Study of Steam Generation and Distribution in a Hospital to Improve Energy Efficiency Using Thermography, Ultrasound, and Gas Analyzer

Estudio de la Generación y Distribución de Vapor en un Hospital para la Mejora de Eficiencia Energética mediante Termografía, Ultrasonido y Analizador de Gases

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Abstract

This investigation studied the energy efficiency of a steam system in a hospital, considering the procedure in ASME EA-3-2009 standard. This is the standard for the energy assessment of industrial steam systems and aims to enhance the energy efficiency and sustainability of industrial steam systems by identifying opportunities and providing recommendations to optimize system performance. The obtained boiler energy efficiency was 80.29 %, by applying an energy balance, reaching 15.33 kW for heat losses in the distribution pipes. Two consistent improvement alternatives were proposed, starting by unifying the pipe diameter of the kitchen area of 5 m, generating a reduction of heat loss from 828 to 600 W, which represented a total annual energy saving of around 2.4 GJ/year. The investment cost is USD 47.40, considering the achievement of the break-even point after 9 months, where the insulation of the pipes with glass wool was considered second and it generated a reduction in losses of 5.98 kW, representing a total annual energy saving of about 62.91 GJ/year, which corresponds to approximately USD 1 672. The expenditure for insulating a 49 m pipe amounted to USD 462, factoring in both the NPV calculation and potential savings. The break-even point was achieved within roughly 4 months, underscoring the economic advantage of implementing the two suggested improvement measures.

Resumen

En esta investigación se analizó la eficiencia energética de un sistema de vapor en un hospital, considerando la norma ASME EA-3-2009. Esta norma se utiliza para la evaluación energética de los sistemas de vapor industriales y busca mejorar la eficiencia energética y la sostenibilidad de estos sistemas identificando oportunidades y proponiendo medidas para optimizar el rendimiento. Se obtuvo una eficiencia energética de la caldera de 80.29 % mediante un balance de energías y las pérdidas de calor en las tuberías de distribución alcanzaron 15.33 kW. Se plantearon dos alternativas de mejora consistentes, iniciando con unificar el diámetro de tubería del área de cocina de 5 m, generando una reducción de pérdida de calor de 828 hasta 600 W, que representó un ahorro de energía anual total de 2.4 GJ/año, aproximadamente. El costo de inversión es de USD 47.40, considerando como alcanzable el punto de equilibrio a los 9 meses, donde se planteó en segundo lugar el aislamiento de las tuberías mediante lana de vidrio y generó la disminución de pérdidas en 5.98 kW, representando un ahorro de energía anual total de alrededor de 62.91 GJ/año, lo que corresponde a USD 1 672, aproximadamente. La inversión para el aislamiento de tuberías con una longitud de 49 m fue de USD 462, considerando el cálculo del VAN y los posibles ahorros, además el punto de equilibrio se alcanza a los 4 meses, aproximadamente, e indica el beneficio económico de aplicar las dos opciones de mejora planteadas.

Index terms—Energy efficiency, shell boiler, heat transfer, steam, NPV.
Palabras clave— Eficiencia energética, caldera pirotubular, transferencia de calor, vapor, VAN.

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1. INTRODUCCIÓN

Los sistemas de vapor en hospitales son esenciales ya que producen vapor saturado para el funcionamiento de equipamiento, como autoclaves, calderas, calderas, lavadoras industrial, secadores e irones, y calentadores. Un sistema de vapor se forma por el calentador, el distribuidor de tuberías, y el consumo de equipamiento, que interactúa con cada uno, por lo tanto, si alguno funciona mal, se dañan el sistema, produciendo pérdidas de calor, relacionadas con los costos [1].

Ahora, el sistema de vapor es uno de los factores más importantes cuando los costos industriales son analizados. Según Palacios al. [2], las fugas de vapor se dan como uno de los más importantes problemas del sistema. Notablemente, las fugas de vapor pueden elevar los costos operacionales. Consecuentemente, los iniciadores de conservar energía deben comenzar con una completa revisión de estos problemas. Es crucial reconocer que aunque la más mínima fuga puede resultar en costos anuales de USD 7 000 [3].

La eficiencia energética aplicada a sistemas de vapor es fundamental ya que si se usa de la manera correcta, generando la mayoría de la energía para diferentes procesos. Yogesh et al. [4] indican que, al tener una buena eficiencia energética, los combustibles se generaron y ayudan a proteger el entorno, reduciendo la contaminación. Mandrela et al. [5] realizaron un chat de una empresa en la que usaron termografía y ultrasonido para reducir pérdidas de calor y producir ahorros en el sistema de vapor.

En las décadas recientes, varias investigaciones se han realizado, como Ocaña et al. [6], quienes realizaron un análisis energético en un hospital sistema de vapor, aplicando el método propuesto por el ASME EA-3-2009 para la evaluación energética. El objetivo del estudio fue determinar las pérdidas de calor resultantes del sistema de vapor y el condensado de retorno de las tuberías en las cuatro áreas exhibiendo la mayor consumo de vapor. Los resultados revelaron una pérdida acumulativa de 513 GJ/año, equivalente a aproximadamente USD 4 060. Al implementar la aislación en los tubos, se logró una reducción de 195 GJ/año, representando 38 %, por completo aislaron los tubos, los que generaron ahorros de USD 1 542/año. El costo estimado de instalar el sistema de vapor con aislamiento de fibra de vidrio fue USD 3 400, recuperable en un periodo de 2 años y 7 meses, considerando para el cálculo un subsidio del 73 %,

La eficiencia energética de un regulador de vapor fue presentado por Caetano et al. [7]. El método fue designado para analizar un modelo ATA 14 H 3N de la caldera, con una capacidad de 33.3 kg/s de vapor. Los datos fueron obtenidos mediante mediciones con una cámara termográfica y un analizador de gas de escape. La eficiencia energética del generador fue 74.65 %. Los resultados significativos de las pérdidas de calor fueron encontrados en el gas de escape, con 11.18 %, radiación y convección pérdidas de calor de 4 %, presentando un total de 25.35 %. Dos alternativas de mejoramiento fueron concluidas; el primero es reducir el exceso de aire y el segundo es colocar aislamiento térmico en los tubos no aislados. Perez et al. [8] llevaron a cabo un análisis del sistema de vapor de una fábrica, mostrando que la eficiencia energética estaba en 90.7 % y las pérdidas en los tubos sin aislamiento fueron 207.09 kW. Dos alternativas de mejoramiento fueron propuestas, el primero fue el aislamiento térmico de los tubos con fibra de vidrio y el segundo fue el diseño de un precalentador de agua a 70 ºC para alimentar el calentador de vapor. El aislamiento de fibra de vidrio en los tubos generaron una reducción de consumo de combustible de 92.30 %, el diseño del precalentador de agua generó una reducción de combustible de consumo de 64 186.51 L/año, y el coste del mejoramiento fue USD 450, este coste se recuperaría en 1.4 meses.

Ibrahim et al. [9] realizaron una investigación sobre el ahorro de energía en el sistema de vapor industrial de un aceite de palma. Durante el desarrollo, se empleó el sistema SSAT para la evaluación energética junto con el programa de software de aislamiento 3E Plus. Los resultados mostraron varias fuentes de energía, contribuyendo a aumentar los costos para el sistema, con una baja eficiencia energética de solo 59.6 %. El estudio identificó potenciales mejoramientos, específicamente a la instalación de un recuperador de agua y una reducción del vapor de múltiples bombas en un generador. Estos mejoramientos elevaron el eficiencia del sistema a 77 % desde su estado inicial. Consecuentemente, la calidad de vapor aumentó, conduciendo a un ahorro de energía de 75.28 GJ y ahorros de combustible de 598.3 t/año.

El análisis de eficiencia energética de 5 calderas fue realizado por Santana et al. [10]. Las temperaturas de referencia de los tubos aislados y no aislados fueron obtenidas con una cámara termográfica Testo 875. El estudio incluyó la determinación de la pérdida útil en el sistema de alimentación y un análisis de pérdidas de calor en las tuberías de transmisión. Las pérdidas de calor en los tubos aislados fueron de 13.32 y 4.22 kW para los tubos no aislados, con un total de 17.53 kW de pérdidas de calor. El análisis demostró que la pérdida de calor total de la instalación fue de 14.43 kW, con respecto a 243.02 kW, que representaba la pérdida útil, obteniendo un sistema de eficiencia energética de 94 %. 

El estudio termodinámico de un planta termoeléctrica de 49 MW fue desarrollado por Retirado et al. [11]. El método aplicado utilizó un algoritmo para evaluar tanto las pérdidas de vapor como las pérdidas de exerquía. Los resultados mostraron una eficiencia termodinámica en el uso de energía superior a 90.1 %, mientras que la eficiencia de exerquía fue registrada en 45.5 %. Suntivarakorn et al. [12] proponen mejorar la eficiencia de exerquía a través del uso de un sistema de control de la combustión automática. Este mejoramiento permitió reducir las emisiones de gases del chimenea para secado de aceite, calefacción avanzada, y...
regulating the burner air intake using a fuzzy logic control algorithm. Experimental findings demonstrated that heat recovery and fuel drying reduced fuel moisture content by 3 wt.%, leading to a 0.41 % increase in boiler efficiency. Air preheating resulted in a 35 °C temperature rise, contributing to a 0.72 % boost in efficiency. The fuzzy logic-controlled air system exhibited an accuracy of 89.15 %, correlating with a 4.34 % efficiency increase. When all three systems operated concurrently, a collective boiler efficiency increase of 5.15 % was achieved, translating to annual fuel savings of 246.88 tons.

Erbas [13] conducted a comprehensive study on the thermal performance of a coal-fired boiler, employing the energy balance method in accordance with the ASME PTC 4 standard. The analysis focused on a 75-ton-per-hour capacity boiler installed in the mining industry. The performance test revealed an impressive boiler efficiency of 93.07 %. Utilizing the indirect method, the top three contributors to the overall losses in the boiler were identified as water heat loss in the fuel at 4.03 %, dry flue gas loss at 3.23 %, and the proportion of unburned coal in the waste products at the end of combustion at 1.65 %.

Sagaf et al. [14] performed a prediction on the efficiency deterioration of two boilers in two power plants, with an individual capacity of 660 MW in Indonesia. ASME PTC 4 guidelines and linear regression method were followed. The findings highlight that the primary sources of thermal losses stem from hydrogen burning moisture in the fuel and heat loss attributable to moisture in the fuel. Notably, the degradation in boiler efficiency is measured at 0.19 and 0.4 % per year for the first and second units, respectively. One of the causes of the boiler efficiency deterioration is the use of coal of variable quality and the accumulation of ash in the economizer that reduces heat transfer.

This work aims to propose energy efficiency improvements by analyzing the boiler and distribution piping system, applying energy management techniques, and obtaining operating parameters to determine improvements. This document is divided as follows, Materials and Methods detail the utilized methodology to develop the study, focusing on energy management techniques to achieve outcomes. Results study the data in order to determine the improvements in the system. Finally, in Conclusion, the qualitative and quantitative information is synthesized to demonstrate the fulfillment of this work.

2. MATERIALS AND METHODS

2.1 Steam generation, distribution and consumption

Steam generation is carried out in a vertical boiler Model V1X69-150-9, with a power of 30 BHP, which runs on diesel and has the capacity to generate 469.47 kg/h of steam, working at a nominal pressure of 620.5 kPa.

The process begins when raw water from the public network is sent to two 64 m³ tanks, from which point the fluid is sent by a pumping system to the water treatment process, where it is softened with cationic resins. Then, the water is sent to the storage tank, where it is mixed with the condensate that returns from the process. The water is then supplied by a multi-stage pump to feed the boiler, usually at a temperature of 60 °C. On the other hand, the fuel is stored in a 7 570-liter main tank, and from there, it is pumped to the secondary tank that feeds the boiler. Subsequently, it is distributed by gravity to the burner, having an average fuel consumption of 18.93 liters per hour, at a fuel oil temperature of 18 °C.

Fig. 1 shows the distribution of steam generated to a storage manifold and to the consuming areas of the hospital steam system. The Laundry area consists of three sections, the Garment Drying area which is composed of an industrial dryer with a capacity of 60 kg, the Garment Washing area with two industrial washers of 45 kg capacity and the Ironing area with a 3 HP ironer. The next area is the Kitchen with 115-liter kettles, followed by the Sterilization area, which consists of two Autoclaves with 180 kW power. The last area is the Power House which consists of a water heater capable of generating 13.25 liters per minute.

Figure 1: Diagram of the hospital's steam system

2.2 Technical system status

The steam system has a main boiler and a reserve boiler and its thermal insulation is in good physical condition. Several sections of the steam and condensate return piping are not insulated, generating heat loss to the environment and energy costs, and also causing poor steam quality. The areas where most of the insulation is missing are the laundry and sterilization areas. Table 1 shows a review of the condensate distribution and return system. The external diameter (d_o), insulated (L_o) and...
Table 1: Data for insulated and non-insulated pipes and temperatures

<table>
<thead>
<tr>
<th>Pipe section</th>
<th>$d_p$ [mm]</th>
<th>$L_a$ [m]</th>
<th>$L_e$ [m]</th>
<th>$d_p$ [%]</th>
<th>$t_a$ [mm]</th>
<th>$T_{w,m}$ [K]</th>
<th>$T_f$ [K]</th>
<th>$T_e$ [K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boiler to Manifold</td>
<td>38.1</td>
<td>12.5</td>
<td>0.8</td>
<td>6</td>
<td>30</td>
<td>318.15</td>
<td>428.15</td>
<td>296.15</td>
</tr>
<tr>
<td>Manifold to the kitchen</td>
<td>19.05</td>
<td>22</td>
<td>4</td>
<td>15</td>
<td>30</td>
<td>313.2</td>
<td>410</td>
<td>294.2</td>
</tr>
<tr>
<td>Manifold to laundry</td>
<td>31.75</td>
<td>0</td>
<td>5</td>
<td>100</td>
<td>30</td>
<td>314.15</td>
<td>419.15</td>
<td>294.15</td>
</tr>
<tr>
<td>Manifold to steam autoclave</td>
<td>25.4</td>
<td>75</td>
<td>5</td>
<td>6</td>
<td>25</td>
<td>320.15</td>
<td>418.15</td>
<td>294.15</td>
</tr>
<tr>
<td>Manifold to water heater</td>
<td>25.4</td>
<td>7</td>
<td>1</td>
<td>13</td>
<td>30</td>
<td>318.15</td>
<td>417.15</td>
<td>294.15</td>
</tr>
<tr>
<td>Manifold to condensate return</td>
<td>31.75</td>
<td>107</td>
<td>6</td>
<td>5</td>
<td>30</td>
<td>311.15</td>
<td>368.15</td>
<td>294.15</td>
</tr>
</tbody>
</table>

non-insulated pipe length ($L_p$), insulation thickness ($t_a$), ambient temperature ($T_a$), surface temperature outside the insulated ($T_{w,m}$) and non-insulated ($T_f$) pipes were measured. For these records, a SATIR thermographic camera was used, taking measurements in 6 sections. To gather operating parameters of the system, pressure gauges and thermometers are strategically installed at key points of interest, complemented by the use of a DN100 electromagnetic flowmeter.

2.3 Technical system status

For the steam system diagnosis and subsequent energy efficiency improvement proposals, the recommendations of the ASME-EA-3-2009 standard [16] were followed. It is utilized for energy performance evaluation of steam systems, which suggests collecting and analyzing design, operation, energy use, and system performance data to identify opportunities for energy efficiency improvement. Efficiency was evaluated by the energy balance method [17]. Fig. 2 lists the causes for heat losses, due to dry flue gas loss ($a$), $H_2$ loss ($b$), fuel moisture ($c$), moisture in the air ($d$), CO loss ($e$), fly ash loss ($f$), surface loss ($g$) and bottom ash loss ($h$).

$$M_{air, theoretical} = \frac{[1.6 \cdot C + 34.8 \cdot \left( H_2 \cdot \frac{O}{8} \right) + 4.35 \cdot S]}{100}$$

(1)

Where $M_{air, theoretical}$ is the theoretical air quantity per fuel quantity. Excess air ($EA$) is obtained from gas analysis data by applying equation (2):

$$EA = \frac{O_{%}}{21-O_{%}} \cdot 100 \left[ Data_{gas\, analysis} \right]$$

(2)

Then the amount of actual air mass ($M_a$) entering, per amount of fuel, is determined [18]:

$$M_a = \left[ 1 + \frac{EA}{100} \right] \cdot M_{air, theoretical}$$

(3)

The mass of dry flue gas ($m$) is then determined:

$$m = \frac{C \cdot PMCO + N_{2} + 2 \cdot S + 0.77 \cdot M_a + 0.23 \cdot (M_a - M_{air, theoretical})}{100 \cdot PMC}$$

(4)

It is essential to identify the diverse thermal losses associated with the boiler, commencing with the determination of heat loss due to dry flue gas ($L_a$). This loss, considered the primary one for the boiler, is derived using equation (5):

$$L_a = \frac{m \cdot C_p \cdot (T_{w,m} - T_e)}{PCI} \cdot 100$$

(5)

The thermal loss due to the evaporation of water formed by $H_2$ in the fuel ($L_2$) is related to the combustion of hydrogen, which causes a heat loss because the combustion product is water, which is converted into steam and is calculated with the equation (6):

$$L_2 = \frac{9 \cdot H_2 \cdot \left[ 584 + C_p \cdot \left( T_{gas} - T_e \right) \right]}{PCI} \cdot 100$$

(6)

To determine the loss due to the moisture present in the fuel ($L_3$), it is considered that the moisture that enters with the fuel produces a superheated steam, and it is determined by equation (7):

$$L_3 = \frac{H_2 \cdot O_{%} \cdot \left[ 584 + C_p \cdot \left( T_{gas} - T_e \right) \right]}{PCI} \cdot 100$$

(7)
In computing the loss attributed to moisture in the air ($L_4$), the mass of vapor in the air is ascertained using psychrometric charts. The quantification of this moisture-related loss is achieved through the application of equation (8):

$$L_4 = \frac{M_a \cdot M_{R_d,\text{air}} \cdot C_p \cdot (T_{\text{sat}} - T_a) \cdot 100}{PCI}$$  \hspace{1cm} (8)

Heat losses due to incomplete combustion ($L_5$) are the products containing CO, H$_2$, and hydrocarbons that are in the combustion, value determined with equation (9):

$$L_5 = \frac{\% \text{CO} \cdot C}{\% \text{ CO} + \% \text{CO}_2 \cdot PCI} - 100$$  \hspace{1cm} (9)

The surface losses ($L_6$) are calculated by knowing the boiler surface area and temperature:

$$L_6 = 0.548 \left( \frac{T_{\text{boiler}}}{55.55} \right) - \left( \frac{T_{\text{sat}}}{55.55} \right) + 1.957 \cdot \left( T_{\text{boiler}} - T_e \right)^{1.25} \left( \frac{196.85 \cdot V_m + 68.9}{68.9} \right)$$  \hspace{1cm} (10)

Once the different heat losses have been determined, the boiler efficiency is determined by equation (11) [19]:

$$n_{\text{add}} = \left[ 100 - \left( \sum \text{Total losses} \right) \right]$$  \hspace{1cm} (11)

### 2.4 Calculation of heat losses in pipelines

Fig. 3 presents the calculation method established in the scientific literature to determine losses in insulated steam pipes, based on NOM-009-ENER-1995 [20].

![Figure 3: Methodology of thermal losses in insulated pipes](image)

In the equations presented in Fig. 3, the dimensionless geometric coefficient for pipes ($C$) is denoted as 1.016. The variable $V$ represents wind speed, which, in this scenario, is assigned a value of zero due to the entirety of the installation being situated within the confines of a building. Additionally, $E$ and $k_{\text{insulation}}$ correspond to the emissivity and conductivity of the insulation, respectively, while $T_w$ signifies the operating temperature of the boiler.

In the absence of insulating material, pipes are regarded as horizontal cylindrical surfaces that dissipate energy through a combination of convection and natural radiation. The calculation method established in the scientific literature to calculate the losses in non-insulated steam pipes was considered with the modeling proposed by Cengel [21]. The operating conditions are considered to be stationary and the formulation synthesized and systematized in Fig. 4 is used.

![Figure 4: Methodology of thermal losses in non-insulated pipes](image)

### 2.5 Calculation of heat losses in pipelines

The primary alternatives proposed include pipe diameter standardization and thermal insulation, with the initial focus on pipe diameter standardization. The steam pipes within the hospital exhibit varying diameters, spanning from 12.7 mm (0.5 in) to 38.1 mm (1.5 in). Certain sections necessitate this range of diameters to meet the steam demand of the associated equipment. However, in the section from the steam manifold to the boiling pans located in the kitchen, 2 sizes of pipe...
diameters prevail, due to the lack of adequate couplings to maintain a single pipe diameter, which generates a greater heat loss. For this pipe section, the diameter is unified by acquiring the fittings and pipe of the same size, and the calculation is made in the pipe section, by applying the calculation model shown in Fig. 4. After calculating the pipe loss by unifying the diameter, the loss due to the variety of diameters is determined by means of equation (12):

\[ q_{\text{diameter}} = q_{\text{diameter, variety}} - q_{\text{diameter, unified}} \]  

(12)

In addition, it is proposed to implement thermal insulation in the non-insulated distribution pipe sections. Initially, the thermal losses for each uninsulated pipe section are reassessed by introducing thermal insulation. The calculations adhere to the methodology outlined in Fig. 4. Given that the surface temperature outside the insulation \( T_{\text{sup}} \) is not known, an initial temperature value is assumed, and the corresponding thermal losses \( (q_{\text{a}}) \) are then determined. To obtain that the calculated thermal losses \( (q_{\text{a}}) \) are admissible, the iterative approach suggested in NOM-009-ENER-1995 [20] is used, with equation (13):

\[ T_{\text{sc}} = T_{\text{op}} - \frac{q_{\text{a}}}{2\pi \cdot K_{\text{insulation}}} \ln \frac{d_{\text{op}}}{d_{\text{sc}}} \]  

(13)

This equation determines if the calculated value is acceptable, by obtaining the calculated surface temperature \( T_{\text{sc}} \) and comparing it with the surface temperature of the insulation, initially assumed for the calculation of losses in the insulated pipes. If \( T_{\text{sc}} \) is equal to \( T_{\text{sup}} \), then the heat losses are acceptable, if \( T_{\text{sc}} \) is different from \( T_{\text{sup}} \) then the \( q_{\text{a}} \) calculations must be repeated, equaling \( T_{\text{sc}} \) to \( T_{\text{sup}} \), and then the temperatures are compared again. After determining the losses of the insulated pipes, the heat loss due to lack of insulation is determined with equation (14):

\[ q_{\text{insulation, lack}} = q_{\text{non-insulated}} - q_{\text{insulated}} \]  

(14)

3. RESULTS

3.1 Boiler energy efficiency analysis

Table 2 details the results of heat losses after calculating the boiler efficiency, with the objective of finding opportunities for improvement.

<table>
<thead>
<tr>
<th>Heat Losses</th>
<th>Value [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>( L_1 )</td>
<td>10.41</td>
</tr>
<tr>
<td>( L_2 )</td>
<td>7.52</td>
</tr>
<tr>
<td>( L_3 )</td>
<td>0.03</td>
</tr>
<tr>
<td>( L_4 )</td>
<td>0.21</td>
</tr>
<tr>
<td>( L_5 )</td>
<td>0.01</td>
</tr>
<tr>
<td>( L_6 )</td>
<td>1.54</td>
</tr>
<tr>
<td>( \eta )</td>
<td>80.29</td>
</tr>
</tbody>
</table>

It is observed that the highest loss is due to dry combustion gases, with a percentage of 10.41 %, due to the excess of air present in the combustion equal to 15 %, followed by the loss of hydrogen in the fuel with 7.52 %. The heat loss by radiation and convection is within the parameters, according to Saidur et al. [22], which indicates that these losses represent a maximum value of 2 %. Finally, a boiler efficiency of 80.29 % is obtained, which is a value that can be considered tolerable, although it can also be improved for the type of fuel used, according to Retirado et al. [23].

3.2 Thermal loss analysis of the piping system

The analysis aimed to identify opportunities for improvement by assessing thermal losses in both insulated and non-insulated pipes within the system. The pipes were segmented into six sections, and characteristic geometric values of the pipe system, along with surface temperatures in both insulated and non-insulated pipes, were measured. Surface temperatures ranged from 40 °C for insulated pipes to a maximum of 150 °C for non-insulated pipes values falling within the application range for glass wool as per ASTM C-552 [24] and C-1696 [25]. Table 3 outlines the losses per section, with the total loss calculated at 15.33 kW.

![Figure 5: Section losses in insulated and non-insulated pipes](image_url)

<table>
<thead>
<tr>
<th>Heat Losses [kW]</th>
<th>Boiler to Manifold</th>
<th>Manifold to the kitchen</th>
<th>Manifold to laundry</th>
<th>Manifold to steam autoclave</th>
<th>Manifold to water heater</th>
<th>Manifold to condensate return</th>
</tr>
</thead>
<tbody>
<tr>
<td>In insulated pipes</td>
<td>0.41</td>
<td>0.50</td>
<td>1.52</td>
<td>2.17</td>
<td>0.19</td>
<td>3.19</td>
</tr>
<tr>
<td>In uninsulated pipes</td>
<td>0.22</td>
<td>1.31</td>
<td>4.20</td>
<td>0.85</td>
<td>0.17</td>
<td>0.62</td>
</tr>
<tr>
<td>Total, in sections</td>
<td>0.63</td>
<td>1.81</td>
<td>5.72</td>
<td>3.02</td>
<td>0.35</td>
<td>3.81</td>
</tr>
<tr>
<td>Total, in pipes</td>
<td>15.33</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 5 presents the actual loss values for the insulated and uninsulated pipes in each section. The current total heat loss of the piping system is 15.33 kW, of which, 7.36 kW is lost in the uninsulated pipes, representing 48 % of the total losses. Currently, the steam system falls outside the parameters defined by the NEC energy efficiency standard regulations [26], as it exceeds the permissible maximum energy loss of 4% attributable to inadequate insulation [27].
3.3 Steam system efficiency improvements

The piping section from the steam manifold to the kitchen area is made up of two diameters, due to the lack of fittings and piping, identifying a 5 m section with different diameters. It generates greater heat loss, therefore, the effect of unifying the pipe diameter is considered. The analysis is carried out regarding the installation of 19 mm (0.75 in) pipe, of the same diameter as the other pipes in this section. In this manner, a heat loss reduction of 228 W is generated and represents a total annual energy saving of about 2.4 GJ per year, which corresponds to USD 64. Fig. 6 compares the heat loss in the section without unifying the pipe diameter and unifying the diameter, which was calculated considering that the steam distribution system works 8 hours per day, the whole year.

![Figure 6: Loss per diameter cross-section](image)

3.4 Insulation of distribution pipeline

For the analysis, it is proposed to install an insulator with the same characteristics as those installed in the insulated pipes. The insulators possess the characteristic of being composed of glass wool material featuring an aluminum coating adhered to it, with insulation thickness ranging from 25 to 33 mm. Through measurements with a thermographic camera, it was determined that these insulation thicknesses guarantee that the surface temperature is within the parameters allowed in Cengel [21], where the surface temperature should not be higher than 60 °C. Fig. 7 shows the effect of insulating the pipe sections without thermal insulation.

![Figure 7: Comparison of thermal losses in distribution pipelines](image)

In the Boiler to Manifold section, there was a loss reduction of 0.19 kW. Next, for the Manifold to the kitchen section, a reduction of 0.88 kW was obtained, in the Manifold to laundry, 3.63 kW, in the Manifold to steam autoclave section, 0.71 kW, in the Manifold to water heater section, 0.14 kW, and in the Manifold to condensate return section, 0.44 W. All these savings produce a total reduction of 5.99 kW, and represent a total annual energy saving of 62.91 GJ per year, which corresponds to approximately USD 1672, value that was calculated considering that the distribution system works 8 hours per day during 365 days of the year.

A financial analysis was conducted to assess the project, aiming to determine the necessary investment costs and the anticipated payback time. Hence, the estimated implementation cost for insulating the steam piping system amounts to USD 9.43 per meter. When taking into account the Net Present Value (NPV) and potential savings, the break-even point becomes achievable within 4 months. Beyond this timeframe, the initial investment proves worthwhile, generating additional income. This rapid payoff not only recoups the investment but also generates surplus funds that can be directed toward further investments in hospital machinery and infrastructure. Additionally, an analysis was conducted by implementing pipe diameter standardization. The examination revealed that the investment required to standardize the pipe diameter in the Manifold to kitchen section, covering a distance of 5 m and including the acquisition of necessary pipes and accessories, amounts to USD 47.40. According to NPV calculations, the break-even point for this investment could be realized after 9 months.

4. CONCLUSIONS

The efficiency of the boiler was obtained by calculating the six thermal losses present in the combustion, determining an efficiency of 80 %. The greatest loss is produced by the dry combustion gases, representing 10.41 %, attributed to the excess air that has a presence in the combustion of 15 %. To produce improvements, excess air can be reduced to 10 % by increasing the frequency of burner maintenance. By improving the excess air, the flue gas temperature is also reduced, which was 295 °C, data that were recorded using technological equipment, such as a thermographic camera and gas analyzer. The heat losses of the 6 sections of the system were 15.33 kW, of which 7.36 kW is lost in the uninsulated pipes, representing 48 % of the total thermal losses.

With the unification of the pipe section from the Manifold to the kitchen, a reduction of heat loss from 828 to 600 W is generated. It represents a total annual energy saving of 2.4 GJ per year, which corresponds to approximately USD 64. The thermal insulation of the pipes with glass wool represents a total annual energy saving of 62.91 GJ per year, related to USD 1672, both scenarios were calculated considering that the system operates 8 hours per day during the 365 days of the year. The expenditure for insulating a 49-meter pipe is USD 462. This cost, factoring in the NPV of the investment and potential savings, leads to the break-even
point being achieved after 4 months. Similarly, for the standardization of the pipe diameter in a 5-meter section, involving the purchase of necessary pipes and fittings at a cost of USD 47.40, the NPV analysis suggests the break-even point could be attained within 9 months. These findings underscore the economic benefits associated with implementing the proposed improvement options.

REFERENCES


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