ENERGY BALANCE IN A RIJKE TUBE PULSATING COMBUSTOR

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Abstract*. An experimental investigation has been carried out to study the effects of pulsations on the heat flux to the combustor wall. Pulsating combustion increases the amount of heat that is transferred through the combustor walls. This fact is important mainly in boiler processes. This work shows some results obtained by means of energy balance in two configurations of a laboratory Rijke pulsating combustors. The fuels utilised in the experiments were liquefied petroleum gas, ethanol, and charcoal. The results indicated a 20% increase in the heat fluxes compared to the non-pulsating process. A relation between oscillations conditions (amplitude and frequency) and heat transfer rates was also observed.*

Keywords. *Combustion, Pulsating Combustion, Energy Balance.*

1. Introduction

At the moment, fossil fuels provide most of the primary energy for the industrial and emerging industrial nations. As a consequence, pollutants emissions are increasing. Therefore, research on combustion is, nowadays, concerned to the development of new technologies and systems that minimize pollutants emissions and increase the overall efficiency of the process.

Pulsating combustion, due to its unique feature, has been identified as one of the possible technologies that can potentially reach these objectives. The most important characteristics of this process are: reduction in pollutants emissions; high combustion efficiencies, increased in the convective heat transfer. All of these can take place with lower investments (Ferreira and Carvalho, 1988).

When oscillatory combustion takes place, any variable that describes the process (such as pressure, temperature, and velocity) varies periodically with time at every burner location. Some burners are designed to provide acoustics oscillations during the combustion process; for instance, pulsating Rijke and Helmohtz combustors (Carvalho et al., 1989). In such devices, the pulsating condition occurs spontaneously. However, it is difficult to predict and control its occurrence. For a conventional combustion chamber it is necessary to provide external actuators in order to have pulsating combustion; for example, a loudspeaker strategically placed to induce oscillations.

In fact, the spontaneous pulsating phenomena have two important characteristics. The first one is that oscillation is built up without any appreciable external excitation. The second one is that after the oscillation has increased until certain amplitude, it is maintained at this state. Higgins firstly identified this phenomenon in 1777. He observed that if small jet of ignited gas is inserted into a tube opened at both ends, a steady and intense noise, at the natural frequency, is produced for certain conditions.

In 1859, Rijke noted that strong acoustics oscillations were obtained when one heated wire screen was placed in the lower half of an open ended vertical pipe. Figure (1) shows an original Rijke tube and its acoustic wave structure. This type of pulse combustor has been used extensively to study several aspects of oscillatory combustion with different fuels, for example: wood [Zinn et al, 1982; Carvalho et al, 1989]; coal [Miller et al, 1982; Carvalho et al, 1984 and Carvalho et al 1987]; charcoal [Ferreira et al, 1992]; agriculture residue [Torres et al., 1992]; and ethyl alcohol [Dubey et al, 1997, Lacava et al, 1997 and McQuay et al., 2000]. The Rijke oscillations occur in a pipe with both ends open.

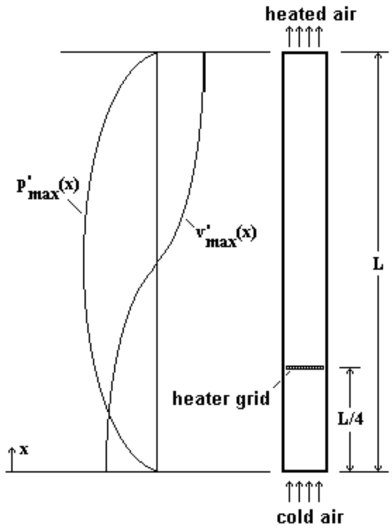


Figure 1 – Rijke Tube with open ends and the acoustic wave structure.

In 1879 Rayleigh observed that Rijke tube would not oscillate if the heated gauze were located in the upper half of the tube. On the other hand, tones higher than the fundamental could be excited for certain gauze location. He explained the observation, on the self sustained vibration, stating that if heat is periodically added and subtracted to the flowing air at some specific moment, it could be excited or damped. Rayleight did not presented a mathematical proof for such observations. Putnam and Dennis (caput, Couto, 1989) could state, mathematically, Rayleigh´s criterion in a precise form. If damping forces are neglected, Rayleigh´s criterion is written as follows,

$$\oint Q' \, p' \, dt > 0 \, , \tag{1}$$

where Q' is instantaneous heat rate release to stream, p' is difference between mean and instantaneous pressure value (acoustic pressure), and t is the time integral over an oscillation cycle. This criterion establishes the condition under which a small disturbance is amplified in the course of time by its interaction with a heat source. It is important to observe that the Rayleigh criterion is a necessary but not sufficient condition for presence of oscillation. Experimental observations have shown that oscillation can be initiated by the interaction between aerodynamics properties of the flow and the heat source (Ferreira, 1989).

Carvalho et al. (1989) have presented an analysis defining the heater location that drives the maximum amplitude of acoustic oscillations in a Rijke tube. This analysis is based in the mathematical formulation of the Rayleigh criterion. For a Rijke-tube of length L, the maximum amplitude oscillation was obtained in several positions. The results were experimentally checked and good agreement was observed.

In the Rijke combustors, a bed if burning solid fuels, an atomizer for liquid fuels and a gas injector for gas fuels replace the original metallic gauze. The possibility of operating with different kinds of fuels has turned this combustor versatile. As a consequence, several research works can be seen in the field of pulsating combustion.

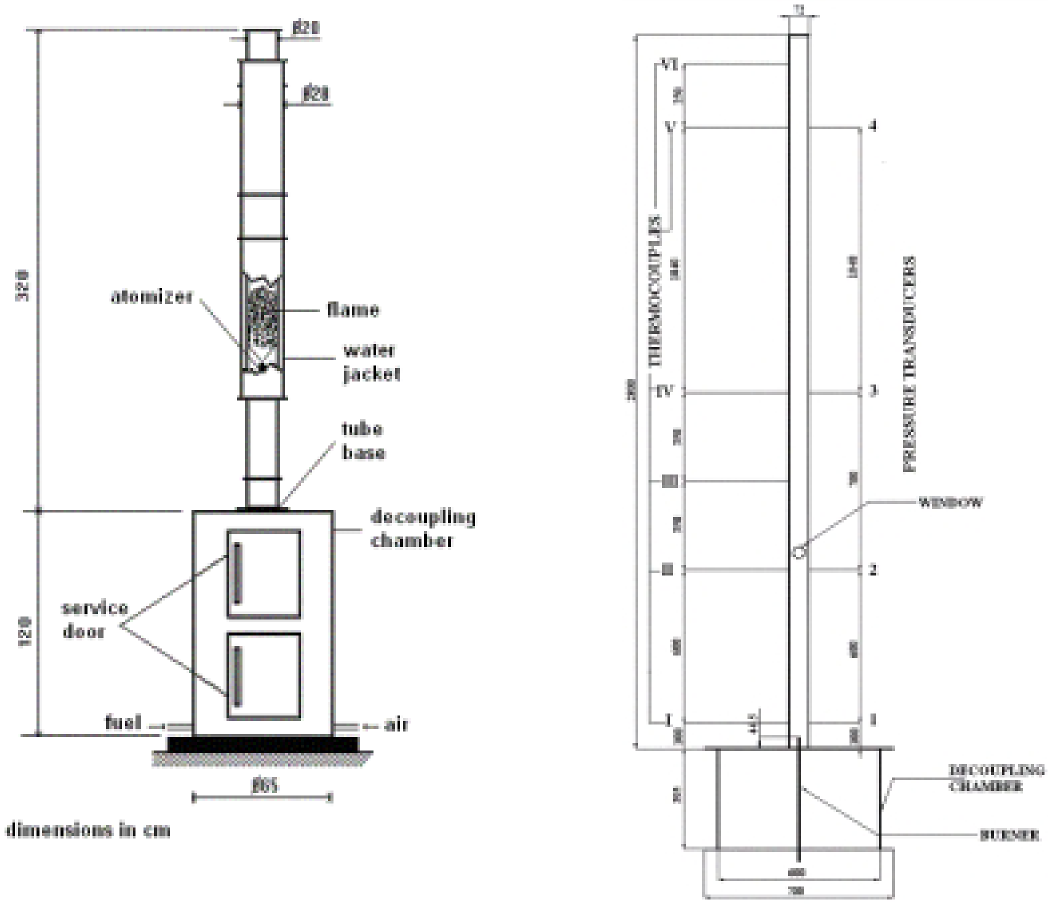
This work presents an energy balance in laboratorial scale Rijke-tubes combustors in which liquefied petroleum gas (LPG), ethanol, and charcoal were used as the burning fuel. The main objectives were to quantifier the heat transferred to combustors internal wall when the combustion is oscillatory and non-oscillatory. Also, the influence of frequency and amplitude pressure wave on the heat transfer process was analysed.

2. Experimental setup

As mentioned before, the experiment was conducted with liquefied petroleum gas (LPG), ethanol, and charcoal as the fuel. The experiments with liquid and solid fuels were conducted in the Combustor 1, with 0.2 m internal diameter and variable length (2.4, 2.8 and 3.2 m). For the LPG tests, the combustor had 7.2 cm internal diameter and fixed length of the 2.8 m. This second device (Combustor 2) operated with LPG. It had an improvement in the collecting data system over Combustor 1. Figure (2) shows schematically both combustors.

Decoupling chambers were placed at the bottom end of the combustors for better control of the combustion air mass flow rate. This technique is important to keep the required open-end boundary condition at the tube entrance. In both Rijke tube combustors a water-jacketed was provided to allow the heat transfer analysis by water mass flow rate and the temperature difference between the jacket entrance and jacket inlet.

Combustor 1 Combustor 2



dimensions in mm

Figure 2 - Schematic diagrams for both Rijke tube combustors.

The fuel gas (LPG) used in this study is a mixture of hydrocarbons. Its formula can be taken as $C_{4.25}H_{7.824}$, with a molecular weight of 49 g/gmol. LPG and air mass flow rates were measured with orifice plates. The burner main body for gas fuel had a nominal diameter of 1.2 cm. Its length was nearly 3 cm, which could be increased by the addition of segments. The LPG injector had six 1-mm diameter holes.

For ethanol injection (Combustor 2) a commercial air-assisted solid cone atomizer was utilised. Ethanol mass flow rates were measured with a rotameter, and atomisation air and combustion air mass flow rates were measured with orifice plates.

In the case of charcoal, a metallic wire grid attached to a steel ring held the combustion bed. The grid was supported by a sliding rod mechanism that allows variation of the combustion bed position inside the combustor. A system, which consisted of a perforated cylinder attached to a motor of variable speed, was responsible for fuel feeding. The combustion rate was controlled by the motor speed. The air mass flow rates were measured by orifice plate and a CO/CO infrared analyser and an O_2 thermomagnetic analyser determined the combustion conditions (air-fuel ratio).

In all the experiments, the fuel-feeding device (gas injector for LPG, atomizer for ethanol, and grid for charcoal) was positioned near L/4, where L is the tube length. At this location, it is possible to get maximum value of the cyclic integral and the maximum amplitude acoustic oscillation, as pointed by Carvalho et al. (1989).

For both combustors, temperature and acoustic pressure were measured with K-type thermocouples and piezoelectric pressure transducers (amplitude and frequency measurements), respectively. The transducers were individually linked to charge amplifiers. Both thermocouple and pressure transducer signals were stored and analysed by a computerised data system.

In Combustor 1 the thermocouples were positioned at 25 cm (before the flame), 150 cm (close central tube region) and 308 cm (close output region) from the tube inlet section. One pressure transducer was placed at 160 cm above bottom, or at the combustor half-length, where the pressure oscillation amplitude is higher. The others transducers were placed at 15 cm and 80 cm, from the tube inlet section. In the combustor 2 the thermocouples were placed at 100 mm, 700 mm, 1050 mm, 1400 mm, 2440 mm and 2690 mm upper bottom, as shown in Fig. (2). Also, pressure transducers were positioned in 100 mm, 700 mm, 1400 mm and 2440 mm.

3. Results and discussion

Experiments were conducted to quantify the total heat transfer process from the flame and burned gases to combustor internal wall. To quantify experimentally the kind of heat transfer process (radiation, convection, or conduction) is a very complex task, due the intense modifications in the physical and chemical processes that take place in the presence of an acoustic field. For example, changes in soot formation, with lower heat transferred by radiation, or increase in the mass transport (more heat transferred by convection). Therefore, this experimental study is concerned to the total heat transferred to the wall. The heat transferred to the wall is calculated by energy and mass balance of the water flow.

The equivalence ratio, in all graphs, is the defined as

$$\left(\frac{\dot{m}_F}{\dot{m}_{AIR}}\right) \Bigg/ \left(\frac{\dot{m}_F}{\dot{m}_{AIR}}\right)_{STOIC}, \qquad (2)$$

where $\dot{m}_{AIR}$ is the air mass flow rate, $\dot{m}_F$ is the fuel mass flow rate and $\left(\dot{m}_F / m_{AIR}\right)_{STOIC}$ is the stoichiometric fuel/air ratio.

3.1 Charcoal (Combustor 1)

In the following experiments the combustor (diameter 0.2 m) was tested with lengths of 240 and 320 cm, and the fuel used in the experiments was charcoal, particle size 2-4 mm. Heat transfer rates are presented for the two investigated lengths. The observed oscillation frequencies were approximately 90 and 70 Hz, respectively. The heat transfer to the wall increased in the presence of oscillations and was higher for the 320 cm combustor. In order to compare the influence of the frequency, heat transfer was measured in the same area, as shown in Fig. (3) and Fig. (4). Tables (1) and (2) present the equivalence ratio, pressure amplitude and nominal power in all the conditions measured during experiments. It can be observed that in most of the tests the pressure amplitude was higher at the grow air mass flow rates and did not depend on the combustor length. The acoustic pressure decreased, along the experiment, in all cases. Probably, this is due to transient effects, during the experiments. As the combustor became hotter the location of combustion bed position, for maximum acoustic amplification, shifted downwards from the initial quarter length position.

Comparisons between the experiments are presented in Fig. (3) and Fig. (4). For the same heat transfer area (combustion bed modulus), a maximization in heat transfer occurred for the longest residence time.

The experiments show strong evidence that heat transfer to the wall of combustor is favoured by the acoustic oscillations. For the same pressure amplitude, the effect is more significant when operating with the 320 cm combustor and frequencies of the order of 70 Hz.

Table 1. Pressure amplitude for different operating conditions with combustor length to the 320 cm

Nominal power (kW)	Equivalence ratio	Lower pressure amplitude (mBar)	Higher pressure amplitude (mBar)
34.4	0.73	15	20
39.1	0.83	14	20
41.8	0.88	16	22
48.1	0.76	15	20
52.4	0.83	17	25
57.1	0.91	20	27

Table 2. Pressure amplitude for different operating conditions with Combustor Length to the 240 cm

Nominal power (kW)	Equivalence ratio	Lower pressure amplitude (mBar)	Higher pressure amplitude (mBar)
38.1	0.75	16	20
43.9	0.88	15	18
48.7	0.97	15	18
51.3	1.03	18	20
59.8	0.89	20	24
61.9	0.91	20	24
61.4	0.92	22	28

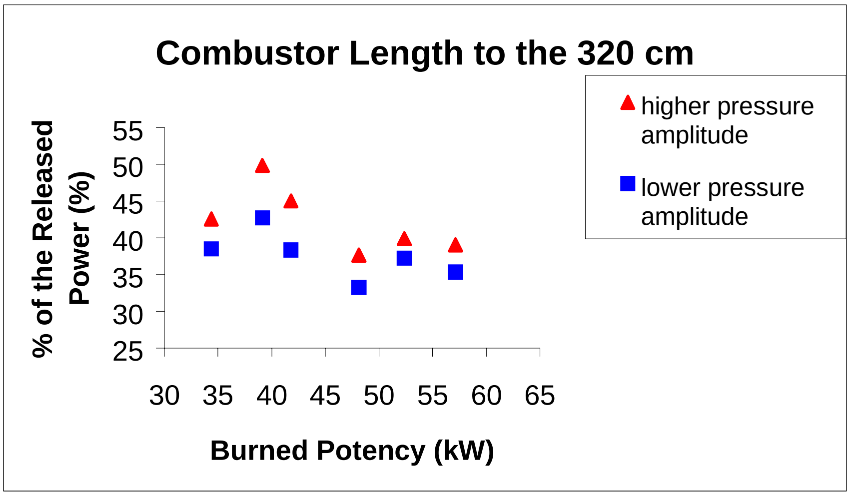


Figure 3: Percentage of heat transferred to water in a burned fixed section (Combustor length 320 cm).

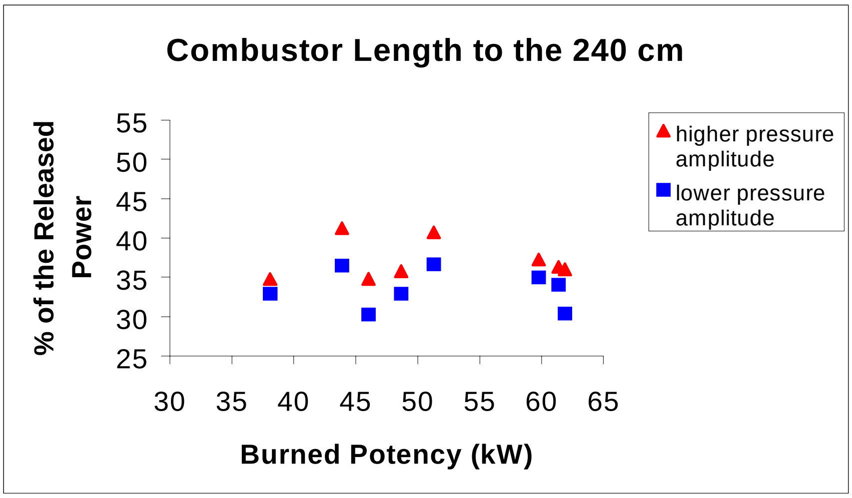


Figure 4: Percentage of heat transferred to the water in a burned fixed section (Combustor length 240 cm).

3.2 Ethanol (Combustor 1)

Table 3 shows the experimental conditions under which oscillations were obtained using the solid-cone atomizer. These conditions are characterized in terms of combustion air (m_{comb}), fuel mass flow rate (m_f), and atomising mass flow rate (m_{atm}). I this table P_0 is the operating pressure of the atomising air, r_1 is the ratio m_{comb}/m_f, and $T_1 - T_3$ and $P_1 - P_3$ are the average temperature and pressure, respectively. The frequency of oscillation was 73 Hz in all the tests. The acoustic pressure amplitudes ($P_1 - P_3$) are average values, taking from the time-resolved pressure.

Figure 5 shows the energy balance for the testes described in Tab. (3) (all results are under oscillations). For rich combustion, the percentage of energy released by the combustion reactions transferred to the wall (Hw) increases with the increase of total air mass flow rate. From Tab. (3), it is observed that the overall average pressure amplitude increased with increasing the combustion air mass flow rate, with fuel mass flow rate is kept constant. This is because the flame length decreased as the oxidiser flow rate increased (visual observation) by virtue of the better reactants mixing due to higher turbulence levels. These effects concentrate the heat release in the lower half of the tube, intensifying the oscillations amplitude, as expected by the Rayleigh criterion (Eq. 1).

For a generic lean combustion, without oscillations, when the oxidiser mass flow rate increases it is expected that less heat is transferred to the combustor walls, since; (i) more energy is concentrated in combustion gases due the higher presence of inert gases; (ii) the total mass flow rate increases and, as consequence, the residence time (or heat transfer time) decreases; (iii) more oxidiser and higher turbulence levels in the flame region reduce the soot presence, and the heat transfer rate by radiation decreases. However, as pointed in Fig. (5), for pulsating combustion with equivalence ratio higher than unit, the relation of total energy transferred to wall and the energy retained in the output gases practically did not change with the increase of oxidiser mass flow rate. Basically, the three statements discussed before reduce the heat transfer. However, higher Reynolds number and the increment of acoustic pressure amplitude have a tendency to increase the convective heat transfer rate, thus compensating possible reduction in the heat transfer. This trend was observed only for certain limiting value of air flow rate, above which the energy expend to heat the excess of air became significant, and less energy is available to sustain oscillations. In Fig. (5) only the cases where oscillation occurred are plotted.

Table 3. Experimental conditions under oscillations of the fundamental mode were obtained.

Test number	m_f (g/s)	m_{comb} (g/s)	m_{atm} (g/s)	P_0 (kPa)	r_1	r_2	T_1 (ºC)	T_2 (ºC)	T_3 (ºC)	P_1 (kPa)	P_2 (kPa)	P_3 (kPa)
1	2.42	25.51	0.302	35	10.5	8.0	24.5	802	584	0.940	1.272	1.242
2	2.42	27.53	0.303	35	11.4	8.0	25	793	589	1.062	1.470	1.442
3	2.42	29.41	0.302	35	12.1	8.0	25	770	590	1.194	1.642	1.620
4	2.69	24.30	0.308	40	9.0	8.7	41	879	687	0.830	1.067	1.107
5	2.69	26.52	0.310	40	9.9	8.7	42	868	688	0.951	1.338	1.308
6	2.69	28.10	0.305	40	10.4	8.8	42	858	692	1.087	1.539	1.508
7	2.69	31.77	0.308	40	11.9	8.7	44	830	689	1.209	1.715	1.676
8	3.22	24.57	0.322	45	7.6	10.0	38	890	717	0.943	1.300	1.239
9	3.22	29.15	0.315	45	9.1	10.2	41	896	729	1.088	1.500	1.448
10	3.22	31.00	0.318	45	9.6	10.1	39.5	877	725	1.211	1.698	1.643
11	3.49	23.82	0.346	51	6.7	10.1	24	856	587	0.868	1.200	1.155
12	3.49	26.36	0.345	51	7.6	10.1	24	865	663	1.199	1.459	1.510
13	3.49	29.79	0.344	51	8.5	10.1	24	860	665	1.231	1.687	1.619

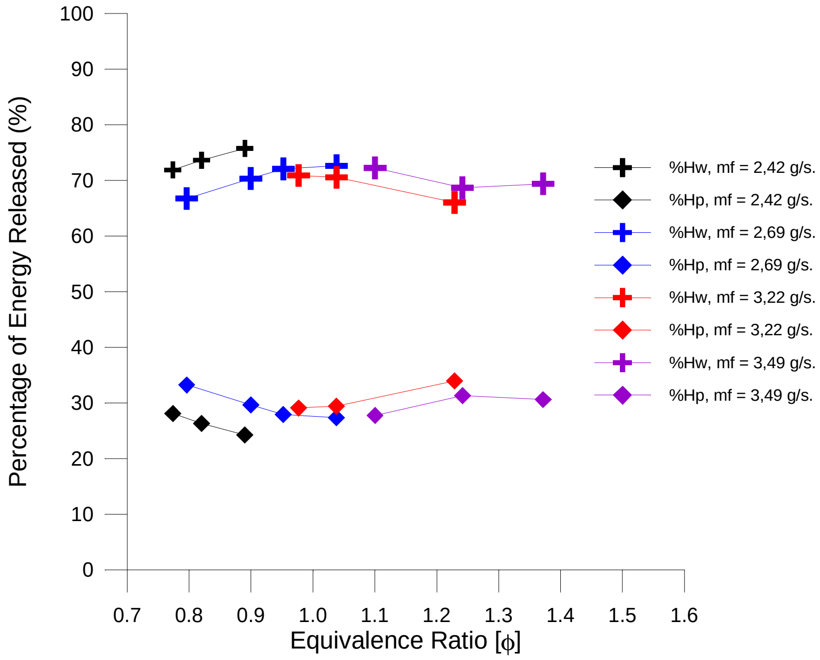


Figure 5 - Results of the energy balance for the testes described in Tab. (3)[1].

3.3 LPG (Combustor 2)

A description of the experiments operated with LPG follows in this section. The goal was to obtain a quantitative comparison between steady and pulsed jet to the convective heat transfer.

Fluid mixing is the controlling factor for a wide class of non-premixed flames (Bilger, 1976, Faeth and Samuelson, 1986). In such flames, the chemical reaction occurs rapidly compared to turbulent mixing process. As consequence fluid mechanics alone govern global parameters, such as flame length. Pulsating intensifies the local turbulent mixing and the fuel consumption near the jet exit. The enhancement in the mixing and combustion regimes reduces the overall length of the jet flame.

The LPG mass flow rates were fixed in 0.15 and 0.30 g/s, corresponding to total power on the order of 6.9 and 13.8 kW, respectively. Cooling water flowed through the jackets (between the inner and outer tubes) entering at the

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[1] *Note - Hw is percentage of energy released by the combustion reactions that is transferred to the wall, and Hp is percentage of energy released by the combustion reactions that is contained in the output combustion gases.*

decoupling chamber end and exiting at the section end tube. The cooling water flow was kept at 15 l/min. The percentage of the energy fuel content transferred to the cooling water is presented in Fig. (6) and Fig. (7), for both combustion regimes. The values were calculated from the assessment of the water inlet and outlet temperatures and mass flow rates.

The residence time varied between 1.3 to 2.0 s for fuel mass rate of 0.15 g/s, which is sufficient for complete combustion. At 0.30 g/s the residence time varied between 0.5 and 0.8 s, which is not sufficient to complete the combustion reactions despite the shortening in time for heat transfer to walls.

Comparing the results presented in Fig. (6) and (7) it is possible to observe the influence of residence time. The percentage of the power release when the fuel rate was 0.15 g/s are greater than the 0.30 g/s fuel rate, independent of the regime combustion, pulsating or non—pulsating.

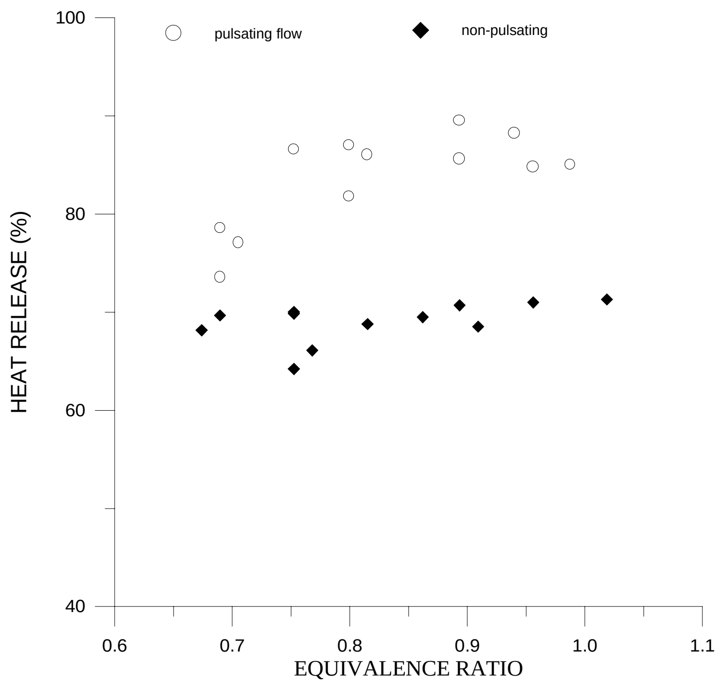

Figure 6. Percentage of the heat release to the water (fuel rate of the 0.15g/s)

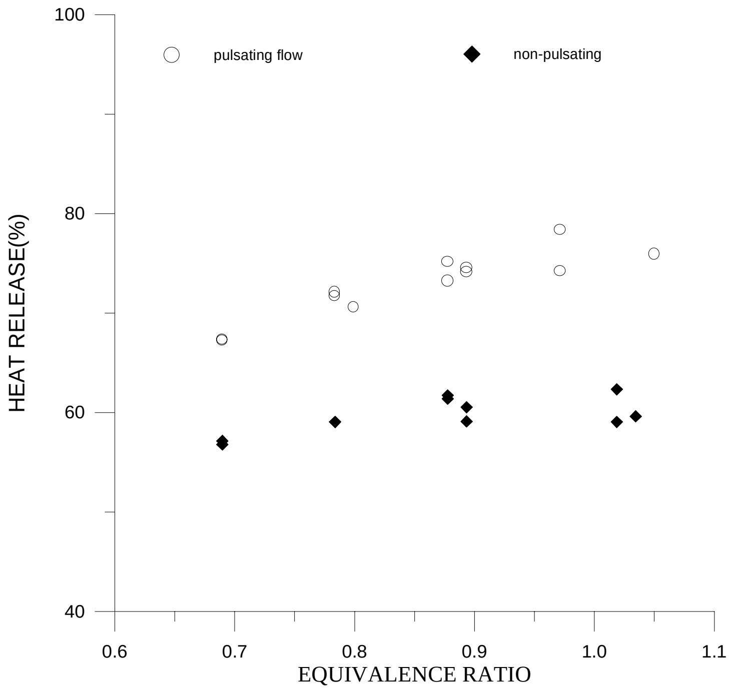

Figure 7. Percentage of the heat release to the water (fuel rate of the 0.30 g/s)

In Tab. (4) and Tab. (5) it can be seen the comparison of the heat released between pulsating and non-pulsating process for fuel mass flow rate of 0.15g/s.

Table 4. Comparison between pulsating and non-pulsating regimes (fuel mass flow rate = 0,15 g/s).

Equivalence ratio	Increased heat transfer to water jacket (%) [pulsating – non-pulsating] / [(non pulsating)]
1,0	19,1
0,9	26,3
0,8	25,7
0,7	10,7

Table 5. Comparison between pulsating and non-pulsating regimes (fuel mass flow rate = 0,15 g/s).

Equivalence ratio	Increased heat transfer to water jacket (%) [pulsating – non-pulsating] / [(non pulsating)]
1,0	26,8
0,9	22,2
0,8	21,3
0,7	17,9

4. Conclusions

The objective of the experiments with charcoal, ethanol and LPG fuel was to verify the enhancement in the heat transfer to the cooling water in oscillatory combustion. The following conclusions can be drawn from these experiments:

1. With charcoal the heat transfer to jacket is higher at 70 Hz than 90 Hz. Also, heat loss to water improves as amplitude grows. In oscillatory conditions the solid fuel behaves as fluidized bed, where particles are constantly removing the superficial ash layer. This exposes the material in contact with air which favouring the heat transfer by radiation and convection.

2. With liquid and gas fuels the pressure amplitude increased with increasing of the combustion air flow rate, at constant fuel flow rate. This is because the flame length decreased as the oxidiser flow rate increased.

3. Acoustic driving would be enhanced by the large variation in flame area and hence, in the heat release rate. In oscillatory conditions the flame is shorter, bluish and without soot. On the other hand, flames in a quiescent atmosphere are longer, yellowish and sooty. This is an evidence of the change in mixing rate. With pulsation combustion the combustor can be though as a stirred reactor.

4. Convective heat transfer in oscillating flow can be much higher than those of steady turbulent flow at the same mean Reynolds number as observed in LPG experiments. Experimental measurements demonstrate that flow rate oscillation can enhance the heat transfer by a factor of 20%. This reduces the size and cost of the system.

5. In Rijke tube combustion the predominance on heat transfer process is convection because it is an environment adverse for radiation mode. The combustor diameter is very small compared to the length; so that the mean free path for radiation is very low, which makes convection the paramount way for heat transfer, and as consequence improvement in this mechanism has an important impact over the whole heat transferred. However, in any combustion device the presence of pulsating condition will change both convection and radiation modes; than the total heat transferred to the wall will depend on how the pulsating conditions change these modes.

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