ISSN 2176-5480

22nd International Congress of Mechanical Engineering (COBEM 2013)
November 3-7, 2013, Ribeirão Preto, SP, Brazil

PRELIMINARY DESIGN OF A COMBUSTION CHAMBER FOR MICROTURBINE BASED IN AUTOMOTIVE TURBOCHARGER

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Abstract. *The design of a combustion chamber for gas turbine is one of the most critical process on the development of this unit. Researching at literature reveals that even though there are already methodologies presented in a concatenated manner, these methods have been developed to conventional gas turbines. Microturbines are very small generators that burns liquid and gaseous fuels, performing a Brayton cycle and ranged less than 250 kW. These kind of turbogenerators are ideally suitable for distributed generation applications due their capability in supply reliable and stable power in stand alone or parallel operation and also are used as auxiliary power unit (APU) on aircrafts. Utilization of off-the-shelf commercial turbochargers is shown a viable solution for microturbines development due the compressor and turbine assembly has already developed on the turbocharger manufacturer. The present paper presents a conceptual design of a reverse flow combustion chamber from a of-the-shelf automotive turbocharger assembly. The design begins on the thermodynamic analysis of the system based on compressor performance chart and combustor performance requirements and come up with the steps to design and presents the dimensional values computed.*

Keywords: *microturbine, distributed generation, combustion chamber*

1. INTRODUCTION

During the last few decades, several attemps have been done to develop microturbines with efficiency levels close to those of larger gas turbines. Particularly for microturbines with power rated below 100 kW, many developments have failed to obtain sufficient efficiency, reliability and cost effectiveness due the small-effects: low Reynolds in the turbomachinery passages causing relatively high viscous losses, relatively high tip clearances due the manufacturing tolerances and bearing limitations, large area-to-volume ratios resulting in high heat losses and inadvertent heat transfer to the compressor and high auxiliary system losses. Another factor in costs is that the development of efficient turbomachinery optimized for a particular cycle is very expensive and only may be justified with very large production volumes. This factor finds an interesting opportunity in using small automotive turbocharger components. Off-the-shelf automotive turbochargers already presents sufficient efficiency for gas turbine cycles and the price is low due the large production volumes (Visser et. Al.,2011). Moreover, the assembling of compressors and turbines of different models may be done in order to reach best efficiency.

Several studies have been done in Brazil in order to investigate, organize and catenate the methodologies of Melconian & Moldak (1985) and Lefebvre (1983) for gas turbines combustor design. Works of Conrado (2002) e de Silva (2006) pioneered the organization of these information on a systemic sequence. The same methodology were used by Navia (2010) for the development of a methodology of a multi-fuel combustor.

Although the Melconian & Moldak (1985) and Lefebvre (1983) methodologies have been developed for the design of aeronautical and large industrial gas turbine combustor, it were used for microturbine combustion chambers by Kulshreshtha & Channiwala (2005) and Visser (2011). Tsai (2004) and Ribeiro (2003) developed a microturbine based on a automotive turbocharger. Once the performance of the turbomachinery have been already defined by the turbocharger manufacturer, the combustor design shall begin from the compressor and turbine performance charts analysis.

Since the thermodynamic parameters are defined, the reference areas shall be computed by analyzing the chemical and aerodynamic performance of the reactor. Then the liner dimensions were computed and also the airflow, the swirler, and the admission holes.

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ISSN 2176-5480

2. COMBUSTION CHAMBER

The combustion chamber, or combustor, of a gas turbine is the device that receives the pressurized air from compressor and promotes its mixture with the fuel in order to release the heat energy through a combustion reaction. Gas turbines works with high excess of air, usually out of the flammability limits, and so a flame tube, or liner, is used to improve the distribution of air through the reactor.

Basically the liner divides the combustion chamber in three zones: primary zone, secondary or intermediate zone and dilution zone. At the primary zone a recirculation zone shall be developed to ensure the stability of the flame. The function of each zone is well defined by Lefebvre (1983)

The main function of the primary zone is to anchor the flame and provide sufficient time, temperature, and turbulence to achieve essentially complete combustion of the incoming fuel–air mixture. The importance of the primary-zone airflow pattern to the attainment of these goals cannot be overstated. Many different types of flow patterns are employed, but one feature that is common to all is the creation of a toroidal flow reversal that entrains and recirculates a portion of the hot combustion gases to provide continuous ignition to the incoming air and fuel.

If the primary-zone temperature is higher than around 2000 K, dissociation reactions will result in the appearance of significant concentrations of carbon monoxide (CO) and hydrogen (H2) in the efflux gases. Should these gases pass directly to the dilution zone and be rapidly cooled by the addition of massive amounts of air, the gas composition would be “frozen,” and CO, which is both a pollutant and a source of combustion inefficiency, would be discharged from the combustor unburned. Dropping the temperature to an intermediate level by the addition of small amounts of air encourages the burnout of soot and allows the combustion of CO and any other unburned hydrocarbons (UHC) to proceed to completion.

The role of the dilution zone is to admit the air remaining after the combustion and wall-cooling requirements have been met, and to provide an outlet stream with a temperature distribution that is acceptable to the turbine. The main components of a combustor are shown on Fig. 1

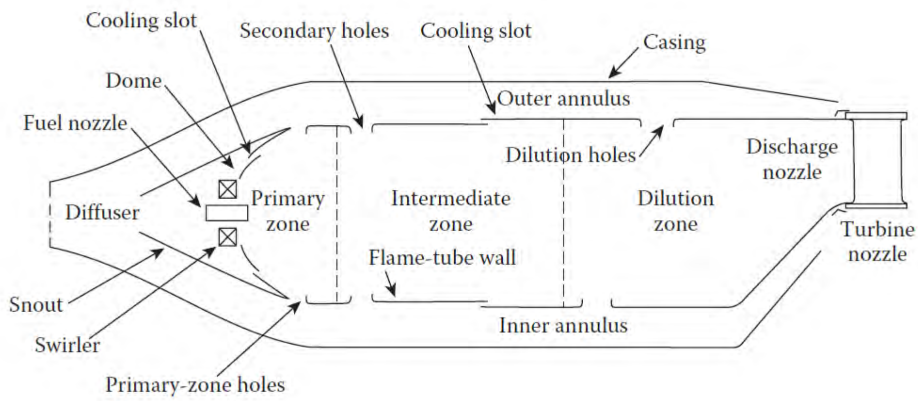

Figure 1. Combustion chamber

The function of the diffuser is to decrease the airflow velocity at the combustor inlet. A swirler is positioned in order to create a toroidal recirculation zone.

3. MICROTURBINE

The microturbine is an assembling of 110/55 AQM compressor of the Swchitzer S500 turbocharger and an 87PJ1 turbine of a Swichtzer S400 turbocharger. Compressor and turbine are both single stage and centrifugal. The combustion chamber is a single-can type operating in reversal flow. Natural gas is fed via a single gas injector connected to a pressure reducer valve. The design point was chosen on compressor performance map analysis for 77 krpm.

At this point the compressor isentropic efficiency is 75.40%. The turbine chart defines the maximum Turbine Inlet Temperature (TIT) of 923.15 K. At this condition the mass fuel flow is 6.42 x 10^{-3} kg/s for methane. The total air mass flow feed by compressor is 0.58 kg/s. Performing the thermodynamic analysis, it have been found that the compressor outlet temperature is 447,07 K and the compressor outlet pressure is 310 kPa.

Table 3 summarizes the main microturbine characteristics. Figure 2 shows the microturbine assembly.

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Table 1. Microturbine characterization

Parameter	Value
RPM	77 krpm
compressor isoentropic efficiency	75.40 %
compressor outlet temperature - T_3	447.07 K
compressor outlet pressure - P_3	310 kPa
turbine inlet temperature - T_4	923.15 K
fuel	Methane - CH_4

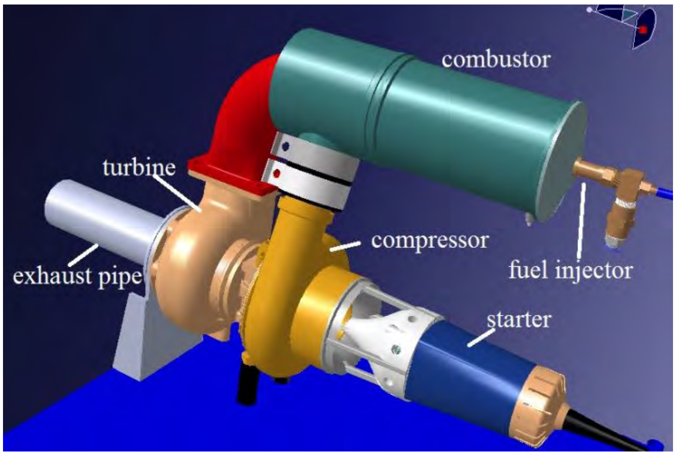

Figure 2. Microturbine Assembly

4. CONCEPTUAL DESIGN

4.1 Reference areas and diamenters

Once the design point is defined the reference area is computed for aerodynamic considerations by Eq. (1).

$$A_{ref} = \left[143.5 \left(\frac{\dot{m}_3 . \sqrt{T_3}}{P_3} \right)^2 . \frac{\frac{\Delta P_{3-4}}{q_{ref}}}{\frac{\Delta P_{3-4}}{P_3}} \right]^{0.5} \quad (1)$$

Where ΔP_{3-4} / q_{ref} is the ratio of the total pressure drop across the combustor and the reference dynamic pressure and ΔP_{3-4} / P_3 is the ratio of the total pressure drop across the combustor to the inlet total pressure. The dimensionless aerodynamic factors are shown in Table 2.

Table 2. Dimensionless aerodynamic factors and reference dimensions calculated.

Reference	$\frac{\Delta P_{3-4}}{P_3}$	$\frac{\Delta P_{3-4}}{q_{ref}}$	A_{ref} [m²]	D_{ref} [m]
Melconian & Moldak (1985)	5,30 %	40	0.013	0.13
Lefebvre & Ballal (2010)	7,00%	37	0.011	0.10

The flame burning velocity theory had been formerly developed by Greenhough & Lefebvre (1957) and then revised by Lefebvre (1966). The model correlates the main operation parameters (pressure, temperature and mass flow) with the combustion chamber dimensions in a dimensionless parameter θ expressed on Eq. (2).

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$$\theta = \frac{P_3^{1.75}.A_{ref}.D_{ref}^{0.75}.\exp\left(\frac{T_3}{300}\right)}{\dot{m}_3} \quad (2)$$

This correlation is shown graphically on Fig 3. Melconian & Moldak (1985) suggests a value for θ of 73 x10^6 for best efficiency. Then A_{ref} = 0.007 m² and D_{ref} = 0.05 m.

As the reference parameters computed for aerodynamic considerations comprises the burning velocity model considerations, a value of 0.013 m² were chosen for the reference area and a value of 0.13 m for the diameter. The ratio between the flame tube area and the reference area for tubular combustion chamber shall be 0.7, then A_{ft} = 0.01 and D_{ft} =0.11. The reference area is now assumed as the casing area (A_{cas}). Figure 3 shows the combustion efficiency chart for conventional combustors.

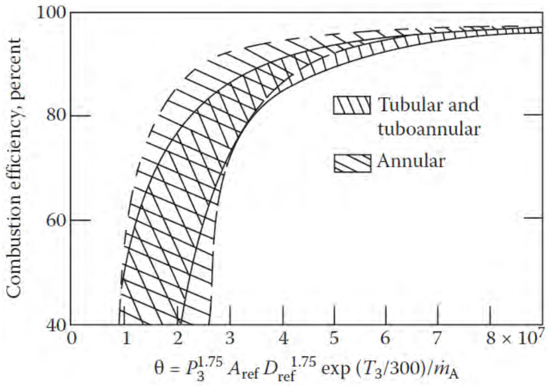

Figure 3. Design chart for conventional combustors.

4.2 Preliminary airflow distribution

The second step is to determine the airflow and the length of each zone.

Firstly the amount of air that shall be reserved for cooling through the liner using Eq. (3) (Odgers, 1980)

$$\frac{\dot{m}_{cool}}{\dot{m}_3} = 0.10.T_3 - 30 \quad (3)$$

The airflow at the remaining zones is computed from the overall equivalence ratio (Φ_{ov} = 0.190). Melconian & Moldak (1985) defines that the primary zone equivalence ratio (Φ_{pz}) never be richer than 1.5 in order to minimize smoke, carbon monoxide (CO) and unburned hydrocarbons (UHC) on the exhaust gases. Furthermore it determines that the minimum flame temperature is 1600 K, which corresponds an equivalence ratio of 0.5 for the design point. On the other hand a maximum flame temperature of 1773 K shall be regarded to prevent nitrous oxides (NO_x) and other pollutants due thermal dissociation. Then a maximum value of equivalence ratio is 0.6. Its value have been chosen for design development. The equivalence ratio of secondary zone (Φ_{sz}) shall not be higher than 0.8. Table 3 summarizes the airflow characterization.

Table 3. Airflow characterization.

	Overall	Cooling	Primary Zone	Secondary Zone	Dilution Zone
equivalence ratio	0.201	-	0.60	0.40	0.190^(1)
airflow - %	100	14.71	31.67	15.83	53.63^(2)

(1) considering that all the remaining cooling air enters at dilution zone

(2) for admission holes determination

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4.3 Combustor zones length

Melconian & Moldak (1985) suggests that the primary zone length (L_{pz}) shall be within 2/3 a 3/4 of D_{ft}. Second value is related with better combustion efficiency and have been chosen. Similarly a value of a half of D_{ft} shall be taken for secondary zone length (L_{sz}). Total length of the flame tube shall consider the transverse quality (TQ) of temperature distribution on the combustor exit in order to preserve the turbine vanes. A method for calculation of total length is given in Lefebvre (1983) and a TQ of about 15% were considered. Dilution zone length (L_{dz}) is the difference between total length and the sum of primary and secondary zone lengths. Table 4 shows the lengths of the flame tube

Table 4. Flame tube zones lengths.

Total	Primary Zone	Secondary Zone	Dilution Zone
0.230 m	0.082 m	0.054 m	0.094 m

4.4 Recirculation zone

Diffuser in its simplest for is merely a diverging passage in which the airflow from the compressor is decelerated, and this reduction in velocity causes an increase in static pressure. The efficiency of this conversion process is of paramount importance because any losses that may occur are manifested as fall in total pressure across the diffuser (Lefebvre, 1983).

The design of the diffuser is usually dictated by space restrictions of the engine. The final design of the diffuser will actually represent the compromise among these space restrictions, pressure losses and uniformity of the exit flow. Although the presence of diffuser is of relevant importance, the authors decided do not use a diffuser in the design due the space limitations of a reverse flow combustor which have been chosen.

Nevertheless the presence of an recirculation zone surrounding the fuel injector is need in order to create aerodynamic stabilization of the flame. The recirculation zone is created at this design via an axial swirler and a divergence on the dome as shown in Fig. 4

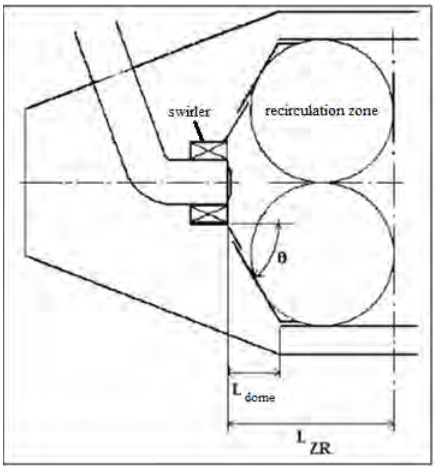

Figure 4. Recirculation zone.

The swirler design is well described in Melconian & Moldak (1985) and starts of Eq. (4) of Knight and Walker (1957) that relates the pressure losses in combustion chambers.

$$\frac{\Delta P_{sw}}{q_{ref}} = K_{sw}\left[\left(\frac{A_{ref}}{A_{sw}}\right)^2 .\sec^2\beta - \left(\frac{A_{ref}}{A_{ft}}\right)^2\right].\left(\frac{\dot{m}_{sw}}{\dot{m}_3}\right)^2 \tag{4}$$

Where $\Delta P_{sw}/q_{ref}$ is the pressure drop on swirler (3%-4% of P_3), K_{sw} is the geometrical coefficient of the blades (1.30 for thin straight blades and 1.15 for thin curved blades) and β is the blade angle (usually within 45° and 70°). The ratio between the swirler airflow and the total airflow shall be within 3 and 12% of the total flow that exits the compressor.

Intensity of the flow on recirculation zone is given by the swirl number (Beér & Chigier,1972) (Eq. (5)) which is a dimensionless parameter that relates the components of angular and axial moments and shall be greater than 0.6 to ensure the recirculation.

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$$S_n = \left\{ \frac{2}{3} \cdot \left[\frac{1-\left(\frac{D_{int,sw}}{D_{sw}}\right)^3}{1-\left(\frac{D_{int,sw}}{D_{sw}}\right)^2} \right] . \tan \alpha_{sw} \right\} \tag{5}$$

The angle of the dome is given by Eq. (6). and the length of dome in Eq. (7). The recirculation zone diameter (D_{rz}) shall be twice the swirler diameter.

$$\theta_{dome} = \operatorname{acos}\left[\frac{-D_{ref}.(D_{ft}-2.D_{sw})-(D_{ft}-4.L_{rz}).\sqrt{(D_{ft}^2-4.D_{ft}.D_{sw}+4.D_{sw}^2-8.D_{ft}.L_{rz}+16.L_{rz}^2)}}{2.D_{ft}^2-4.D_{ft}.D_{sw}+4.D_{sw}^2-8.D_{ft}.L_{rz}+16.L_{rz}^2}\right] \tag{6}$$

$$L_{dome} = \frac{D_{ft}-D_{sw}}{2.\tan \theta_{dome}} \tag{7}$$

Then the recirculation parameters of the recirculation zone are summarized on Table 5.

Table 5. Recirculation zone parameters.

Parameter	Value
swirler inner diameter	0.018 m
swirler outer diameter	0.030 m
blade angle	60°
$\dot{m}_{sw}/\dot{m}_3$	9.5%
9 thin straight blades (thickness = 0.001 m)	K_{sw} = 1.30
recirculation zone diameter	0.06 m
angle of dome	75°
length of dome	0.010 m

4.5 Gas temperature profile

Defining the temperature profile through combustor is need in order to compute the flame tube cooling. It aims to find the higher temperature points at the flame tube surfaces. Calculation of the gas temperature profile and the heat transfer at the liner walls was done according the empirical methodology based in combustion efficiency described by Melconian & Moldak (1985) e reviewed by Gosselin et Al. (1999) and for this purpose the combustion chamber is divided in four zones: recirculation zone, remaining of primary zone, secondary zone and dilution zone. For each zone the local flame temperature will be assumed to vary linearly between T_{in} and T_{out}.

The recirculation zone is assumed to be an partly stirred reactor, then the temperature along this zone is assumed the mean value of Eq (8).

$$T_{mean,zr} = \frac{T_3}{3} + 2.\frac{T_{out,rz}}{3} \tag{8}$$

The initial temperature of RZ is T_3 and the outlet temperature of ZR is calculated by Eq. (9).

$$T_{out,rz} = T_3 + \ _{rz}.\ \ T_{\ =1} \tag{9}$$

Where:

$$\eta_{rz} = 0.56 + 0{,}44.\tanh[1.5475x10^{-3}.(T_3 + 108.\ln p_3 - 1863)] \tag{10}$$

And for the remaining of primary zone:

$$T_{out,pz} = T_3 + \ _{pz}.\ \ T_{\ pz} \tag{11}$$

$$\eta_{pz} = 0.71 + 0.29.\tanh[1.5475x10^{-3}.(T_3 + 108.\ln p_3 - 1863)] \tag{12}$$

For the secondary zone

$$T_{out,sz} = T_3 + \ _{sz}.\ \ T_{\ sz} \qquad (13)$$

$$\log\left(\log\frac{1}{\eta_{sz}}\right) = 0.911.\log\ \ _{T3} + 8.02.\ \ _{sz} - 1.097 + D^* \qquad (14)$$

Where:

$$D^* = 0.736 - 0.0173.\left(\frac{\Delta P_{3-4}}{P_3}\right)^{-1} \qquad (15)$$

$$_{T3} = \left(10^{-3{,}054.\phi_{zs}{}^{-1{,}205}}\right).\left(T_3^{1{,}2327.\phi_{zs}{}^{-1{,}205}}\right).\left(\frac{\dot{m}_{comb}}{V_{zp}.P_3^n}\right) \qquad (16)$$

and

$$V_{zp} = \ .D_{tc}^2.\left[\frac{(L_{zp}-L_{zr})}{4} + \frac{L_{zr}}{12}\right] \qquad (17)$$

Assuming that in the inlet of recirculation zone the temperature is 1102 K and the inlet temperature for each zone is the outlet temperature of the preceding zone, the final temperature for each zone is shown on Tab. 5 and gas temperature profile is shown in Fig. 5.

Table 5. Combustion zone out temperature.

recirculation	primary zone	secondary zone	dilution zone
1102	1360	1281	921

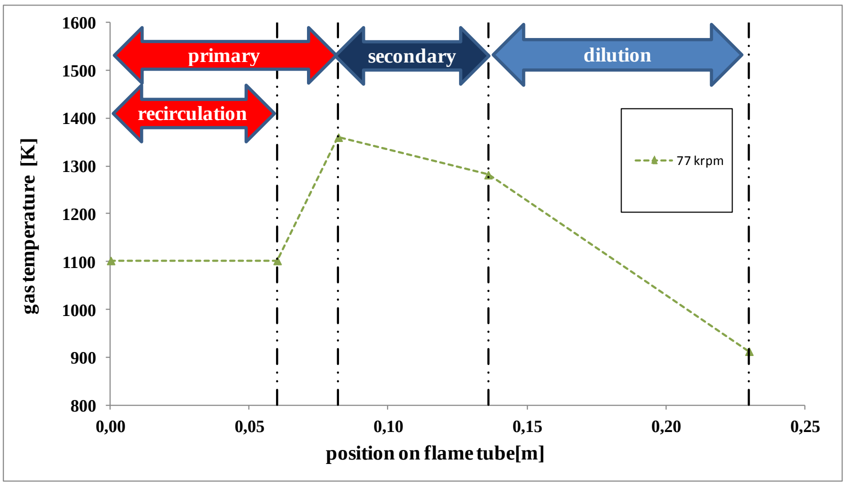

Figure 5. Recirculation zone.

4.6 Heat transfer to the walls

In order to define the temperature on the walls of the liner an uncooled temperature calculation have been done as defined in Lefebvre (1983) and Gosselin et al. (1999). For the purpose of analysis, a liner may be regarded as a container of hot flowing gases surrounded by a casing, with air flowing between the container and the casing. Broadly, the liner is heated by radiation and convection from the hot gases inside it; it is cooled by radiation to the outer casing and by convection to the annulus air (Lefebvre, 1983). Figure 6 show the basic heat transfer process.

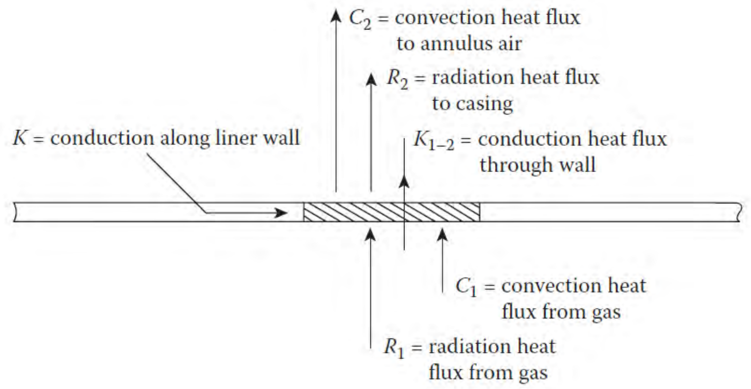

Figure 6. Basic heat transfer processes.

The basic modeling for heat transfer on liner wall assuming that the thermal resistance of the flame tube may be neglected (very small thickness) is given by Eq. 18.

$$(R_1 + C_1) = (R_2 + C_2) \tag{18}$$

Where R, C and are the radiative and convective fluxes and the subscript 1 and 2 means internal and external respectively.

Then it may be summarized as:

$$\mathrm{R_1} = 0.5.\sigma_{\mathrm{SB}}.(1+\varepsilon_{\mathrm{w}}).\varepsilon_{\mathrm{g}}.\mathrm{T_g^{1.5}}.\left(\mathrm{T_g^{2.5}} - \mathrm{T_{w1}^{2.5}}\right) \tag{19}$$

$$\mathrm{C_1} = 0.020.\frac{\mathrm{k_g}}{\mathrm{D_{ft}}^{0.2}}.\left(\frac{\dot{\mathrm{m}}_\mathrm{g}}{\mathrm{A_{ft}}.\mu_\mathrm{g}}\right)^{0.8}.(\mathrm{T_g} - \mathrm{T_{w1}}) \tag{20}$$

$$\mathrm{R_2} = \sigma_{\mathrm{SB}}\frac{\varepsilon_{\mathrm{w2}}\varepsilon_{\mathrm{carc}}}{\varepsilon_{\mathrm{cas}}+\varepsilon_{\mathrm{w2}}(1-\varepsilon_{\mathrm{cas}})(\mathrm{A_{w2}}/\mathrm{A_{cas}})}(\mathrm{T_{w2}^4} - \mathrm{T_3^4}) \tag{21}$$

$$\mathrm{C_2} = 0.020.\frac{\mathrm{k_a}}{\mathrm{D_{an}}^{0.2}}.\left(\frac{\dot{\mathrm{m}}_\mathrm{an}}{\mathrm{A_{an}}.\mu_\mathrm{a}}\right)^{0.8}.(\mathrm{T_{w2}} - \mathrm{T_3}) \tag{22}$$

Where the gas emissivity is given in Eq. 23.

$$\varepsilon_g = 1 - \exp\left[-0.0746.\mathrm{L_u p_3}.(\phi.0{,}9\mathrm{l_b})^{0.5}.\mathrm{T_g^{-1.5}}\right] \tag{23}$$

The methane flame were assumed as non luminous flame and the luminosity factor (L_u) is the unity. For tubular combustors $l_b = 0.6D_{ft}$.

Gas viscosity and thermal conductivity are represented by μ and λ respectively. The subscript "an" means the annulus area which is is the difference between A_{cas} and $A_{ft.}$

After a interactive process the calculated wall temperature and the wall temperature profile aer shown on Tab. 6 and Fig. 6.

Table 6. Combustion zone wall out temperature.

recirculation	primary zone	secondary zone	dilution zone
524	651	695	714

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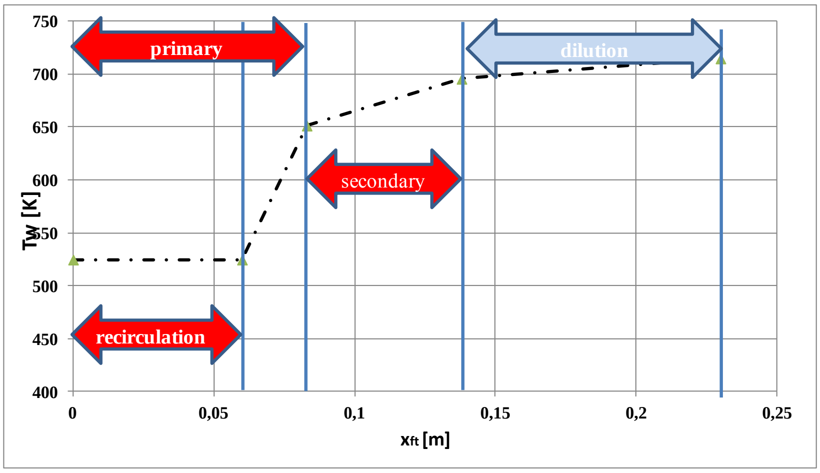

Figure 7. Wall temperature profile.

A typical value for maximum operating temperature on the walls is 1100 K. The maximum wall temperature which occurs on the exit of the flame tube is 712 K. Gosselin et. al (1999) defines the accuracy for this prediction in ± 100 K for primary zone and ± 30 K at the other zones. Thus it may be concluded that at the design point there is no need of cooling slots then the design assumes that all the air held for cooling will enter to the inner side of the flame tube via a row of orifices at the end of dilution zone. This assumption ensures the convection flow at the annulus area through the entire liner.

4.7 Admission holes

The determination of the admission hole size is done via an iterative process after determining the air mass flow rate that enter into each zone. The iterative process occurs due the discharge coefficient ($C_{d,h}$) is unknown. The sequence of calculation described in Lefebvre (1983) is presented below:

1. Calculation of bleed ratio (β_h) that is defined as the ratio between the air mass flow that enter the hole (total mass flow) and the air mass flow that occurs in the annulus area.

$$\beta_h = \frac{\dot{m}_{h,t}}{\dot{m}_{an}} \tag{24}$$

2. Establish a reasonable value of discharge coefficient.

3. Determine the total area of holes ($A_{h,t}$) for each row using Eq. (25) assuming the pressure loss through a hole (ΔP_{ft} / P_3) as 0.6.

$$\frac{\Delta P_{ft}}{P_3} = \frac{143{,}5.(\dot{m}_{h,t})^2.T_3}{P_3^2 C_d^2 A_{h,t}} \tag{25}$$

4. Calculate the hole area ratio (α_h) that is the ratio between the total hole area and the annulus area (Eq. (26)) and the ratio between β_h and α (Eq. (27).

$$\alpha_h = \frac{A_h}{A_{an}} \tag{26}$$

$$\mu_h = \frac{\beta_h}{\alpha_h} \tag{27}$$

5. Calculate the pressure loss factor (K) using the Eq. (28) where δ is the momentum loss factor (δ = 0.8 for plain holes).

$$K = 1 = \delta.\left\{2.\mu^2 + \left[4.\mu^4 + \left(\frac{\mu^2}{\delta^2}\right).(4.\beta - \beta^2)\right]^{0.5}\right\} \quad (28)$$

6. Replace into the Eq. (29) the value of pressure loss factor to obtain the value of $C_{d.h}$.

$$C_d = \frac{(K-1)}{\delta[4K^2 - K.(2-\beta)^2]^{0.5}} \quad (29)$$

Once the total area of the holes is computed for each zone the quantity of holes and rows shall be defined.

Table 7. Admission holes.

	primary zone	secondary zone	dilution zone	cooling
number of holes	12	7	11	30
diameter [mm]	8	9	11	4
x [mm]	64	86	130	205

5. CONCLUSION AND COMMENTS

The conceptual design of a combustion chamber which shall be assembled on a off-the-shelf automotive turbocharger based on the methodology suggested by Melconian & Moldak (1985) and Lefebvre (1983) is viable.

Even these references proposes the methodology for aircraft engines combustors and for industrial purpose, all dimensions computed are suitable for the machining and the assembling on microturbine.

Large turbines operates with P_3 levels of MPa and T_3 about 1000 K. The microturbine proposed works with T_3 = 447 K and P_3 = 310 kPa, so it may be reasonable that at this condition the gas temperatures are low due their dependence on T_3 and also the wall temperatures which results of the gases temperatures and there is no need of cooling slots along the liner.

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ISSN 2176-5480

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