14th International Heat Pipe Conference (14th IHPC), Florianópolis, Brazil, April 22-27, 2007.

DESIGN OF A HEATER FOR NATURAL GAS STATIONS ASSISTED BY TWO-PHASE LOOP THERMOSYPHON

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ABSTRACT

The thermal model, used in the preliminary design of a gas station heater, is presented. The aim of this research is to develop equipment that should be a new option to heat the natural gas at city-gates stations. The two-phase close loop thermosyphon technology is employed for indirect heating, with water as working fluid and the material selected is carbon steel. At present, water-bath heater is widespread used for gas industry to heat the natural gas. Its large size, the need for water reposition, its low thermal efficiency, are the main operational problems of the traditional equipment. To design better equipment is the main challenge of the present work. Furthermore, the recent success obtained by the use of two-phase thermosyphons in many different types of equipment for industry encourages this research. The heater design procedure is showed. This new concept resulted in small equipment. On the other side, it is not possible to use one simple on-off burner, which is used in the water-bath heater, the conventional equipment. One transient analysis is made to define how many levels the burner must present to work properly.

KEY WORDS: **natural gas heater; gas pipeline station; loop thermosyphon; thermal design.**

1. INTRODUCTION

This paper shows the preliminary design of a two-phase thermosyphon heater, used to raise the temperature level of natural gas in city-gate stations, which is a coupling point between natural gas transport system and the local gas delivery company. The delivery pressure established by the contracts is smaller than the pressure found in the transporter pipelines. The pressure reduction may cause the natural gas temperature decrease resulting in potential problems, like the material embrittlement and the hydrate formation. Thus, the city-gate station demands heating systems in order to guarantee an acceptable delivery gas temperature range.

One safe and rational way to heat one hydrocarbon fuel by fire is using the indirect heater. The water-bath-heaters are the most common type of natural gas heater on city-gate stations. This type of heater, showed in Figure 1, is an uncomplicated system composed by one vessel filled with water where a serpentine pipe, through which natural gas flows is submerged. In this same water, a fire tube, that provides the heat demanded to maintain the water around 70 ºC, is swamped. The actual water-bath heater (Figure 1) is an old well-known project, which requires constant water reposition due evaporation process and has a low relation power/weight. However, it has the industry confidence, resulted from the fact that it is easy to produce and operate. Also, it is almost entirely electric independent. Sullivan (1971), in his US patent, proposed some modifications to this design, using the one-phase thermosyphon principle to increase the heat transfer. He also proposed a different geometry to raise the ratio power/weight. These same advantages are desirable here, although, based on the success of the laboratory in other equipment development using thermosyphon technology, even better thermal results are expected. Other important desired features are: no need for water reposition, low-cost, reliability, safe operation and electric independence.

The Figure 2 illustrates the lay out of a city gate station, with details of the heating unit. In addition, one pressure-reduction valve is shown to clarify how the heating system works.

Figure 1: Actual water-bath heater.

The natural gas comes from transporter's pipeline, on left side of the Figure 2. Then the gas flow is divided in two parts: the first pass through the heater and the second by-pass the heater. With this layout, the three-way valve has the control of natural gas temperature at point (2) by changing the percentage of gas bypassed from the heating system.

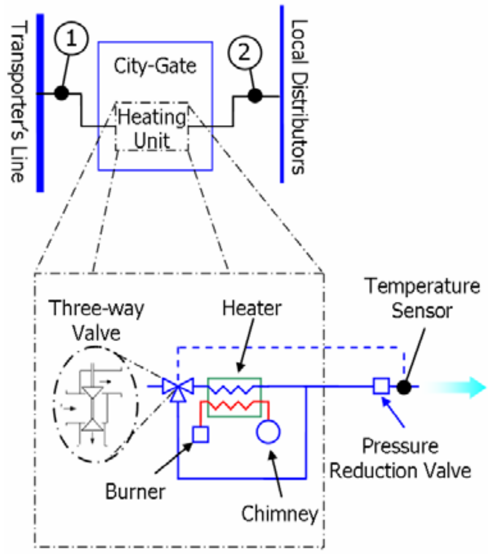


Figure 2: Layout of a city-gate, with heating unit details and a three-way valve scheme.

1.1. Literature review

Many authors suggested the use of closed loop thermosyphons technology to solve many industrial thermal problems, such as Pioro and Pioro (1997), who stress the advantages of this high thermal performance devices. According to Vasiliev (2005), one single loop thermosyphon, called vapor dynamic thermosyphon, can transport up to 10 kW, for several meters of distance in the horizontal direction. Its thermal resistance ranged between 0,03 and 0,05 K/W. Lamfon, et al. (1998), also present other examples of applications involving cooper-water two-phase loop thermosyphons. .

According to Antoniuk and Pohner (2002), in most of the two phase loops, intermittent or repeated operational failures are due to the presence of non condensable gases (NCG), which can not be fully eliminated from the system. Also, it is unavoidable that small amounts of NCG will be formed during the system lifetime. So, the equipment design should be NCG-tolerant. Vasiliev (2005) states that a loop thermosyphon, in which the vapor flow in one co-axial gap, can push the non-condensable gases to the gas trap. Therefore, in this case, thermosyphons can work with non-condensable gases inside.

The Chinese experience, represented by the work of Zhang and Zhuang (2003), shows that carbon steel - water heat pipes can be applied in actual applications, provided that the carbon steel-water compatibility problem is solved. Moreover, they affirm that the cost of heat pipe equipment has been greatly dropped by the use of the pair carbon steel-water. An application of a heat pipe steam-generator for sulfuric production, reported in their paper, indicates the operation reliability of a heat pipe under severe working conditions, including high temperatures and severe corrosion environments.

The literature review developed in this work did not show any application concerning the use of a closed loop thermosyphons technology for heating high-pressure gas (~ 50 bar).

This paper presents a summary of the pre-design methodology for a loop thermosyphon heater. First, the equipment is designed for critical conditions, which establishes its main dimensions.. Then, the results of one transient analysis for three charge conditions are shown and explored.

2. CRITICAL CONDITION DESIGN

This section defines the critical condition and the equipment geometric configuration. The heat transfer coefficients are estimated, to find the heat transfer areas. In addition, the main characteristics and constrains are shown.

Figure 3 shows, schematically, the closed loop thermosyphon configuration selected for this preliminary study. The hot gas passes inside the pipes that are externally flooded in the working fluid. The working fluid evaporates and the vapor passes through the steam-water separator. The vapor flows through the pipe until the condenser drum. At the condenser, heat is released to the natural gas pipe, causing the condensation outside the serpentine gas pipe. The circuit closes by working fluid return to the evaporator by the pipe located at the bottom of the condenser drum.

The operational temperature of the equipment is defined to be of 115 ºC with a range of $\pm 10^{o}C$. Since the desired delivery gas temperature is 20 ºC, the heater should operate at low temperatures, just a few degrees above the gas temperature. However, if the heater works at pressures below atmosphere, i.e. temperatures below 100 ºC, the air from the ambient can go into the vessel when leakage occurs. Pressures above atmosphere result in more thick walls and the adoption of more strict safety rules. Even though, pressures above atmosphere are justified because they avoid the risk of air to go inside the vessel, which could reduce the reliability and lifetime of the equipment

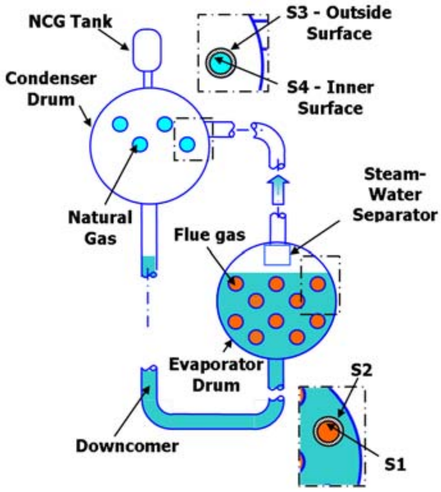


Figure 3: Geometric configuration proposed in preliminary design.

The selection of the water level, defined to cover all tubes in the evaporator drum, together with the steam-water separator, avoid geyser problems. If the water level was at the condenser bottom, the separator would not be necessary. This would raise the system water mass and therefore, the thermal inertia A cost/benefit analysis is necessary for a decision of the best water level position. Another issue to be considered is the NCG presence and generation. The literature reports many studies of this subject, specially the hydrogen generation of the pair water-steel. Therefore, the present equipment should be NCG tolerant. A tank is placed at top of the condenser drum to collect the gases at the startup and during the normal operation. This subject has been studied at the laboratory thorough an experimental apparatus.

When it comes to the Natural Gas Industry, the security and reliability are major factors to be considered, when the equipment is designed. Another important requirement of this project is the electricity autonomy of the system, i.e. no motor or fans are allowed. The impossibility of the use of an electrical fan to raise the hot gas velocity at combustion air side, constitutes the main constraint in achieving better thermal efficiency.

2.1. Theoretical model

In actual heaters, the combustion chamber is flooded inside the thermal bath working fluid. This is an interesting thermal solution, specially for cases where no fan is applied, such as in the present case. Fans can be used to blow the hot gases through the combustion chamber, improving the heat exchange between hot combustion gases and the working fluid. However, at the critical condition design stage, the heat transfer mechanism that involves the combustion chamber is not considered and the heat released is disregarded. Nevertheless, many combustion calculi are made to prepare this model for further combustion calculations.

The maximum power required from heater is defined as the maximum mass rate for which the whole city-gate is projected, with the maximum enthalpy difference possible. To perform the calculation, the natural gas is considered like pure Methane. Then the maximum power demanded can be expressed by Equation 01

$$Q_{tot,NG,MAX} = m_{NG}\left(h\left(T_{NG,2},P_{NG,2}\right)-h\left(T_{NG,1},P_{NG,1}\right)\right), \quad (01)$$

where the $h(T,P)$ is the thermodynamic enthalpy of Methane; $T_{NG,1}$ $P_{NG,1}$ are the temperature and pressure at the transporter pipeline; $T_{NG,2}$ and $P_{NG,2}$ are the temperature and pressure of the local distributor; and m_{NG} is the natural gas mass rate.

In addition, the total power delivered by the hot gas core is larger than the useful power. Therefore, the total power is composed by: the useful power, the losses to atmosphere and the heat rate used to rise the working fluid temperature up to the maximum defined operating temperature, according to the expression:

$$Q_{burner} = Q_{useful,NG} + Q_{loss} + Q_{rise\_T,WF} \quad (02).$$

The last term of this equation is defined as the energy used to rise the temperature from the lowest to highest operational temperatures, according to:

$$Q_{aquec,WF} = m_{WF} C(T_{WF,MAX} - T_{WF,MIN}) / \Delta t , \quad (03),$$

where, m_{WF} is the working fluid mass, C is the water heat capacity evaluated at mean temperature (~110 ºC), and Δt is the time interval desired. However, the power transferred to the working fluid is less than the power supplied by the burner. The thermal efficiency of the heat exchange process between the flue gas and the working fluid is defined to be at 60 % for this preliminary critical condition design.

To estimate the heat transfer coefficients at the four heat transfer areas (S1, S2, S3 and S4) showed at detail in Figure 3, the equations presented at Table 1 are used. The Reynolds numbers play an important role in heat transfer process at the hot gas side. To calculate it, the flow mass and its chemical composition need to be estimated.

The combustion chamber usually is not adiabatic, and releases heat to the working fluid. This heat has an important effect in the heat transfer by convection inside the pipe because it changes the inlet temperatures of pipes. In the present paper, this heat is not considered in the calculation, because the combustion chamber is not designed yet. It will be the taken into account in the next step of the project. The hot gas temperature at inlet is defined to be the in the same level of the temperature measured in the water-bath heater, 750 ºC.

Table 1: Equations used in critical condition design

$\frac{\bar{h}_{s4} D_{H,NG}}{k_{NG}} = 0.0214(Re_{NG}^{0,8} - 100) Pr_{NG}^{0,4}$ (04),	heat transfer coefficient (HTC) in natural gas pipe;
$\bar{h}_{s3} = 0.728 \left(\frac{\rho_L^2 g \Delta h_v k_L^3}{\mu_L (T_{WF} - T_{s3}) d} \right)^{1/4}$ (05),	HTC outside gas pipe;
$\frac{\bar{h}_{s1} D_{H,HG}}{k_{HG}} = 0.0214(Re_{HG}^{0,8} - 100) Pr_{HG}^{0,4}$ (06),	HTC inside hot gas pipe;
$\bar{h}_{s2} = 1.95 q^{0,72} (P_{WF}[bar])^{0,24}$ (07a), $q = Q_{tot,NG,MAX} / A_{s2}$ (07b),	HTC outside hot gas pipe; heat flux;
$M \Delta h_v = \bar{h}_{s3} (T_{WF} - T_{s3}) A_{s3}$ (08),	mass flow between drums;
$CH_4 + 2e(N_2 + 3,76O_2) \Rightarrow CO_2 + 2eH_2O + 7,52eN_2 + 2(e-1)O_2$ (09),	simple “complete oxidation” of CH_4 with Air
Where h is the heat transfer coefficient; D is the hydraulic diameter; k: thermal conductivity; Re: Reynolds number; Pr: Prandtl number; μ: dynamic viscosity; ρ: density; g: gravity; Δh: vaporization enthalpy; T: temperature; q: heat flux; Q: heat rate; P[bar]: pressure in bar; A: area; e: air excess; M: mass. In addition, subscript $_{NG}$ is related to natural gas; $_{S}$ are the heat transfer areas; $_{WF}$: working fluid; tot: total; $_{L}$: liquid; $_{MAX}$: maximum.	

In Eq. 8, complete combustion, with no dissociation of CH4-air mixture, is used to predict the composition and mass flow at defined excess of air, represented by the letter *e*. The properties are calculated according to the molar fraction weighted mean. The apparent molecular mass is used to predict density, according to the ideal gas theory. The hot gas mass rate is defined dividing the power sourced by the burner Q_{burner} by PCI of natural gas.

Since the heat rate, the temperature difference and the heat transfer coefficients are know, the heat transfer area can be determinate. The main dimensions can be defined by try and error procedure varying diameter, length, and number of passes. One example of a equipment for a heating power of 0,1 MW is shown in Figure 4. It is divided in 3 vessels to reduce the drum diameter.

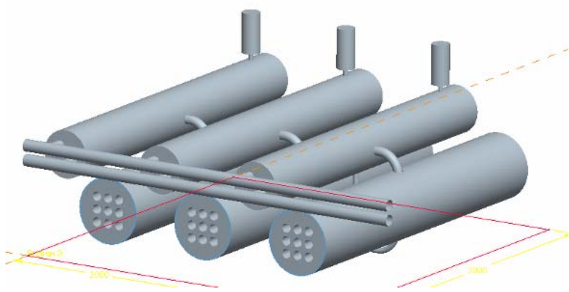


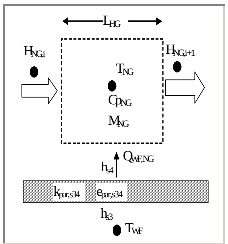
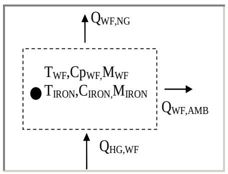
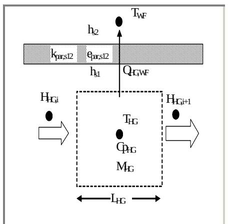
Figure 4: Preliminary design result for power of 0.1 MW, and (2m x 2m size)

The small size of the equipments with this geometric is a remarkable characteristic. Small equipments presents reduced the thermal inertia, costs, and losses to the environment. On other side, small thermal inertia forces the burner to start and shutoff many times. This is not desired because transient process causes more pollution and higher chance of failure. Actually, one of the most common problems in actual heater is the blow-off at startup. For the present case, a dynamic analysis of many parallel on-off burners or of one burner which power is controlled by the working fluid temperature is needed to make this equipment one real possibility.

3. TRANSIENT THERMAL ANALYSIS

The behaviors of the three-way valve and of the burner are responsible for the transient response of the working fluid and of the material walls, which, in turn, can present a large thermal inertia. Data about the dynamic response of a temperature controlled three-way valve is not available in common data sheets. However, it is known that the time response is fast, around 1,5 s for many applications (Sanson, 2006). A thermosyphon in operation feels almost immediately the valve variation. Then, in the proposed model, the valve response time corresponds to the minimal timestep used to solve numerically the system of equations.

Table 2: Equations used in model

$T_{NG,i+1} = T_{WF} + \left(T_{NG,i} - T_{WF}\right) e^{-\left(\frac{\overline{U}_{s4} P'_{s4}}{\frac{\dot{m}_{NG}}{n_{\sec}} Cp_{NG}} L_{NG}\right)}$ (10a), $\overline{U}_{s4} P'_{s4} = \left(\frac{1}{h_{s4} \pi d_{s4}} + \frac{\ln(d_{s3}/d_{s4})}{2\pi k_{par;s43}} + \frac{1}{h_{s3} \pi d_{s3}}\right)^{-1}$ (10b),	
$\frac{m_{WF}\left(E(T_{t+1}) - E(T_t)\right) + m_{IRON} C_{IRON} \left(T_{t+1} - T_t\right)}{\delta t} = Q_{HG,WF} - Q_{WF,NG} - Q_{ENV}$ (11),	
$T_{HG,i+1} = T_{HG,i} + \left(T_{HG,i} - T_{WF}\right) e^{-\left(\frac{\overline{U}_{s1} P'_{s1}}{\frac{\dot{m}_{HG}}{n_{\sec}} Cp_{HG}} L_{HG}\right)}$ (12a) $\overline{U}_{s1} P'_{s1} = \left(\frac{1}{h_{s1} \pi d_{s1}} + \frac{\ln(d_{s2}/d_{s1})}{2\pi k_{par;s12}} + \frac{1}{h_{s2} \pi d_{s2}}\right)^{-1}$ (12b)	
Where, T is temperature, $\overline{U}$ is the overall heat transfer coefficient; δt: time interval; P\`: perimeter and E is the internal energy.	

In the equipment design, a burner, with many discrete levels of burning power is considered. In the present model, the burner power control is made based on the information about the maximum and minimum working fluid temperatures and on the natural gas mass flow meter.

The thermal model developed to represent the transient response consists of a system of three-coupled equation, shown in Table 2, solved numerically. Eq. 10a represents an energy balance of a control volume of the natural gas inside pipe, when the pipe in externally contact with vapor. Eq. 10.b shows the overall thermal conductance between the natural gas and working fluid. Eq. 11 represents a transient energy balance of the working fluid plus the material of the vessel. Finally, Eq. 12.a is the same heat balance of Eq. 10.a, but for the combustion hot gas, while Eq. 12.b is the overall conductance between the hot gas and the working fluid.

3.1. Transient Results

Figure 5 shows an example of equipment transient behavior, when it is subjected to a linear raise of the natural gas mass rate. The line crossing the figure is the natural gas mass rate normalized at interval 0-5, where 0 represents no flow and 5 the maximum flow rate. The curve at the bottom of the figure represents the behavior of the three-way valve, which governs the natural gas fraction flowing through the heater, where 0 correspond to all flow by-passing the heater and 1 means that all natural gas flow inside the heater. The curve, called "Natural Gas heater outlet" correspond to the natural gas temperature at end of the serpentine. Finally, the "NG After Pressure Reduction" curve shows the natural gas temperature at point (2) in Fig. 2.

In this example, the burner tries to respond the demand of energy, starting at the burning level one, as shown in the "burner level of power" line. Then, the increasing demand of power makes the burner go to level 2 (see right axis). At this stage, the temperature of working fluid raises very fast, as represented by the first sharp slope of the working fluid temperature curve. When the maximum allowed working fluid temperature is achieved, the burner comes back to level one. Almost thirty minutes later, the level 2 is turned on again and the level 3 is quickly achieved. Again, the burner power level is too high. So, the power is reduced to level 2. The level 3 is the highest burner power level for this simulation. The right side of the vertical line named "Maximum steady state capacity" is the region of the figure where the heating power is less than the power demanded (natural gas heating demand plus the environment losses). At this region, the three-way valve increases the flow to the heater to compensate the decrease in temperature of the working fluid. The working fluid temperature is decreasing until that three-way valve pass all flux through the heater. At this condition, the conductance between the natural gas and the working fluid is a maximum.

After that, the "Maximum Instantaneous Capacity" is achieved, the delivery temperature decrease, and the system fails.

The positive slope of the working fluid temperature curve depends on the difference of the power delivered by the burner and power demanded. This difference tends to reduce by the use of many level burners. However, the negative slope of the working fluid temperature curve depends of the heat capacity of the equipment. Even with a many level burner, the temperature curve slope will depend on the equipment thermal capacity, which in turn is small and do not depends on the burner. Therefore, the many level burners will not solve completely the problem of many on-off cycles.

The thermal resistance between the working fluid and the natural gas flow controls the “Maximum Instantaneous Capacity”. Larger resistances demand more temperature gradients for the same power and vice versa.

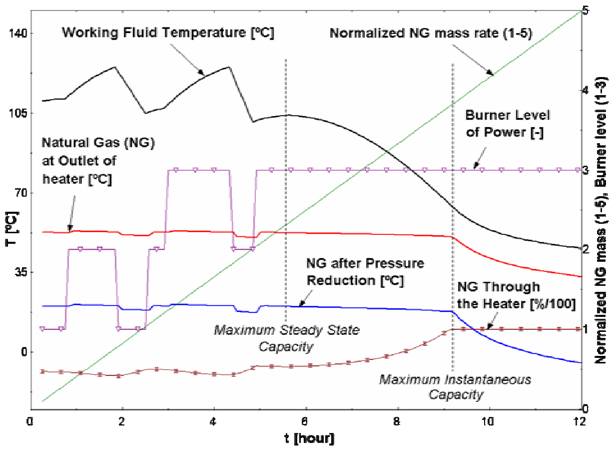


Figure 5: The transient response of temperatures, burner level and percentage of NG passing through the heater for a linear raise of NG mass rate.

4. CONCLUSIONS

The thermal model, which evaluates the effect of many variables in the thermal performance of a natural gas heater assisted by thermosyphon technology, presents reasonable results showing to be a valuable tool for the design of such equipment.

The results presented in this work shows that many discreet power level burners improves but not solve the on-off cycle problems. On the other side, the technology proposed will result in smaller equipment. This is good if one considerers its weight and size. But, small equipment result in small thermal capacity, forcing the burner to turn on and off many times. Such equipment would require more refined control burning power systems.

The choice of the closed two-phase thermosyphon technology instead of the water bath heater solves the problems related to water replenishing and increase the working fluid heat transfer coefficients. However, the pressure difference between the vessel and the environment increases, which demands procedures to avoid leakage.

5. ACKNOWLEDGEMENT

The authors would like to acknowledge the support of PETROBRAS and CNPq during this project.

6. NOMENCLATURE

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