HEAT PIPE AIR HEATERS IN STENTERS FOR TEXTILE INDUSTRY

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Abstract. *Stenter machines are used in textile industry for several purposes, like moisture removing of fabrics, fabric width control, dry-heating process, dry-curing and thermo-fixation of finishing in fabrics. Stenters are the major energy consumer in textile mills. Systems used for heating stenters are based on Steam, gas, circulating of thermal oil, electricity and combined systems. Direct use of flue gases of combustion inside stenters are avoided to prevent stains on fabrics. The high energy consumption of stenters is due to using huge air exhaust mass taxes from the machine while maintaining temperatures in the range of 100-200 ºC. This is to avoiding dry-fabric contamination with outgassing of substances that could also result in stains on fabrics. This work proposes a model for designing hot-air generators with heat recovering of exhaust gases in stenters. Heat pipes are used as a low cost alternative for indirect heating of air and heat recovering of exhaust gases. For calculating heat and mass balances in stenter and hot-air generator, a routine was developed and implemented in EES. This routine also calculates combustion stoichiometry of several fuels. For designing the heat pipe heat exchanger, TROCATER software was used. The design model incorporates a mathematical programming model for design optimization of heat exchangers. With the proposed model is possible to evaluate fuel savings in comparison with conventional equipment, and showed itself a useful tool for the thermal design of this novell hot-air generators for stenters.*

Keywords*: Mathematical programming, optimization, natural gas*

1. INTRODUCTION

Polymeric materials, such as polyester, are thermoplastics, soften itselfs when heated and harden when cooled. During production, the fabrics are subjected to various deformations which cause shrinkage in the process of wetting and washing. In fabric made of synthetic fiber deformation can be reversed by heating the fabric, while kept under dimensional control.

In textile industries, stenters are used to promote drying and dimensional stability (thermofixation) of fabrics. The energy used to dry the fabric corresponds to 50% of energy consumed in the process of finishing the fabric, with 25% of that total going to the stenters (Xuo, 2004). The reason for this high energy consumption is the needing of high air exchange taxes inside the stenter for the elimination of substances released during the process that may compromise the fabric coloration, and in this exhaustion that there are the biggest losses. At exhaustion there is a huge waste of thermal energy, due to discharge of hot gases exhausted, without heat recovery. Energy expended in raw, 60 to 80% is lost in this disposal (Xuo, 2004).

Stenters may have direct heating through electrical resistors or combustion gases blown directly into the fabric, or indirect heating by using hot oil or steam coils inside the system. Resistors are rarely used due to the high cost of electric energy, and direct combustion of fuel on the fabric can cause stains that make it impractical for certain quality standards for fabrics. The main problems with indirect heating are the thermal losses in generating and transporting the hot fluid (oil or steam).

This work proposes a methodology for optimum design of a novell type of indirect heating system for stenters. Part of the gases extracted from the stenter (mainly composed of air) serves as a pre-heated burning oxidizer to a combustion chamber, as shown in Fig. 1.

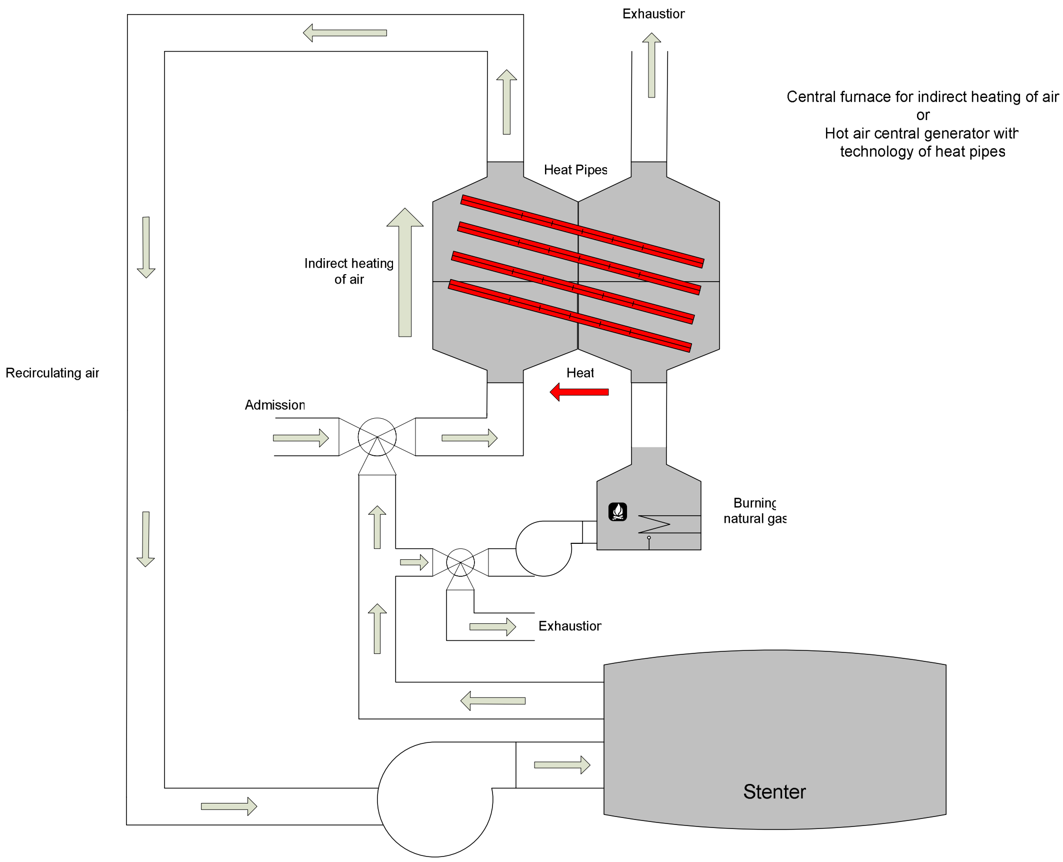

Figure 1. Proposed indirect heating system for stenters with pre-heating of combustion oxidizer

Combustion gases transfer heat to recirculation gases by means of a heat pipe heat exchanger. A system of ducts should be used to supplement the partial recirculation of air. The proposed lay-out for the equipment is shown on Fig. 2.

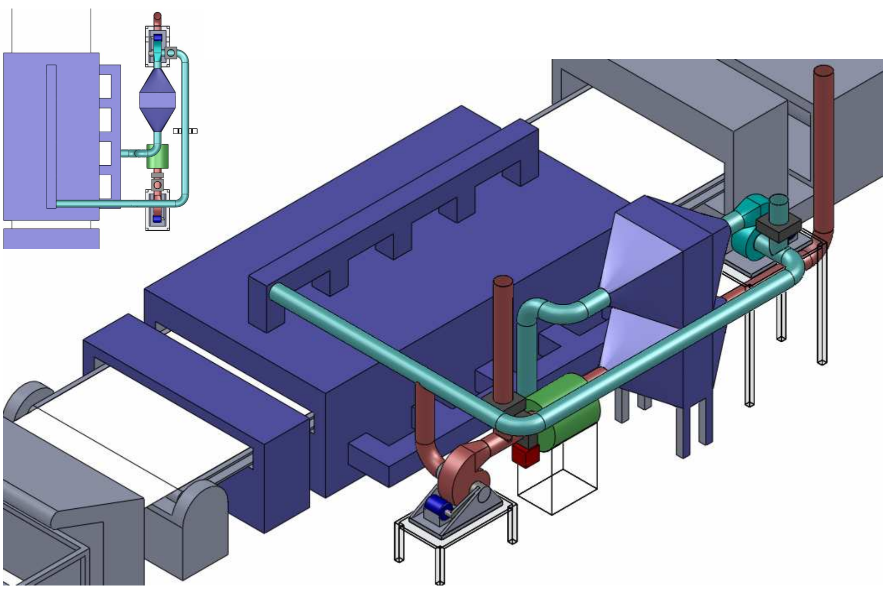

Figure 2. Layout of the proposed air heating system for stenters

Using the indirect burning system the heat is transferred indirectly to the air that heats the fabric, so the particles resulting from combustion does not affect the final product. The combustion chamber uses the stenter exhaustion air as oxidizer, minimizing energy waste.

2. THE SELECTED CASE

Based on an energy diagnosis of Stenters of a textile company (Barros *et al.*, 2009) it was chosen one within the equipment characterized in the report for study and a basis for calculations. The chosen stenter (Fig. 3) works with indirect heating where hot oil circulates in an internal heat exchanger to each module of stenter through which the air is blown.

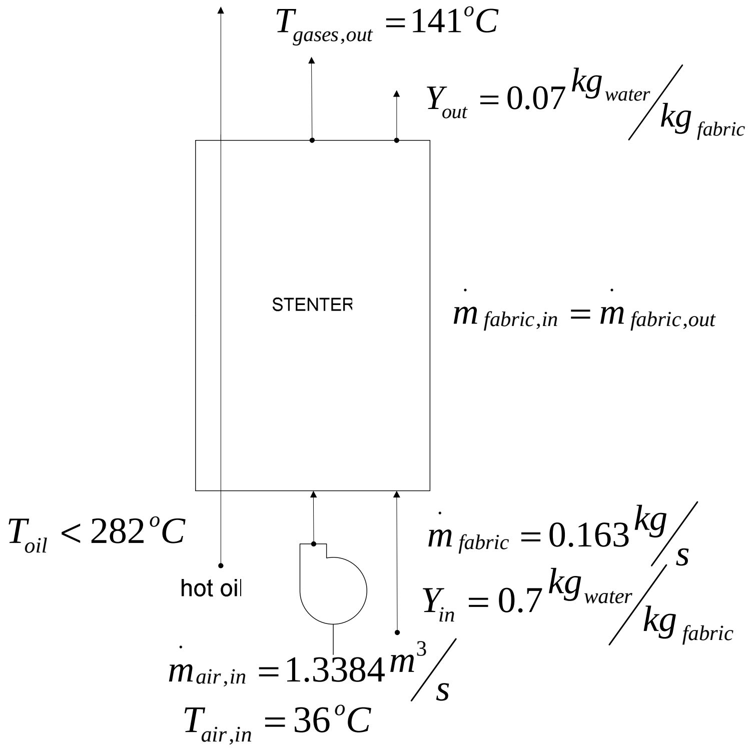

Figure 3. Stenter chosen for case study

Where $\dot{m}_{fabric}$ is the mass flow rate of fabric that come in the stenter, $\dot{m}_{air}$ is the mass flow rate of air, T_{air} is the air temperature, T_{oil} is the oil temperature, T_{gases} is the gases temperature, Y is the ratio between water mass and fabric mass.

Air is blown through of 16 blowers two positioned on each module the same side, with a potency of 10 cv each. While the exhaust is made for 2 fans of 15 cv. Figure 3 shows the values of mass and temperature of the operating day of stenter 8 used for the calculation of mass and energy balance.

The main thermal loads that exist in the Stenters are heating air which is replaced with exhaustion (sensible heat) and evaporation of water in fabric (latent heat).

Based on data obtained (Fig. 3) it evaporates 0.103 kg/s of water from the fabric along its path through the interior of total stenter. This results in evaporation of a heat load of 232 kW. The heat load of heating of fresh air is 157.4 kW. Therefore, the energy demand for water evaporation from the fabric represents 60% of the heat load of the stenter 8.

Considering the need to evaporate water from the fabric, was propose a concept of heating of gases of stenter with heat pipes where there is a recirculation of gases. In each module, there would be a gas collection and return to the heater, with only partial renovation. So, the mass flow rate of gas recirculation may be different from the renovation flow rate. Without recirculation, the necessity of hot gases to evaporate water from the fabric would impose an excessive renovation of air in the stenter.

3. MODELING OF GAS RECIRCULATION SYSTEM

With the objective to evaluate the oxygen content of the oxidizer gas that goes to the firing system, a mass and energy balances was implemented for modeling the heating system proposed pipeline and a stoichiometric balance for the burner in the software EES (Engineering Equation Solver), based on a regime operation of the recirculation gas system, according to operating conditions reported by Barros *et al.* (2009).

To be feasible to install the proposed system the oxygen contained in the gases extracted from the stenter must be the minimum acceptable or higher, for that occur the combustion of fuel in the chamber, and still, the water concentration should be small, because the flame could be muffled. But, the gas temperature in these pipelines will be above 100°C at any point, what will move by pipeline is a mixture of atmospheric air and superheated steam. Thus, there is no psychometric limitation of the water concentration in the gas mixture, and could be 100% water vapor, there was no need of breathe air at the stenter and a minimum content of oxygen in the gas mixture.

Aiming to evaluate the oxygen content of the fuel that goes to the firing system was implemented a balance of mass and energy to the proposed pipeline in the ESS software. The interface input and output of the program can be seen in Fig. 4.

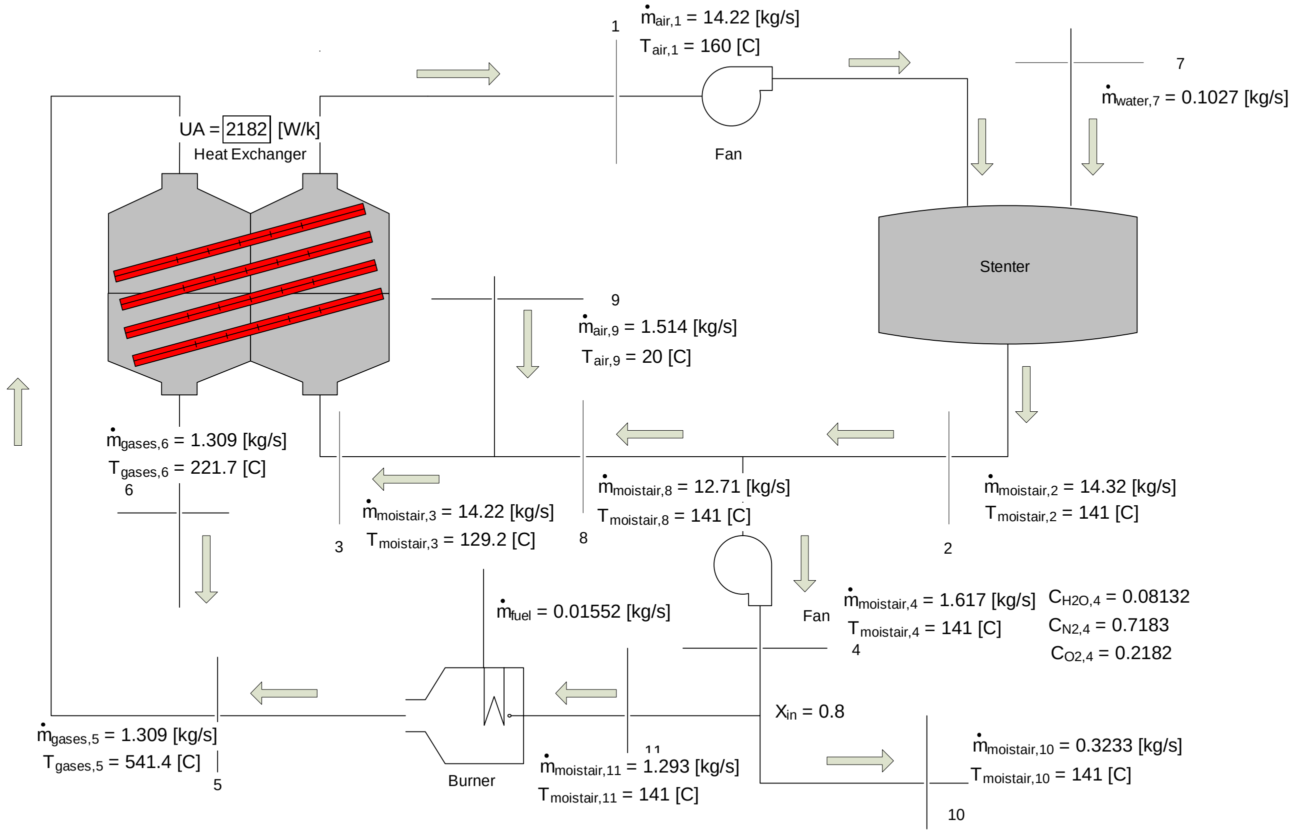

Figure 4. Interface input and output of the ESS program to calculate the balance of mass and energy

Where $\dot{m}_{moistair}$, $\dot{m}_{water}$, $\dot{m}_{gases}$ and $\dot{m}_{air}$ is the mass flow rate of moist air, water, gases and air, respectively, in each defined point, T_{air}, $T_{moistair}$ and T_{gases} is the temperature of air, moist air and burned gases, respectively, in each defined point, $\dot{m}_{fuel}$ is the mass flow rate of combustible necessary in the burner, X_{in} is the ratio between point 11 and 4 of the mass flow of moist air, UA is the global coefficient of heat transfer, $C_{H2O,4}$, $C_{N2,4}$ and $C_{O2,4}$ are the concentration of water nitrogen and oxygen, respectively, in the air at the point 4.

The initial data of the program are the input and output temperatures of the stenter, as well as the inlet temperature of air, water inlet temperature and ambient temperature. Moreover, the mass of evaporated water and air renovation are known.

It all begins getting to the raw air at 160°C (point 1) gathering with the humidity of the fabric (point 7) and this mix is obtained in point 2 which is the process that already happens today. Below, the equations for the mass and energy balance in the stenter.

$$\dot{m}_{moistair,2} = \dot{m}_{air,1} + \dot{m}_{water,7} \tag{1}$$

$$\dot{m}_{air,1}.cp_{air}.(T_{air,1} - T_{moistair,2}) = \dot{m}_{water,7}.cp_{water}.(100 - T_{water,7}) + \dot{m}_{water,7}.2257.2 + \dot{m}_{water,7}.cp_{steam}.(T_{moistair,2} - 100) \tag{2}$$

Where cp_{air} is the heat specific of the air, cp_{water} is the heat specific of the water, cp_{steam} is the heat specific of the steam. The calculation of energy balance has a first portion of the air, the second portion to the heating of water, the third for the phase transition of the same and finally the term that corresponds to the superheated water vapor.

After point 2 there is a bifurcation control (which may be through ventilators, dampers, variable frequency) dividing the flow between 4 and 8, the latter corresponds to the stenter air recirculation, where there is admission of air (point 9) in the system, lowering a little the temperature of the mixture and renewing it, then going to the exchanger and this will restart the cycle. The other side will go to the burner, and then also for the exchanger and thereby heating the mass coming of point 3 (mixture of 8 and 9).

$$\dot{m}_{moistair,2} = \dot{m}_{moistair,8} + \dot{m}_{moistair,4} \tag{3}$$

$$\dot{m}_{moistair,10} = \dot{m}_{moistair,4} + \dot{m}_{moistair,11} \tag{4}$$

$$\dot{m}_{moistair,3} = \dot{m}_{air,9} + \dot{m}_{moistair,8} \tag{5}$$

$$T_{moistair,3} = \frac{T_{moistair,8}.\dot{m}_{moistair,8} + T_{air,9}.\dot{m}_{air,9}}{\dot{m}_{moistair,3}} \tag{6}$$

The burner placed would need a potency of 827 kW, according to the calculations made to supply the demand of heating required for the exchanger. This is due to low flow. A relief valve was placed immediately after point 4, to try to increase the delta temperature and thereby improve the efficiency of the exchanger, and this valve regulated to allow a passage of 80% of the fluid to the burner.

$$\dot{m}_{gases,5} = \dot{m}_{moistair,11} + \dot{m}_{fuel} \tag{7}$$

$$\dot{m}_{moistair,11} = X_{in}.\dot{m}_{moistair,4} \tag{8}$$

Fuel consumption was calculated based on an Inferior Heat Power equal to 53,338 kJ/kg, being spent, 0.01552 kg/s of natural gas. The program calculated the oxygen concentration in the gases supplied as oxidant to the combustion chamber, and a 400% of equivalent excess air was found, which is sufficient for the combustion process. A plausible explanation is the large quantity of air aspirated in the stenter, causing the mass of evaporated water in the mixture negligible.

It was already included in the program the use of heat pipes, where the total conductance of the exchanger was calculated using the computer software TROCATER (Borges *et al.*, 2007) this also helps in sizing the exchanger which will be seen later.

The heat exchanger it's corresponding to has to be equal in the three equations below. The first takes into account the total conductance of the exchanger and the logarithmic temperature difference, and the other two take into account the energy lost and won respectively by the fluids in question, resulting in an exchange of 445 kW.

$$Q_{troca} = \frac{UA.\Delta Tml}{1000} \tag{9}$$

$$Q_{troca} = \dot{m}_{gases,5}.cp_{gases}.(T_{gases,5} - T_{gases,6}) \tag{10}$$

$$Q_{troca} = \dot{m}_{air,1}.cp_{air}.(T_{air,1} - T_{moistair,3}) \tag{11}$$

Where cp_{gases} is the heat specific of the burned gases, ΔTml is the logarithmic mean temperature difference between the points 5, 6, 1 and 3, Q_{troca} is the heat transfered in the heat exchanger.

4. SYSTEM OPTIMAL SIZE

Supposing the supply of gas burner at 500°C, it is intended to dimension a heat exchanger with heat pipes that would meet the thermal load on the desired temperature regime established. For this, the program will be used TROCATER software (Borges *et al.*, 2007) for design of heat exchangers with thermosyphon that uses mathematical programming techniques for optimization. It is proposed that an initial geometry of the exchanger (tube sizes, width and height of equipment and others) and, so, is calculated the number of pipes necessary to obtain the global coefficient of heat exchange desired, and the resulting dimensions of the equipment. Through the program, will be possible develop a design optimized for the equipment.

The dimensions of the delivery system of hot air involved selecting a power burner and a heat exchanger that could meet the same heat load of heating that is now used by circulating hot oil in the stenter. To dimension the equipment it was used the design code TROCATER (Borges *et al.*, 2007). Some parameters were chosen based on previous experience of the team and people involved in the project as shown in Tab. 1.

Table 1. Values proposed for the variable decision.

Ratio between evaporator length and total length	Diameter of pipe [mm]	Ratio between spacing and diameter of pipe	Ratio of fin maximum height	Space between fin [mm]	Thickness of fin [mm]	Width [m]
0.1	25.4	1.5	0.2	3	1.2	0.5
0.16666	38.1	1.8	0.4	4	1.5	0.75
0.2		2	0.6	5	2	1
0.3		2.16	0.8	6		1.5
		2.5	1			

For the arrangement of the tube bundle, were chosen the values 30, 45, 60 and 90 degrees. For the heat pipe operating temperature of the working fluid above 300°C, provides for the use of naphthalene. For lower temperatures, provides for the use of water.

The technique chosen for optimum synthesis of heat exchanger was exhaustive simulation. So, it was dimensioned configurations of heat exchanger with all possible combinations for the values of the variables above. It was generated 19,200 configurations of heat exchangers. These configurations were tabulated in an Excel spreadsheet. A fragment of this spreadsheet with the monitored results is shown in Tab. 2

Table 2. Fragment of sheet results of exhaustive dimensioning that has 19,200 configurations of heat exchangers

Lev/Ltot	%hmaxfin	S fin [mm]	T fin [mm]	W [m]	Arrang. [Degrees]	Mass equip. [kg]	N tubes	Evap. P.d. [mmH2O]	Cond P.d. [mmH2O]	H.e. L [m]	Re evap.	Re cond.	Max.V. evap. [m/s]	Max.V. cond. [m/s]	P [US$]
2.5	0.4	3	1.2	1.5	30	1965	131	5	32	0.70	2767	14263	4.5	10.9	40565
2.5	0.4	4	1.2	1.5	30	1923	145	5	31	0.78	2707	13950	4.4	10.7	44289
2.5	0.4	4	1.5	1.5	30	2126	145	5	32	0.78	2753	14188	4.5	10.9	44780
2.5	0.4	5	1.2	1.5	30	1865	160	4	29	0.78	2667	13746	4.4	10.5	48252
2.5	0.6	3	1.2	1.5	30	1961	102	5	36	0.53	2956	15235	4.9	11.7	32624
2.5	0.6	4	1.2	1.5	30	1872	116	5	35	0.62	2850	14691	4.7	11.3	36239
2.5	0.6	4	1.5	1.5	30	2114	116	5	36	0.62	2930	15102	4.8	11.6	36823
2.5	0.6	3	1.5	1.5	30	2265	116	6	37	0.53	3054	15740	5.0	12.1	37188
2.16	0.6	3	1.2	1.5	30	1687	123	5	35	0.39	3174	16359	5.2	12.5	37705

Where *Lev/Ltot* is the ratio between evaporator length and tube total length, *%hmaxfin* is the ratio of maximum height fin, *S fin* is the space between fin, *T fin* is the thickness fin, *W* is the width of the heat exchanger, *Arrang.* is the arrangement of tubes, *Mass equip.* is the mass of the equipment, *N tubes* is the number of tubes necessary in the heat exchanger, *Evap. P.d.* is the pressure drop in the evaporator, *Cond P.d.* is the pressure drop in the condenser, *H.e. L* is the length of the heat exchanger, *Re evap.* is the Reynolds number in the evaporator, *Re cond.* is the Reynolds number in the condenser, *Max.V. evap.* is the maximum velocity in the evaporator, *Max.V. cond.* is the maximum velocity in the condenser, *P* is the price of the equipment.

Following the nomenclature used in mathematical programming, each configuration of heat exchanger is called an individual. The objective is to choose individuals with the best overall performance. Following the optimization procedure proposed by Borges *et al.* (2007), where it was made successive cuts in the total list of individuals, based on desirable characteristics of performance:

- Cut to individuals with pressure drop in the condenser above 200 mm H2O, left over 10,000 individuals;

- Cut to individuals with pressure drop in the evaporator above 200 mm H2O, left over 9,245 individuals;
- Following Branan (2002), individuals were cut with a maximum speed in the tube bundle above 24 m/s. 8,397 cases were left in the condenser and the evaporator remained 7,845 cases later;
- Reynolds number less then 2,300 was eliminated in the evaporator to guarantee turbulence, remaining 7,600 individuals;
- There was low Reynolds number in the condenser;

Using the remaining 7,600 individuals, plotted the graph price per pressure drop in the condenser, and shown in Fig. 5. The condenser is the side of the greatest loss.

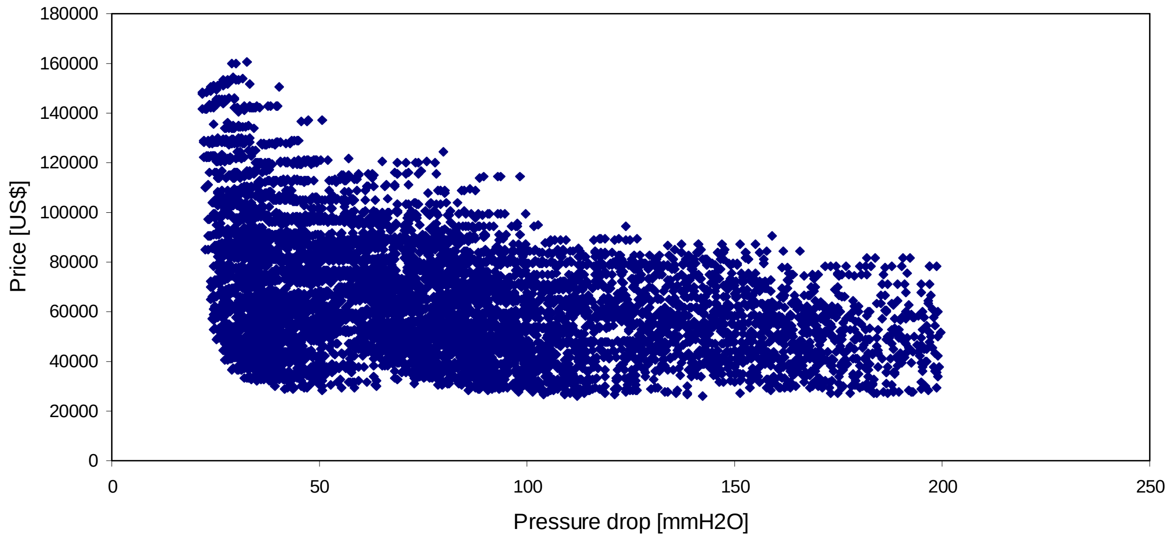

Figure 5. Graphic of price versus pressure drop in the condenser

Following the procedure proposed by Borges *et al.* (2007) is possible to notice that the envelope under the cloud of points from the previous figure is a tradeoff curve (negotiation) between price and pressure drop. The most interesting point of the curve is inflection. To focus on the inflection point we made two cuts: Price greater than US$ 80,000 and pressure drop greater than 75 mm H2O, where it remained 2,473 individuals. To isolate the points near the lower envelope were proposed two trend lines, so that, all points of the envelope were available between these two lines. After this, the remaining points were eliminated, resulting in 306 individuals near the envelope, as shown in Fig. 6.

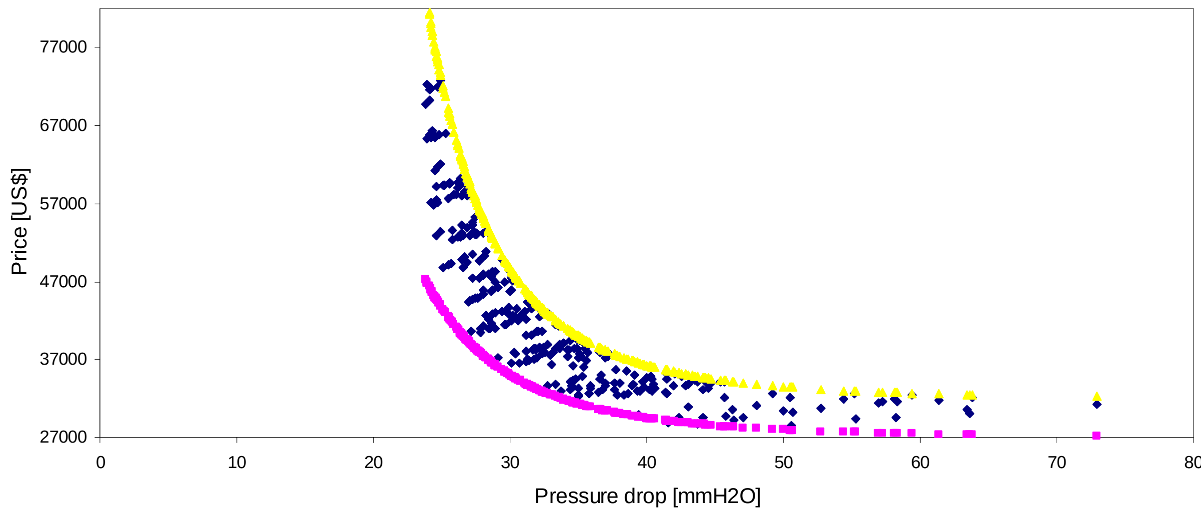

Figure 6. Points near the lower envelope

Only with the selected individuals plotted the graph (Fig. 7) price versus pressure drop in the evaporator.

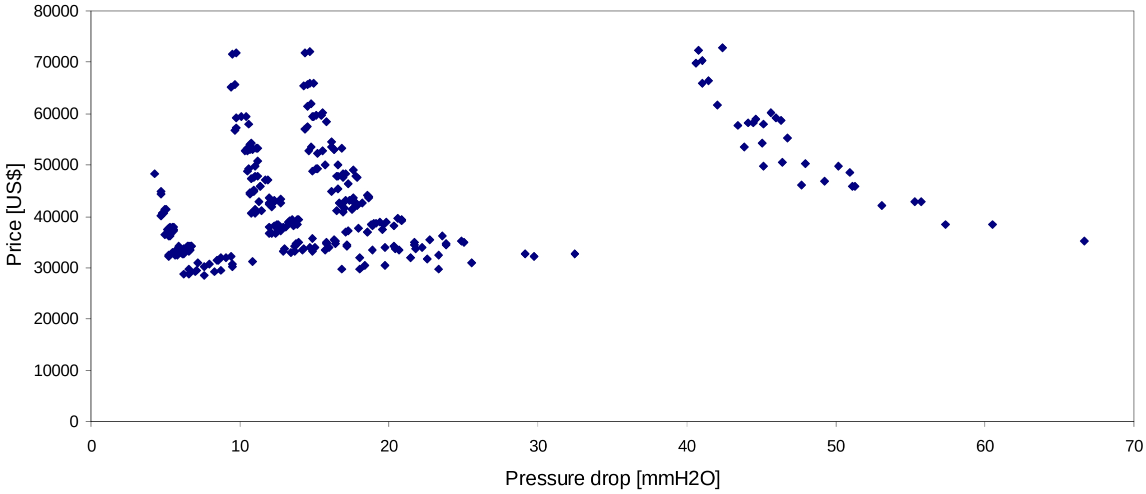

Figure 7. Graphic price versus pressure drop in the evaporator

From this point cloud, we repeated the same procedure used previously to obtain the lower envelope of the cloud of individuals. Finally, remained 76 individuals, there were sorted in ascending order of price.

Based on this tradeoff, it was chosen the individual with the price of US$ 36,097 because it represents a good compromise between price and pressure drop.

Therefore, the final characteristics of the equipment are shown in Tab. 3.

Table 3. Basic characteristics of the heat exchanger dimensioned.

Length of thermosyphon	m	3.01
Relationship between total length and length evaporator		1/3
Diameter of tubes	mm	38.1
Geometric arrangement of the tube bundle	Degrees	30
Number of tubes		116
Number of tubes in odd rows		15
Number of tubes in even rows		14
TOTAL NUMBER OF ROWS		8
Height of fins on the condenser	mm	16.8
Thickness of the fins on the condenser	mm	1.2
Spacing between fins on the condenser	mm	6
Height of fins on the evaporator	mm	16.8
Thickness of the fins on the evaporator	mm	1.2
Spacing between fins on the evaporator	mm	6
Pressure drop on the condenser	mmH2O	29
Pressure drop on the evaporator	mmH2O	3
Reynolds number on the condenser		14201
Reynolds number on the condenser		2336
Maximum velocity on the condenser	m/s	10.8
Maximum velocity on the evaporator	m/s	3.6
Overall width of the exchanger	m	1.5
Total length of the exchanger	m	0.62
Mass of the equipment	kg	2025
Price	US$	36,097
Maximum temperature of steam in thermosyphon	ºC	226
Total Heat Load	kW	445

5. CONCLUSION

As can be seen, this is a compact equipment with a view to its big thermal load. Your external volume would be something a bit larger than 3.1 x 1.5 x 0.62 m. It would has 116 heat pipes with 3 meters long finned in both sides, with 0.9 m evaporator length and 2.1 m condenser length.

This solution only demonstrates the feasibility of the equipment, and even surpassed the supply crisis of Natural Gas in the country, it is important that the technology offered to the industrial sector is perceived as robust and reliable. The development of flexible equipment for natural gas through the heat pipe technology brings to this new consumers the perception of reliability, and allows the establishment of uninterrupted supply contracts, making this a good deal for the industrial consumer.

6. ACNOWLEGDEMENTS

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