7 that the penetration rails used for steam penetration are located on the rear side of the equipment in the opposite direction relative to the exhaust of the steam turbine. A more detailed analysis will show that in fact the whole condenser is located in a position opposite to that proposed by the design. In surface condensers these rails are designed to allow good penetration and distribution of the vapor over all the bundle of condenser pipes, but in the current equipment's position, this purpose is not fulfilled and, on the contrary, affects the vapor's distribution over the pipe bundle by reducing the heat transfer area and the vapor's condensation rate.

Another consequence of the poor position of the condenser has to do with the circulating water's flow entering the condenser; as now the circulation water must run through the system in the opposite direction to the one proposed by the design, because the pipe bundle is not symmetrical with respect to the water flow, there are additional energy losses that reduce the nominal flow. This is evidenced by the high pressure drops within the condenser that were evidenced since its commissioning. According to a study done to the circulation water pump's operation curves of the condensation system, the current flow has a great correlation with the pressure drop currently observed in the condenser.

Taking into account that during the first months of operation a detailed review of the parameters of the condenser operation was made, as well as a review of possible air infiltrations without significant results, it can be affirmed that the initial performance deviation was caused mainly by the effects produced by poor penetration of the vapor and by the reduction of the circulation water flow which generated the incorrect position of the condenser.

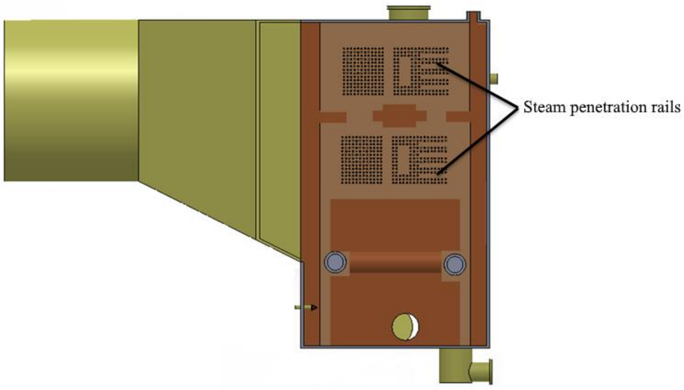

Figure 7. *Actual configuration of the pipe bundles in the condenser*

Figure 8 shows the initial efficiency's percentages lost by the condenser due to the circulating water flow's reduction and to the effects of the poor vapor penetration.

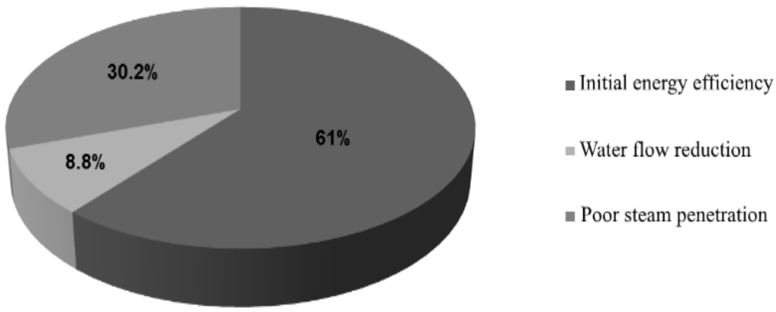

Figure 8. *Initial efficiency decrease in the condenser*

The results show an initial total efficiency's loss of 39%. If the condenser design cleaning factor of 85%, then the percentage of the cleaning factor that is reflected in the turbine load reduction is 24% and represents an initial loss of approximately 1.7 MW; because the initial real loss of steam turbine load was 1,643 MW, it can be concluded that the estimated percentage represents an excellent approximation.

The effects of the poor penetration of vapor on the condenser performance was an aspect that had never been taken into account and it was for this reason that during the first evaluations of equipment performance significant deviations were observed in the standard values of performance. For example, a month after its commissioning, the calculated cleaning factor for the condenser resulted in a 42% percentage, when the logic of the cleaning factor concept implies that in a new equipment this percentage should be around 100%. To eliminate such deviation, it is necessary to include the effects of the poor vapor penetration on the heat transfer coefficients calculated by the conditions of the condenser. This exercise was done and the results were that at one month of operation, the condenser cleaning factor was approximately 97%. This new value has a greater logic regarding to the performance indicator used.

3.3 Optimal cleaning frequency

After the cleaning done to the condenser, during a period of approximately 5 months, a decrease in the efficiency of this equipment was recorded due to the pipes' internal fouling. This decrease was expressed in terms of the theoretical power that could stop producing the turbine. Figure 9 shows the results obtained, where the mathematical correlation of such behavior is also indicated.

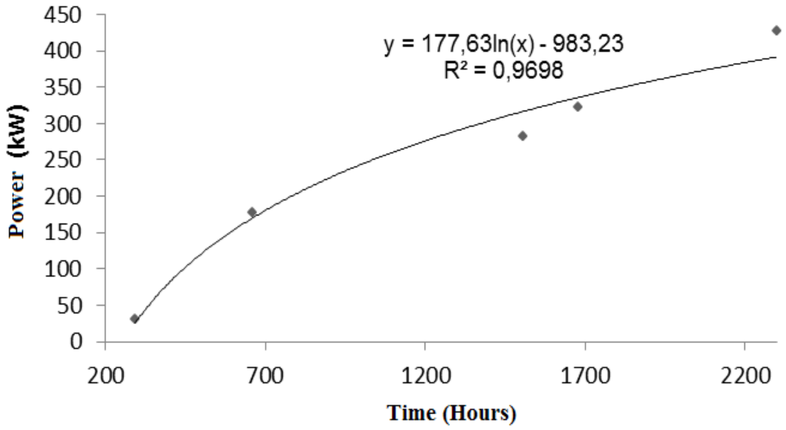

Figure 9. *Performance degradation rate in the condenser*

The turbine power's behavior over time indicates that as the equipment's operating hours increase, the turbine loses generating capacity as result of the condenser pipes' fouling. The curve resembles the asymptotic fouling models, which indicate that as the hours increase the rate of removal of the deposits tends to equalize with the rate of deposition [10]. Replacing the correlation obtained from the turbine power lost in equation (1), we obtain the operation's total cost of the condenser according to equation (2)

$$Total\ cost = nC + V177{,}63Ln\left(\frac{1}{n}\right) - V177{,}63. \qquad (2)$$

From this expression, the cleaning frequency n is calculated, which minimizes the sum of the operating costs and the maintenance cost, $n = 177{,}63V/C$. The cost of a cleaning with the *Hidrojet* system for this equipment, has a cost of approximately 60 MLL pesos and the average cost of a kWh in the studied power plant is 126 pesos, for one the optimum cleaning frequency is 3.5 months. Figure 10 shows total cost's behavior of operation as a function of the cleaning frequency, in which the optimum cleaning frequency value can be observed.

For maintenance frequency values less than 0.0004 cleanings per hour, the total cost of operation of the condenser will increase due to the loss of efficiency due to pipes' inner soiling; while higher values imply an extra cost that does not justify the improvement in performance, so this value of cleanings per hour is the optimal number of maintenance intervals that minimizes the sum of operating costs and the cost of capacitor's maintaining.

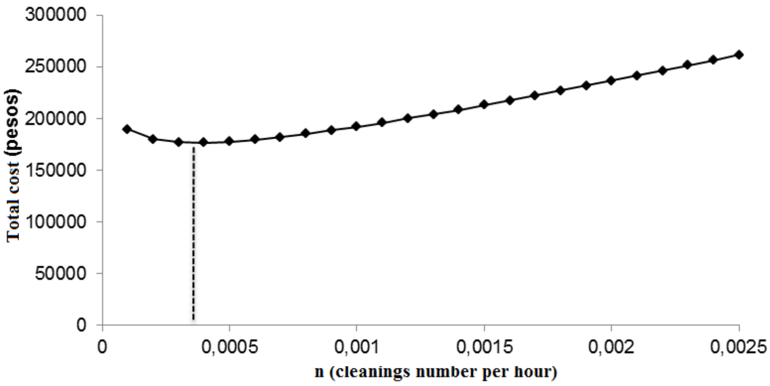

Figure 10. *Optimal cleaning frequency*

Figure 11 shows the behavior of the cleaning cost, performance loss and total cost as a function of the cleaning frequency, where it can be seen that the optimum cleaning frequency is the point at which these last two costs are equalized.

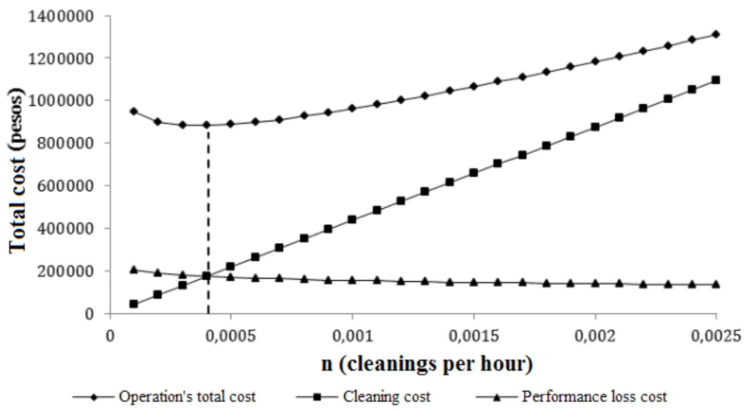

Figure 11. *Optimal cleaning frequency using the Hydrojet system*

4 Conclusions

The article determined the main cause that propitiated to the condenser a low performance at present, as well as the cause of the initial operational deviation with respect to the design performance that was expected of the equipment. A follow-up on the performance of the condenser allowed to determine the rate of degradation of efficiency of the condenser according to the state of the pipes' surface in the last five months and to establish the optimal frequency of cleaning that must be applied on this equipment to reach the best balance point between performance and economy. The low efficiency presented by the condenser before shutdown is associated with the passes of overheated steam that reached the condenser from the drains of the high pressure steam piping and injection steam of the combustion turbine. These steam passes kept the condenser in an overload state and at the same time decreased the available steam flow in the turbine for power generation; their joint effects showed a reduction for the 3 MW steam turbine.

As for the initial deviation of the condenser performance with respect to the design parameters, it is possible to affirm that this was a product of the loss of heat transfer's area, the condensation rate's reduction and reduction of circulation water's flow, due to the incorrect positioning of the condenser relative to the outlet of the steam turbine. Finally, a method was determined to calculate the optimum frequency of cleaning of the surface condenser in a simple and practical way, which can be used by the maintenance departments of the different thermal generation power plants.

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