

BASIC DESIGN PRINCIPLES FOR GAS TURBINE COMBUSTOR

Ana Costa Conrado

Instituto Tecnológico de Aeronáutica, Divisão de Engenharia Mecânica Aeronáutica
Pça. Mal. Eduardo Gomes 50, São José dos Campos – SP, 12228-900
aconrado@fastmail.fm

Pedro Teixeira Lacava

Instituto Tecnológico de Aeronáutica, Departamento de Propulsão
Pça. Mal. Eduardo Gomes 50, São José dos Campos – SP, 12228-900
placava@ita.br

Alberto Carlos Pereira Filho

Instituto Tecnológico de Aeronáutica, Departamento de Física
Pça. Mal. Eduardo Gomes 50, São José dos Campos – SP, 12228-900
acpereira@directnet.com.br

Milton de Souza Sanches

Instituto Tecnológico de Aeronáutica, Departamento de Física
Pça. Mal. Eduardo Gomes 50, São José dos Campos – SP, 12228-900

***Abstract.** The present work shows a methodology for gas turbine combustor basic design. The idea is to provide information for producing designs that require minimum development time. As such, the methodology emphasis is on the practical rather than theoretical aspects of combustor design. Criteria for selecting a suitable combustor configuration is examined followed by design calculations for the dimensions of the casing, the liner, the diffuser, and the swirler. Calculations of gas temperature in the various zones of the combustor and liner wall temperatures in the presence of film cooling are performed along with design calculations for the dimensions of the air admission holes. A computational program was developed based on the sequence of equations discussed in the paper, and an example of design is also presented. The sequence of calculations presented in the present paper does not represent the most advanced methodology possible to design a gas turbine combustion chamber; however, it is able to provide a combustor design that attends gas turbine operation conditions*

***Keywords.** gas turbine combustor, gas turbine.*

1. Introduction

Gas turbines must typically operate over a wide load range. Thus, the combustion chamber should be designed to operate stably over wide range of conditions. Some wanted aspects are: rapid and reliable ignition, operate over a wide range of mixtures without danger of blowoff, have minimum loss in pressure, have uniform exit temperature, be small in size and durable, and have low emission. These aspects are wanted for aircraft and industrial gas turbine combustors, but for aircraft combustors the requirements are more rigorous. Conventional gas turbines consist of an inlet diffuser, a fuel injector, a swirler, and three combustion zones, primary, secondary and dilution.

The aim of the present work is to provide information for conventional combustors design methodology, which requires minimum development time. As such, the methodology emphasis is on the practical rather than theoretical aspects of combustor design. Some equations utilized in the calculation are based on empirical works available in the literature or in the accumulated experience of some decades of gas turbine utilization. The sequence of calculations presented in the present paper does not represent the most advanced methodology possible to design a gas turbine combustion chamber; there are recent articles in the specialized literature which present some new advances in this area. However, the idea of the present work is not to show “the state of the art” of combustion chamber design, but a methodology that does not require sophisticated computational calculations. So that, classical empirical results are used to obtain the combustor configuration and operational parameters.

A criterion for selecting a suitable combustor configuration is examined followed by design calculations for the dimensions of the casing, the liner, the diffuser, and the swirler. Calculations of gas temperature in the various zones of the combustor and liner wall temperatures in the presence of film cooling are performed along with design calculations for the dimensions of the air admission holes. A computational program was developed based on the purposed preliminary design methodology, and a combustor chamber designed using this program is also presented here

2. Preliminary Design Procedure

The procedure purposed by Melconian and Modak (1985) to design a combustor is described in Fig.1. The steps in dashed lines are not part the preliminary design procedure described in the present paper. In the next subsection, just part of the equations utilized in the design procedure is presented, due the limited space. But they are sufficient to the reader understand the design methodology idea. The cited references might help the reader to complete the cast of

equations, but the detailed methodology can be observed in the primary reference of the present article, Conrado (2002). This reference is graduate final work, and it can be obtained directly to the authors.

2.1. Combustor Dimensions

The reference area (A_{ref}) is defined as the maximum cross sectional area of the casing in the absence of a liner; and it is selected by considering the possibility of either chemical or pressures loss limitations. Figure 2 presents the reference length D_{ref} for different combustor configurations.

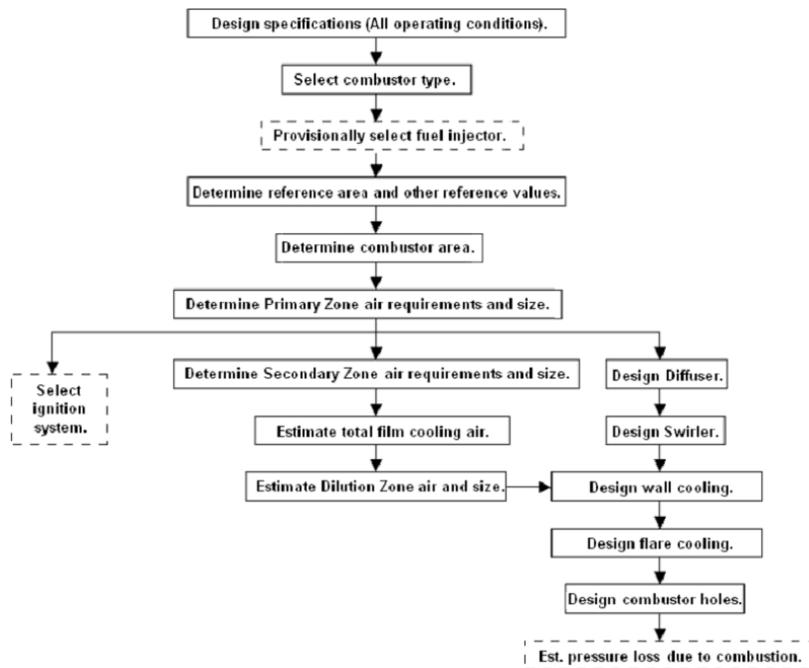


Figure 1. Preliminary Design Procedure (Melconian and Modak, 1985).

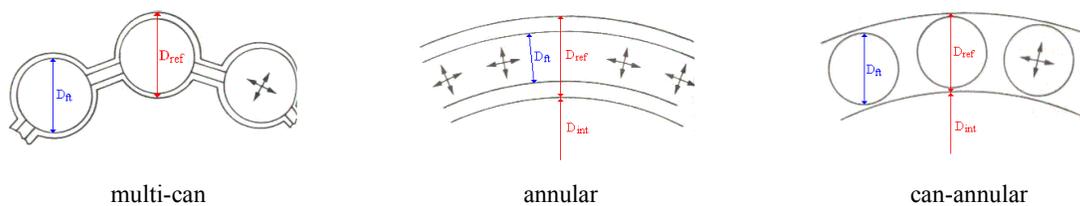


Figure 2. Reference length D_{ref} for different combustor configurations.

The dimensions of a combustor might be determined either by aerodynamic or by chemical rate control. Generally, when the combustor is sized for a specific pressure loss, it will be sufficient to accommodate the chemical process too. However, it is necessary verify all possibilities before the final choice. So that it is recommended attention about the aerodynamic and chemical considerations. The A_{ref} is calculated by the Eq.'s (1) or (2) (Fig. 2).

$$A_{ref} = \pi D_{ref}^2 / 4 \quad \text{for multi-can combustor} \quad (1)$$

$$A_{ref} = [\pi(2 D_{ref} + D_{int})^2] / 4 - (\pi D_{int}^2) / 4, \quad \text{for annular and can-annular combustors} \quad (2)$$

2.1.1. Aerodynamic Considerations

Table 1 presents representative values of pressure loss terms for different combustors, where ΔP_{3-4} is the combustor overall pressures loss, P_3 is the total pressure at the combustor inlet, q_{ref} is the dynamic pressure at the maximum cross-sectional area, \dot{m}_3 is the total air mass flow rate and T_3 is total temperature at the combustor inlet. And the Eq. 3 calculates the reference area.

Table1. Representative values of pressure-loss terms for Combustors (Melconian and Modak, 1985).

Combustor	$\Delta P_{3-4}/P_3$ (%)	$\Delta P_{3-4}/q_{ref}$	$\dot{m}_3 \sqrt{T_3}/A_{ref} P_3$
Multi-can	5,3	40	$3,0 \times 10^{-3}$
Can-annular	5,4	30	$3,5 \times 10^{-3}$
Annular	6,0	20	$4,5 \times 10^{-3}$

$$A_{ref} = \left[R_a \left(\frac{\dot{m}_3 \sqrt{T_3}}{P_3} \right)^2 \left(\frac{\Delta P_{3-4}}{q_{ref}} \right) \left(\frac{\Delta P_{3-4}}{P_3} \right) \right]^{0,5}, \quad (3)$$

where R_a is $143,5 \text{ J.kg}^{-1}.\text{K}^{-1}$.

2.1.2. Combustion Considerations

For any operation condition (fuel/air ratio), the combustion efficiency is a function of Ω , defined in Eq. 4 (Lefebvre and Halls, 1959). The empirical relation between combustion efficiency and the parameter Ω is presented in Lefebvre (1966) and in Melconian and Modak (1985). The major part of combustors has combustion efficiency close to 100% when Ω is 73×10^{-6} (SI units in Eq. 4).

$$\Omega = \frac{P_3^{1,75} A_{ref} D_{ref}^{0,75} \exp(T_3/b)}{\dot{m}_3}. \quad (4)$$

The parameter b in Eq. 4 is the temperature corrector factor, calculated by the empirical Eq.'s 5 or 6 (Herbert, 1962), depending the equivalence ratio in primary zone (defined in Eq. (7)).

$$b = 245 (1.39 + \ln \phi_{PZ}) \quad 0.6 < \phi_{PZ} < 1.0 \quad (5)$$

$$b = 170 (2.00 - \ln \phi_{PZ}) \quad 1.0 < \phi_{PZ} < 1.4 \quad (6)$$

$$\phi_{PZ} = \frac{\phi_{global}}{\dot{m}_{ZP}/\dot{m}_3} \quad (7)$$

In Eq. 7, \dot{m}_{ZP} is the air mass flow rate in primary zone and ϕ_{global} is the overall combustor equivalence ratio. The primary zone equivalence ratio must be chosen into the flammability range. For most hydrocarbon combustion in air, examination of the determined flame limit temperatures for lean mixtures shows that they lie around 1600K (Melconian and Modak, 1985), and pressures has little effect upon this limit. Then, the lean mixture limiting equivalence ratio will be given by a mixture having a temperature rise of $\Delta T = 1600 - T_3$. The corresponding rich extinction value is obtained in a similar way. The flammability limits might be determined by thermochemistry calculations (chemical equilibrium).

Equation 4 is solved with numerical methods to find D_{ref} , replacing the reference area A_{ref} from Eq. 1 or 2 and the temperature correction factor b from Eq. 5 or Eq. 6.

2.1.3. Combustor Area and Diameter

There will be two values of D_{ref} and A_{ref} for each operating condition, one from aerodynamic consideration and other from combustions considerations. These values are used to calculate the combustor sectional area (A_{fit}) by the Eq. 8.

$$A_{fit} = 0.7 A_{ref} \quad (8)$$

This simple relationship is quite satisfactory for single can, multi-can, and can-annular; but for annular combustor a value of about 0.65-0.67 is more appropriated (Melconian and Modak, 1985). The sectional length D_{fit} is obtained from the value A_{fit} and in agreement to the combustor geometry (see Fig. 2).

The values of D_{ref} and D_{fit} must be chosen such as it accommodates the aerodynamic and combustion considerations in every operating condition.

2.2. Preliminary Estimate of Air Distribution and the Size of Combustor Zones

According to Odgers (1980), the amount of film cooling air is:

$$\text{Percent film cooling air (of the total air from the compressor)} = 0.1T_3 - 30 \quad (9)$$

The air on the primary and secondary zone is defined considering the richest operating condition and replacing its equivalence ratio in Eq. 10 (Melconian and Modak, 1985), \dot{m}_{SZ} is the air on secondary zone.

$$\frac{\dot{m}_{PZ} + \dot{m}_{SZ}}{\dot{m}_3} = \frac{(\phi_{global})_{+rich}}{0,8} \quad (10)$$

The length of the primary zone (L_{PZ}) is considered to be $\frac{3}{4}$ of D_{ft} and the secondary zone is $\frac{1}{2}$ of D_{ft} (Melconian and Modak, 1985). The length of dilution zone is a function of Temperature Traverse Quality in the combustor exit (TQ) and Pressure Loss Factor ($\Delta P_{3-4}/q_{ref}$, see Tab. 1). The TQ is defined as peak gas temperature minus mean gas temperature divided by mean temperature rise ($TQ = (T_{max} - T_4)/(T_4 - T_3)$), and the usual range is between 0.05 and 0.30. Typical values for Pressure Loss Factor are presented in Tab.1. Table 2 shows the dilution zone length to sectional length ratio (L_{DZ}/D_{ft}) as function of TQ for different values of Pressure Loss Factor. These results are based on the graphs presented by Melconian and Modak (1985). For values of $\Delta P_{3-4}/q_{ref}$ different of those presented in Table 2, it is necessary an interpolation. For example, the typical value of $\Delta P_{3-4}/q_{ref}$ for multi-can is 40 (as presented in Table 1), there is not a curve for this value of pressure loss in the TQ x L_{DZ}/D_{ft} diagram presented by Melconian and Modak.

Table 2. Dilution zone length to combustor diameter ratio as function of TQ for different values of Pressure Loss Factor.

$\Delta P_{3-4}/q_{ref}$	L_{DZ}/D_{ft}
15	$3,78 - 6.TQ$
20	$3,83 - 11,83.TQ + 13,4.TQ^2$
30	$2,96 - 9,86.TQ + 13,3.TQ^2$
50	$2,718 - 12,64.TQ + 28,51.TQ^2$

2.3. Diffuser

The Diffuser Geometry is represented in Fig. 3(a). The annular area, A_{an} , is the difference between A_{ref} e A_{ft} . The A_3 area is the compressor exit. The A_0 is calculated assuming the air velocity in this sectional area is equal to A_{an} air velocity, then:

$$\frac{A_0}{A_{an}} = \frac{\dot{m}_3}{\dot{m}_{an}} \quad (11)$$

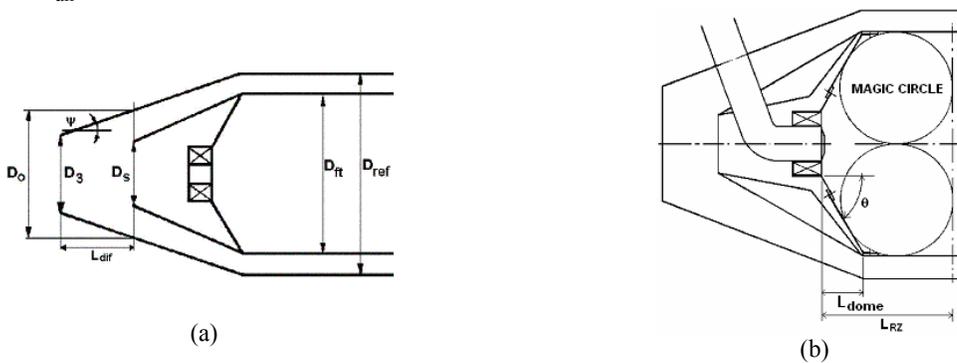


Figure 3. Diffuser geometry (a) and combustor front end (b)

For conventional design, about half of primary zone air mass flow rate would be admitted through the swirler and as dome-cooling. Thus, the air admitted through A_s (\dot{m}_s) is $\dot{m}_{PZ}/2$; and $\dot{m}_{an} = \dot{m}_3 - \dot{m}_s$. The mass flow rate \dot{m}_s corresponds to the sum of the air admitted in primary zone through the swirl and the air admitted through dome-cooling slots.

The angle ψ is obtained by Eq. 12, derived by Kretschmer (cited by Lefebvre, 1983) based upon a formula cited in Perry (1950).

$$\frac{\Delta P_{\text{dif}}}{P_3} = 1.75 \cdot R_a \left[\frac{\dot{m}_3 \sqrt{T_3}}{P_3} \right]^2 \frac{(\tan \psi)^{1.22}}{A_3^2} \left[1 - \frac{A_3}{A_0} \right]^2, \quad (12)$$

and $1.75 \cdot R_a = 502.4 \text{ J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$.

The typical pressure loss in the diffuser $\Delta P_{\text{dif}}/P_3$ is 1%. The area A_s is calculated using Eq. 13 (Melconian and Modak, 1985):

$$\frac{A_s}{A_0} = \frac{\dot{m}_s}{\dot{m}_3} \frac{1}{C_{d,s}}, \quad (13)$$

where, $C_{d,s}$ is the snout discharge coefficient, and for a uniform compressor delivery it should approach unit. The length of the diffuser L_{dif} comes from Eq. 14.

$$L_{\text{dif}} = (R_0 - R_3) / \tan \psi, \quad (14)$$

where R_0 and R_3 are $D_0/2$ and $D_3/2$, respectively (as represented in Figure 3a).

2.4. Swirler

The turning angle of the air flow β_{sw} lies between 45 and 70° (Melconian and Modak, 1985). The swirler area A_{sw} is calculated by the pressure-loss relationship purposed by Knight and Walker (1957).

$$\frac{\Delta P_{\text{SW}}}{q_{\text{ref}}} = K_{\text{SW}} \left[\left(\frac{A_{\text{ref}}}{A_{\text{SW}}} \right)^2 \sec^2 \beta_{\text{SW}} - \left(\frac{A_{\text{ref}}}{A_{\text{ft}}} \right)^2 \right] \left(\frac{\dot{m}_{\text{SW}}}{\dot{m}_3} \right)^2, \quad (15)$$

The total pressure loss across the swirler is the total pressure loss across the combustor minus the total pressure loss inside the snout and minus the total pressure loss in the diffuser. The concordance factor K_{SW} is 1.30 for thin straight blades and 1.15 for thin curved blades. Experiments have shown that the combustor will perform well for amounts ranging from 3% to 12% of the total air (\dot{m}_{sw}), and 8 to 10 blades.

2.5. Recirculation Zone

The length of the recirculation zone approximates two swirler diameters and must be smaller than the primary zone. Equations 16 and 17 calculate the angle and the length of the dome, respectively (see Fig. 3(b)).

$$\theta = \arccos \left[\frac{-D_{\text{ft}}(D_{\text{ft}} - 2D_{\text{SW}}) - (D_{\text{ft}} - 4L_{\text{RZ}}) \sqrt{D_{\text{ft}}^2 - 4D_{\text{ft}}D_{\text{SW}} + 4D_{\text{SW}}^2 - 8D_{\text{ft}}L_{\text{RZ}} + 16L_{\text{RZ}}^2}}{2D_{\text{ft}}^2 - 4D_{\text{ft}}D_{\text{SW}} + 4D_{\text{SW}}^2 - 8D_{\text{ft}}L_{\text{RZ}} + 16L_{\text{RZ}}^2} \right] \quad (16)$$

$$L_{\text{dome}} = \frac{D_{\text{ft}} - D_{\text{SW}}}{2 \tan(\theta)}. \quad (17)$$

2.6. Temperature Calculations

The combustor is divided into four zones: recirculation zone, primary zone, secondary zone and dilution zone. For each zone, the local temperature will be assumed to vary linearly between the zone inlet temperature (T_{in}) and zone outlet temperature (T_{out}). For every zone, the outlet temperature is calculated by the equation (18).

$$T_{\text{out}} = T_3 + \eta \cdot \Delta T \quad (18)$$

2.6.1. Recirculation Zone

In this case, T_{in} is assumed equal to T_3 . ΔT is the temperature rise from T_3 to adiabatic flame temperature for equivalence ratio equal one, and η is the combustion efficiency in the recirculation zone, calculated by the Eq. (19) (Melconian and Modak, 1985).

$$\eta = 0.56 + 0.44 \tanh[1.5475 \times 10^{-3} (T_3 + 108 \ln p_3 - 1863)] \quad (19)$$

2.6.2. Primary Zone (without recirculation zone)

Equation 20 gives the combustion efficiency in this zone, and Eq. 18 is utilized to calculate T_{out} . ΔT is the adiabatic flame temperature for the equivalence ratio ϕ_{PZ} .

$$\eta = 0,71 + 0,29 \tanh[1,5475 \times 10^{-3} (T_3 + 108 \ln p_3 - 1863)] \quad (20)$$

2.6.3. Secondary and Dilution Zone

The secondary zone inlet temperature is equal to the primary zone outlet temperature. To evaluate the efficiency in the secondary zone, first it is necessary to know if mixture is weak or rich (ϕ_{SZ}). In the case of weak mixtures, Eq. 21 calculates the combustion efficiency.

$$\log[\log(1/\eta)] = 0.911 \log \psi_{T_3} + 8.02\phi_{SZ} - 1.097 + D^* \quad (21)$$

where D^* is a empirical constant determined by the equation (22); ψ_{T_3} is the kinetic fuel loading, and for $T_3 = 300K$ is defined as $\dot{m}_f / (V_{PZ} \cdot P_3^n)$, where \dot{m}_f is fuel mass flow rate, V_{PZ} is the primary zone volume, Eq. 23, and n is the reaction order given by Odgers and Carrier (1973) as: $n = 1,0$ for $\phi_{SZ} \leq 0.5$, $n = 2\phi_{SZ}$ for $0.5 < \phi_{SZ} \leq 1.0$, $n = 2/\phi_{SZ}$ for $1.0 < \phi_{SZ} \leq 2.0$, and $n = 1.0$ for $\phi_{SZ} > 2.0$. If T_3 is different of 300K, then the correction equation (Eq. 24) is used for ψ_{T_3} (Venneman, 1959).

$$D^* = 0,736 - 0,0173(P_3/\Delta P_{3-4}) \quad (22)$$

$$V_{PZ} = \pi \cdot D_{it}^2 [(L_{PZ} - L_{RZ})/4 + L_{RZ}/12] \quad (23)$$

$$\frac{\psi_{T_3}}{\psi_{300}} = \left(10^{-3.054y^{-1.205}} \left(T_3^{1.2327y^{-1.205}} \right) \right), \quad (y = \phi_{SZ} \text{ for } \phi_{SZ} \leq 1; y = 1 \text{ for } \phi_{SZ} > 1) \quad (24)$$

For rich mixtures, just the Eq. 25 is utilized to determine the combustion efficiency.

$$\eta = 1/\phi_{SZ} \quad (25)$$

The inlet temperature at the dilution zone is equal to the outlet temperature of the secondary zone, and the combustion efficiency in the dilution zone is also calculated by the Eq.(21).

2.7. Wall Cooling

For the heat transfer evaluation is necessary to select the slot height s , slot wall thickness t and liner wall thickness t_w , the F_{dome} distance from the dome slot to the injector, the number of slots N_{slot} and after that their positions, apart from the selection of the casing material, steel or aluminum. With these values, the slot areas for multi-can chambers and can-annular chambers are obtained as: $A_{slot} = \pi D_{it}$; and for annular chambers

$$A_{slot} = \pi (D_{int} + D_{an}) s + \pi (D_{int} + 2 D_{ref} - D_{an}) s \quad (26)$$

In the section 3, the Figures 8 and 9 show clearly the differences between L_{dome} and F_{dome} , but they are represented in Portuguese, $L_{cúpula}$ and $F_{cúpula}$, respectively.

The airflow rate that enters each slot in a certain operating condition is:

$$\dot{m}_{slot} = \dot{m}_{an} \frac{A_{slot}}{A_{an}} \quad (25)$$

The following step is to determine $\rho_a U_a$, where ρ_a and U_a are the air density and the air flow velocity in A_{an} , respectively.

$$\rho_a U_a = \frac{\dot{m}_{slot}}{A_{slot}} \quad (26)$$

For the dome slot, the mass flow rate, $\dot{m}_{slot,f}$, is $0.03\dot{m}_3$ (Melconian and Modak, 1985). To evaluate the cooling, the temperature in the point just before the next slot air inlet (x) is calculated. In the case of the last slot, the temperature at the end of the combustion chamber is evaluated.

To evaluate the film cooling effectiveness it is necessary to calculate the $\rho_g U_g$, where ρ_g and U_g are the hot gas density and the hot gas flow velocity in A_{ft} , respectively.

$$\rho_g U_g = \dot{m}_g / A_{ft} \quad (27)$$

\dot{m}_g is the hot gas mass flow rate into the combustor (into the liner region). To calculate the film cooling effectiveness, m must be calculated as the $\rho_a U_a$ and $\rho_g U_g$ ratio. With m , the film cooling effectiveness η is given by (Ballal e Lefebvre, 1972):

$$\eta_c = 1.10 m^{0.65} \left(\frac{\mu_a}{\mu_g} \right)^{0.15} \left(\frac{x}{s} \right)^{-0.2} \left(\frac{t}{s} \right)^{-0.2} \quad \text{for } 0.5 < m \leq 1.3 \quad (28)$$

$$\eta_c = 1.28 \left(\frac{\mu_a}{\mu_g} \right)^{0.15} \left(\frac{x}{s} \right)^{-0.2} \left(\frac{t}{s} \right)^{-0.2} \quad \text{for } 1.3 < m < 4.0 \quad (29)$$

where μ_a and μ_g are the air and the hot gas dynamic viscosities, respectively. To calculate μ_a , the air temperature that goes through the slots, T_a , is considered as T_3 . The hot gas temperature, T_g , is obtained from a linear relationship from the entrance to exit of zone. The gas temperature close to the wall is calculated by the equation and it will be utilized to calculate the heat transferred to inner flame tube.

$$T_{w,ad} = T_g - \eta_c (T_g - T_3) \quad (30)$$

The calculation of temperatures at the flame tube wall T_{w1} (inner surface) and T_{w2} (outer surface) involves the following heat balance:

$$Rad_1 + C_1 = Rad_2 + C_2 = k_w (T_{w1} - T_{w2}) / t_w = K_{1-2} \quad (31)$$

where Rad_1 is the heat transferred by radiation and C_1 is the heat transferred by convection to the inner flame tube; Rad_2 is the heat transferred by radiation and C_2 the heat transferred by convection from the outer flame tube surface; and k_w is the wall thermal conductivity. The procedures to calculate R_1 , C_1 , R_2 and C_2 can be found in Lefebvre (1983).

2.8. Holes

Firstly it necessary verifies if there is enough air to enter through every combustion chamber zone holes. It is determined how much air enters through the primary zone, the secondary zone and the dilution zone from the difference between the total amount of air to enter each zone and the amount of air that enters through other means (for example, through the swirler or slots).

After the determination of the mass flow rate in each zone and the hole type, plain or plunged; then it is possible obtain the hole area for each zone. This is an interactive process that follows the related sequence.

- 1 – Determination of the bleed ratio (B), which is the ratio between the hole mass flow rate (\dot{m}_h) and the annulus mass flow rate (\dot{m}_{an}).
- 2 – The hole discharge coefficient ($C_{d,h}$) in the beginning of the interactive process is esteemed as 0.5.
- 3 – Equation 32 estimates the holes total area (A_h) for each row, and $\Delta P_h/P_3$ is approximately 6%.

$$\frac{\Delta P_h}{P_3} = \frac{143.5 \dot{m}_h^2 T_3}{P_3^2 C_{d,h}^2 A_h^2} \quad (32)$$

- 4 – Determination of the orifice area ratio ($\alpha = A_h/A_{an}$), and the bleed/orifice area ratio ($\mu = B/\alpha$).

5 – Equation 33 calculates the factor of pressures loss, K ; and δ , the factor of momentum loss, varies between 0.75 and 0.9 (Kaddah, 1964).

$$K = 1 + \delta^2 \{ 2\mu^2 + [4\mu^4 + (\mu^2 / \delta^2) (4B - B^2)]^{0.5} \} \quad (33)$$

6 – The value of the discharge coefficient ($C_{d,h}$) is recovered in the Eq. 34 (Kaddah, 1964).

$$C_{d,h} = \frac{(K-1)}{\delta [4K^2 - K(2-B)^2]^{0.5}} \quad (34)$$

If the value of $C_{d,h}$ calculated in the sixth step is close enough to the value estimated in the second step, then the right value has been found. If not, a new iteration is done. This process is repeated until a certain precision in the value of $C_{d,h}$ has been achieved or until a limit number of iterations. At the end of the process, the total area of the holes is obtained. The number of holes (N_h) of each combustion zone is a starting value in the program, which evaluates the holes diameter

$$d_h = 2 \sqrt{\frac{A_h}{\pi N_h}} \quad (35)$$

and verifies if the holes fit in the liner, that is, if the sum of the holes diameters is smaller than the liner. In the case of this condition is not satisfied, the program stops, because it is necessary to redefine the number of holes or to reconsider some parameter. When the holes are well dimensioned, the results of the jet penetration angles are presented using the recommendations of NREC Report 1082 (1964).

3. The Computational Program

Based on the methodology described before, a computational program in Delphi computational language was developed. This language was chosen due friendly interface with the user. Is not necessary to know the equations details to use the program, but it is necessary a basic knowledge about the operation conditions parameters and some aspects that must be assumed to combustor design.

The starter values or the results are presented in a sequence of screens; the total is seventeen, including the first one, which is the program presentation. Here just six screens are shown here, due the space limitation.

In the second screen, presented in Fig. 4, the user must inform the operation conditions. The program select four operation situations, because the aircraft gas turbines have four well defined operation situations: maximum thrust, maximum altitude, normal cruise, and idle. The program proceeds the calculation of D_{fl} for these four situations, and the user must choose the best value to accommodate the combustion process for both. The data pointed in Fig 2 correspond to a hypothetical case of multi-can combustor and they are presented in Malconian and Modak (1985), which permits to compare de program calculation with their results.

Condição	Dados de saída do compressor				Dados de entrada da turbina			Eficiência de combustão		
	P3 [MPa]	P2 [MPa]	T3 [K]	m3 [kg/s]	T4 [K]	η_{global}	TQ [%]	%	ml [kg/s]	$\Delta F_{34}/P_3$
1. Ponto de projeto Máxima tração	2	1.93	814	18.1	1500	0.347	20	99.7	0.427	0.07
2. Máxima altitude	0.7	0.68	707	6.9	1387	0.286	20	99.5	0.132	0.07
3. Cruzeiro Normal	1.8	1.77	1060	14.2	1393	0.145	20	99.5	0.140	0.07
4. Idle em solo	0.15	0.148	343	1.05	703	0.128	20	99	0.0091	0.07

Figure 4. Operation conditions (screen 2).

After the initial data operation, the program calculates the flammability limits, and based on these values the user must define how much air enter in primary zone. In sequence, the program shows the results for the reference length D_{ref} from the aerodynamics and combustion considerations, and for the four operation conditions (screen 5, Fig. 5).

Then, the user must define what is the best value for D_{ref} ; the higher value must be chosen because it accommodates all combustion situations and is also convenient for the aerodynamics considerations.

Based on the considerations presented in items 2.1.3 and 2.2, the program calculates the air mass flow rate in each zone and for film-cooling, and the zones length. The next step is the diffuser geometry, according the item 2.3. The user must specify the pressure loss across the diffuser and the discharge coefficient at primary air entrance, typical values are 0.01 and 1, respectively. The diffuser geometry is resumed in screen 7, Fig. 6.

The sequence is to determine the swirler. The user must specify the blades angle, air mass flow rate across the swirler (normally between 3 and 12% of total air), pressure loss factor (≈ 0.25), if the blades are straight or curved, number of blades, and some geometrics aspects, as presented in screen 9, Fig 7.

In the screen 10, Fig. 8, the recirculation region is calculated. The program recommends a value to recirculation region length, based on previous considerations; but the user can change it.

The next program step is temperature through the combustors zones, item 2.6. The program calculates the temperature for the four operation conditions. These values will be important to the film-cooling slots position, as commented in item 2.7. In the screen 12, Figure 9, the user must inform slot height s , slot wall thickness t and liner wall thickness t_w , the F_{dome} distance from the dome slot to the injector, the number of slots N_{slot} and after that their positions, apart from the selection of the casing material, steel or aluminum. Then, the program calculates the temperatures at the flame tube wall T_{w1} (inner surface) and T_{w2} (outer surface), for each operation condition. Based on these results about surface temperature, the user must accept or not the slots position.

Finally, the last program step is to calculate de holes of primary, secondary and dilution zones by the interactive process described in item 2.8. The user must specify the number of holes desired for each combustor zone.

The final screen presented by the program is a combustor geometry resume, Fig 11.

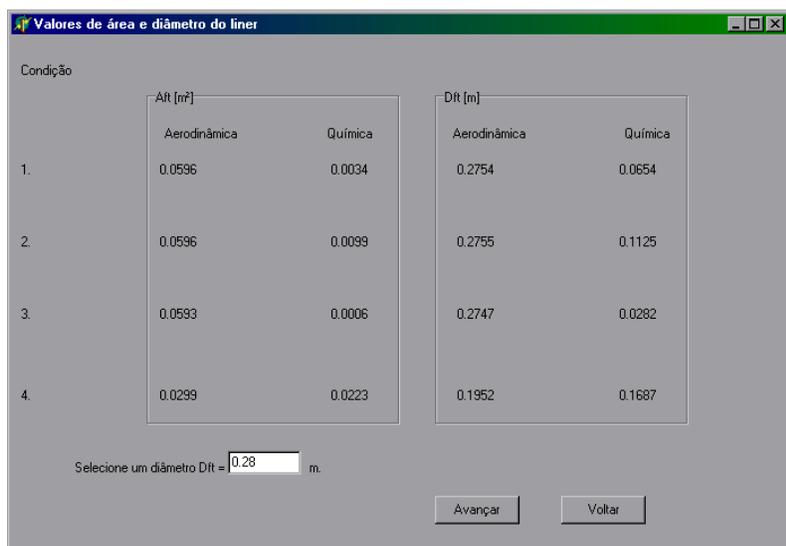


Figure 5. Values for A_{ft} and D_{ft} (screen 5).

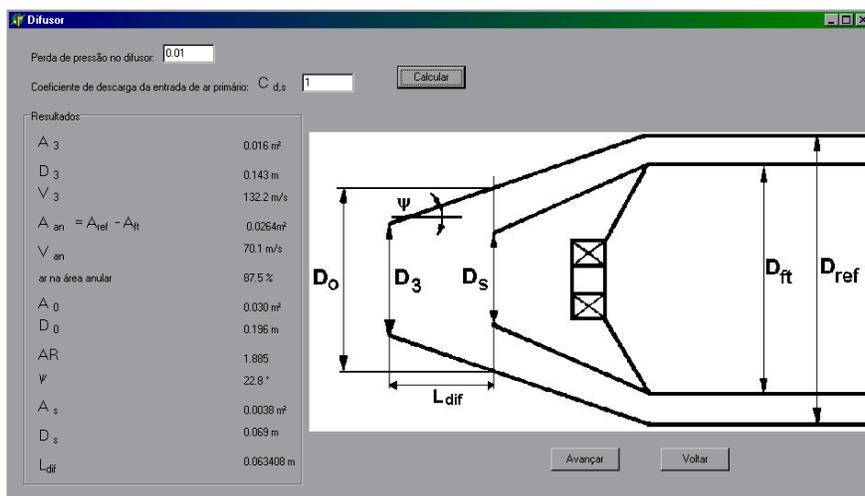


Figure 6. Diffuser geometry (screen 7).

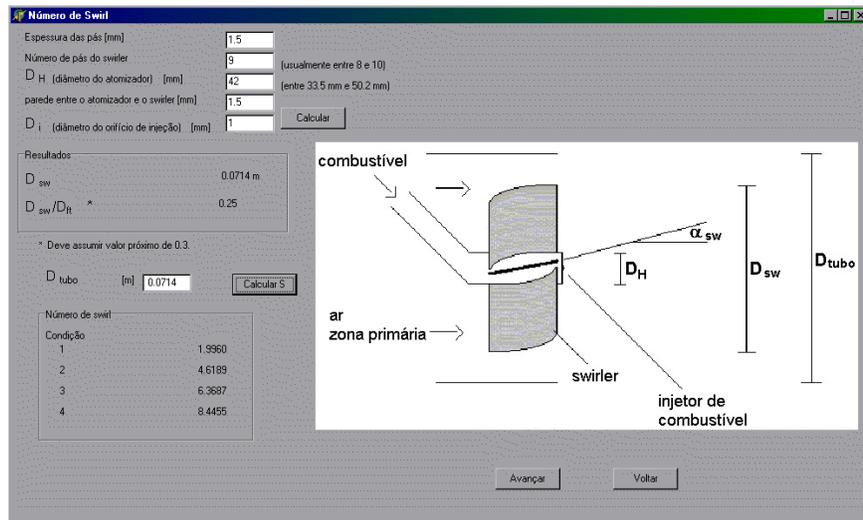


Figure 7. Swirler information (screen 9).

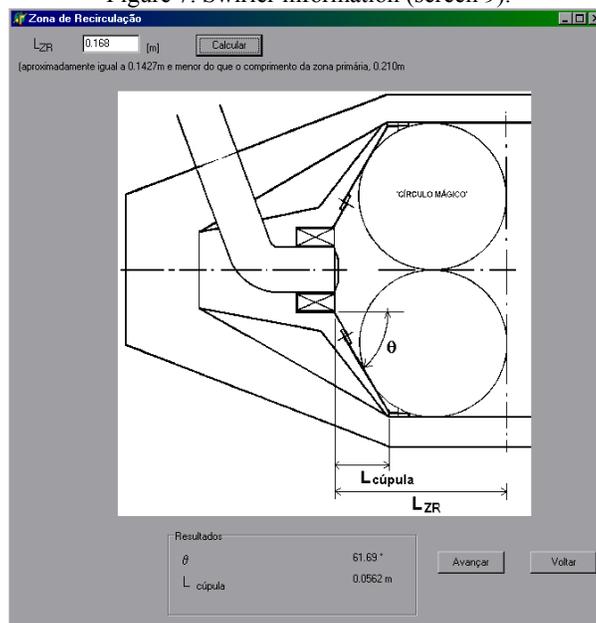


Figure 8. Recirculation zone (screen 10).

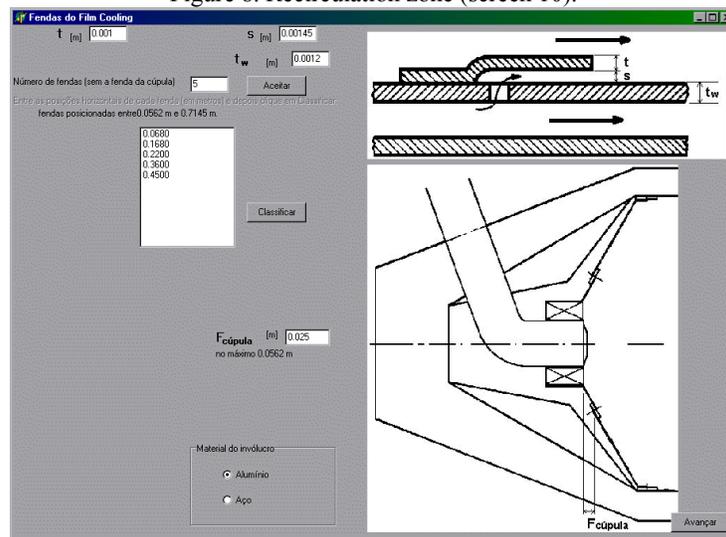


Figure 10. Film-cooling (screen 12).

4. Example of Design

The program presented was utilized to design a single chamber to be coupled in a gas-generator derived of a piston engine turbocharge. In fact, the complete system is a turbo-shaft composed of four gas-generators and one power turbine. Table A presents the design operating condition. Just one operate condition is utilized for the project, corresponding to 0.7 kg/h of air mass flow rate.

Table A. Design operating conditions.

P_3 [MPa]	p_3 [Mpa]	T_3 [K]	\dot{m}_3 [kg/s]	T_4 [K]	ϕ_{global}	TQ %
0.4	0.38	503	0.7	1100	0.347	20
$\eta_{combustion}$	\dot{m}_f [kg/s]	$\Delta P_{3-4} / P_3$	$\Delta P_{3-4} / q_{ref}$	V_3 [m/s] *	A_3 [m ²] *	Mach _{inlet} *
90	0.009676	0.05	20	135	0.00229	0

* where: V_3 is the velocity at the compressor exit; A_3 is the compressor exit area; and Mach_{inlet} is the Mach number distant the gas-generator entrance.

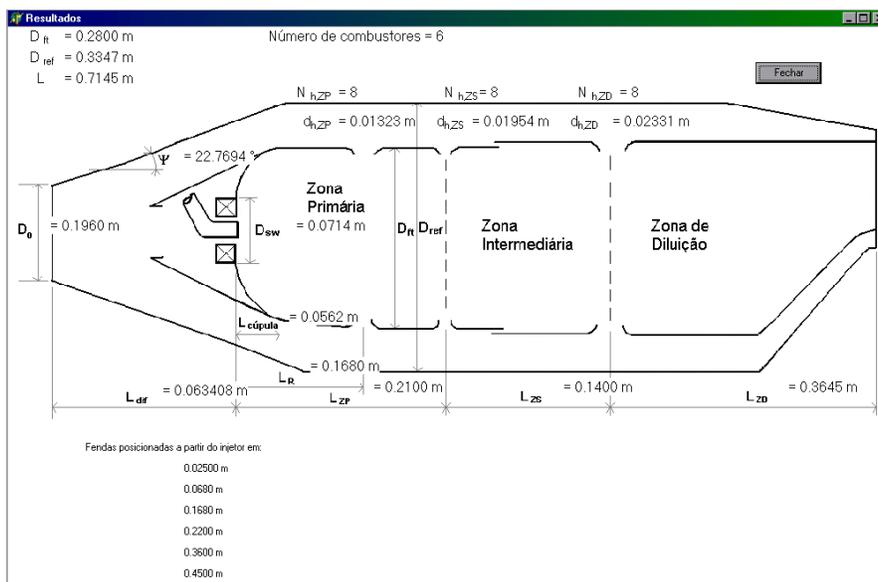


Figure 11. Screen of results resume.

The Fig. 12 presents a combustor (liner) photo. For the present application no wall cooling was necessary due the not so high temperature along the chamber. The Fig.'s 13 and 14 show some experimental results of gas-generator without load.



Figure 12. Combustor (liner) photo.

The results presented in Figure 13 are included in this paper to show that simplified methodology discussed here is able to provide a combustor configuration that attends the gas-generator operation range. At the present moment we can not compare the efficiency of the combustor designed by the methodology present here and, for example, any design refined by computational calculations. So that, we can say the present methodology probably do not provide the most efficient combustor chamber, but it provides a device that attends the gas-generator necessities. To analyze the combustion chamber performance, it is necessary an experimental set up where the fuel flow rate and the air flow rate are decoupled. The air is provided by a compressor independent of the combustion gases discharge; so, it is possible a better control of the combustion chamber operation. That will be next step of the present research project.

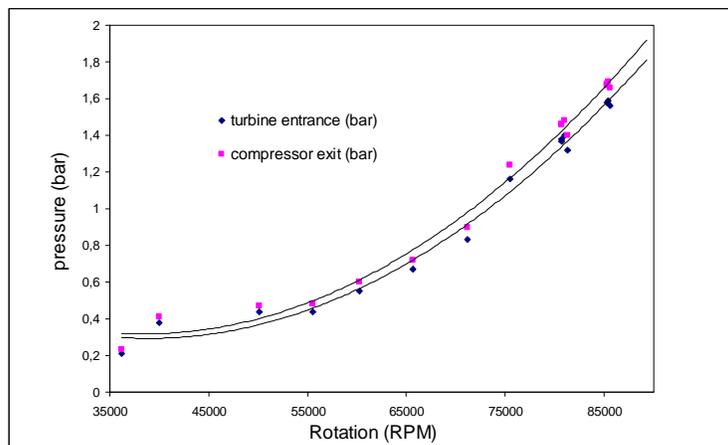


Figure 13. Pressure at the compressor exit and turbine entrance.

5. Conclusions

The present paper shows a methodology for gas turbine combustor basic design. The methodology emphasis is on the practical rather than theoretical aspects of combustor design. So that it has ability to design conventional combustors, and some different combustors concepts as LPP (Lean Pre-vaporized Premixed), RQL (Rich Quench Lean) or short combustors are not included in the present discussion.

The paper also shows the computational program developed based on the equations and methodology discussed, and some program screens were selected to show a calculation example purposed in a gas turbine combustor handbook. And finally, a combustor chamber designed using the computational program was presented to show the capacity to design a practical system.

As commented before, the objective of the present work is to provide conditions for producing designs that require minimum development time. So that it not eliminate some desired refinements in the combustor design based on experimental tests or computation fluid dynamic calculations. The results of this methodology must be utilized as preliminary design step, and, for an efficient final combustor configuration, some refinements are necessary. There are some parameters that can be distant from their real behaviour, due the simplifications adopted; for examples: the recirculation zone length and the temperature distribution. But it is easier to refine an initial configuration by a detailed computational calculation than a complex calculation for all design steps.

6. References

- Ballal, D.R., and Lefebvre, A.H., 1972, "A Proposed Method for Calculating Film-Cooled Wall Temperatures in Gas Turbine Combustion Chambers", ASME Paper 72-WA/HT-24.
- Conrado, A.C., 2002, "Metodologia para Projeto de Câmara de Combustão de Turbinas a Gás", Trabalho de Graduação, Instituto Tecnológico de Aeronáutica.
- Herbert, M.V., 1962, "A Theoretical Analysis of Reaction Rate Controlled Systems, Part II", Eighth Symposium (International) on Combustion, Williams and Wilkins Company, Baltimore, pp. 970-982.
- Kaddah, K. S., 1964, "Discharge Coefficients and Jet Deflection Angles for Combustion Liner Air Entry Holes, College of Aeronautics M.Sc. thesis, Cranfield, England.
- Knight, H.A., and Walker, R.B., 1957, "The Component Pressure Losses in Combustion Chambers", NGTER.143, ARCR&M2987, Aeronautical Research Council, England.
- Lefebvre, A.H. and Halls, G.A., 1959, "Some Experiences in Combustion Scaling", AGARD Advanced Aero Engine Testing, AGARDograph 37, Pergamon Press, New York, pp. 177-204.
- Lefebvre, A.H., 1966, "Theoretical Aspects of Gas Turbine Combustion Performance", CoA Note Aero 163, Cranfield Institute of Technology, Bedford, England.
- Lefebvre, A.H., 1983, "Gas Turbine Combustion", Taylor & Francis.
- Melconian and Modak, J.W., 1985, "Combustors Design", In Saywer's Gas Turbine Engineering Handbook, Theory & Design, Turbomachinery International Publications, Connecticut, Vol. 1.
- NREC Report 1082, 1964, "The Design and Performance of Gas Turbine Combustion Chambers", Northern Research and Engineering Corporation.
- Ogders, J. and Carrier, C., 1973, "Modeling of Gas Turbine Combustors; Considerations of Combustion Efficiency and Stability", ASME Journal of Engineering for Power, Vol.95, No.2, pp. 105-113.
- Ogders, J., 1980, "Combustion Modeling within Gas Turbine Engines", AIAA paper No. 77-52.
- Perry, J.H., 1950, "The Chemical Engineers Handbook", 3rd ed., McGraw Hill Book Company, New York, p. 388.