Analysis of Coolant Mass Flow Requirements for Transpiration Cooled Ceramic Thrust Chambers

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The long-term development of ceramic rocket engine thrust chambers at the German Aerospace Center (DLR) culminates in compact designs of transpiration-cooled fibre-reinforced ceramic rocket engine chamber structures. Achievable benefits of the transpiration cooled ceramic thrust chamber are the reduction of engine mass and manufacturing cost, as well as an increased reliability and higher lifetime due to thermal cycle stability. The transpiration cooling principle however reduces the engine performance. Due to the transpiration cooling the characteristic velocity decreases with increasing coolant ratio. The goal of the chamber development is therefore to minimize the required coolant mass flow.

The wall temperature can be calculated using known heat transfer correlations, for example given by Bartz, and employing a model given in literature for the reduction of the heat transfer coefficient based on coolant mass flow. By this method the required coolant mass flow ratio for different chamber diameters and pressure levels can be calculated. This paper discusses the application potential of DLR’s ceramic thrust chamber technology for high performance engines. Parametric variations of engine sizing (such as chamber pressure and diameter) are performed. For large diameters and high chamber pressures the required coolant ratio is below 1%.

Key Words: Ceramic Thrust Chamber, Chemical Propulsion, Transpiration Cooling, Systems Study

Nomenclature

\[ A \] : cross section area
\[ c^* \] : characteristic velocity
\[ c_p \] : specific heat capacity
\[ d \] : diameter
\[ F \] : thrust
\[ F \] : blowing ratio
\[ I \] : specific impulse
\[ l \] : length
\[ l^* \] : characteristic chamber length
\[ \dot{m} \] : mass flow
\[ Ma \] : Mach number
\[ O \] : surface area (wall)
\[ p \] : pressure
\[ Pr \] : Prandtl number
\[ \dot{q} \] : specific heat flux
\[ r \] : radius
\[ R \] : mass mixture ratio (oxidizer to fuel)
\[ St \] : Stanton number
\[ T \] : temperature
\[ v \] : velocity
\[ V \] : volume
\[ \alpha \] : heat transfer coefficient
\[ \epsilon_c \] : nozzle contraction ratio
\[ \kappa \] : isentropic exponent
\[ \mu \] : dynamic viscosity
\[ \omega \] : temperature exponent of viscosity equation
\[ \rho \] : density
\[ \sigma \] : boundary layer parameter (Bartz equation)
\[ \tau \] : coolant ratio

Subscripts

\[ 0 \] : initial (injection)
\[ aw \] : adiabatic wall
\[ c \] : chamber
\[ cyl \] : cylinder
\[ e \] : exit
\[ fu \] : fuel
\[ k \] : coolant
\[ ox \] : oxidizer
\[ sub \] : subsonic (converging) nozzle
\[ sup \] : supersonic (diverging) nozzle
\[ t \] : throat
\[ vac \] : vacuum
\[ w \] : wall (hot gas side)

1. Introduction

Experiments with porous Ceramic Matrix Composite (CMC) materials for rocket engine chamber walls have been conducted at the DLR since the end of the 1990s. Experiments with the ceramic thrust chamber were previously conducted at the European Research and Technology Test Facility P8 in Lampoldshausen. The experiments were part of the research project KSK (Keramische Schubkammer, German for ceramic thrust chamber). Most recently tests have been conducted at DLRs new P6.1 test bench.

The paper describes methods for evaluation of required coolant mass flow. Resulting minimal coolant flow for different chamber geometries and pressures are calculated and discussed.
2. Status and Applicability

2.1. Status

In 2008 and 2010 test campaigns with 50 mm chamber diameter Ceramic Matrix Composites (CMC) combustion chambers were performed. The campaigns took place at the European Research and Technology Test Facility P8 in Lampoldshausen. In 2011/2012 another test campaign followed on DLR’s new P6.1 test facility. The most recent test campaign was conducted between October and November 2012. The next test campaign is currently scheduled for October 2013. All test campaigns used the propellant combination Liquid Oxygen (LOX)/Liquid Hydrogen (LH2). This is the baseline propellant combination for DLR’s ceramic thrust chamber. It is selected both for specific impulse performance reasons, as well as the favorable coolant properties of hydrogen. Figure 1 shows a ceramic chamber during test operation. More detailed information on previous test campaigns and current status can be found in different publications.\(^1,2\)

![MTS-A thrust chamber at the P6.1 test bench in Lampoldshausen (March 2012).](image)

2.2. Applicability to a full scale rocket engine

The following subsections provide rough estimates for the expected benefits and issues for future flight hardware based on the current state of the technology research.

2.2.1. Mass estimation

The application of a transpiration cooled fiber ceramic thrust chamber potentially reduces the chamber mass. The chamber design and manufacturing process have previously been published.\(^3\) Depending on the applied ceramic materials the mass of the chamber, excluding nozzle and injector, can be reduced by 30\% - 70\% compared with conventional metallic chambers. In a typical engine mass breakdown the chamber mass (excluding injector and nozzle) amounts to roughly 10\% of the total engine mass. Mass savings for application of transpiration cooled fiber ceramic thrust chambers can therefore be expected to be around 5\% of the total engine mass. The engine mass of a sea level engine impacts only little on the payload mass due to staging of the vehicle (except for the case of future Single-Stage To Orbit (SSTO) applications). Even for an upper stage application, where the mass saving directly translates into increased payload mass, the total benefit seems negligible. Consequently the potential mass savings of ceramic thrust chambers, while being beneficial for the launcher system, are not the driver of the technology development.

2.2.2. Cost estimation

Combustion chamber manufacturing cost can be assumed to be roughly 10\% of the engine manufacturing cost.\(^4\) Raw material cost of ceramics offers some potential of savings compared with the increasing costs of metals used for conventional chamber design. The main advantage of the fiber ceramic combustion chamber however lies in the manufacturing process. Chamber segments can be manufactured and machined individually and then be stacked together to form the full chamber, instead of milling the whole chamber from one large block. The time consuming and costly process of galvanization, used in modern regeneratively cooled chambers, is not required. Based on prototype manufacturing cost of the test specimen, estimated cost savings of chamber manufacturing in the order of 50\% seem feasible. For a typical launcher cost savings in the order of one or more million Euro are therefore to be expected.

2.2.3. Engine re-usability

In the late 1960s Pratt & Whitney developed the transpiration cooled XLR-129 rocket engine, with a chamber pressure of approximately 10 MPa. The engine was extensively tested. Based on the results Pratt & Whitney developed in the 1970s a transpiration cooled design for the Space Shuttle Main Engine (SSME). Transpiration cooling was selected in order to fulfill the NASA criteria of 100 time engine re-usability.\(^5\) This engine development of Pratt & Whitney is the only known experimental study dealing with transpiration cooled engine life cycle, durability and re-usability to date. Based on the published results of Pratt & Whitney and theoretical considerations, the life time of transpiration cooled chambers is expected to be at least 10 times higher than that of regeneratively cooled chambers. While this advantage of the discussed technology can only be fully utilized in Reusable Launch Vehicle (RLV), a high robustness associated with re-usability is beneficial for any application.

2.2.4. Wall-material chemical stability

Concerning transpiration cooled fiber ceramic engines with cryogenic propellants one main issue remains. Even though the previously used C/C material is a high temperature resistant and undergoes temperatures up to 2000 K during the pyrolysis of the manufacturing process, this applies to inert gas environment. Figure 2 shows the typical chemical environment in a LOX/LH2 rocket engine thrust chamber. When exposed to combustion chamber hot gases, and especially to oxygen, the wall will react with the gases at much lower temperatures. The test-campaign KSK-KT resulted in wall damage near the injector in all "hot" test runs.\(^6\) In the following test-campaign KSK-ST5 the wall material close to the injector was substituted with aluminum-oxide ceramics. No damage to the wall was observed during and after the test campaign.\(^7\) The level of cooling was however extremely high.
2.3. Reference application

Due to the scaling effects large diameter and high pressure applications are favorable for transpiration cooling. This is mainly because the coolant mass flow introduced into the chamber creates a film on the chamber wall and similarly to film cooling constitutes a loss in chamber performance in terms of specific impulse (cf. Eq. (5)).

For larger diameters the ratio of chamber inner surface to chamber cross section area improves. In other words, in large combustion chambers there is less wall surface area per hot gas flow area.

Large pressures are beneficial because the density and thereby the mass flow increase almost linearly with the chamber pressure, while the specific heat flux increases slightly less, to the power of 0.8 (cf. Bartz’ equation in Eq. (11)). For high pressure and large diameter applications the losses incurred due to transpiration cooling are therefore minimal.

Figure 3 shows thrust levels and chamber pressures for a wide selection of current and historical liquid propellant rocket engines. Radiation cooling can only be applied where the specific heat fluxes are small enough to allow for heat transfer according to Stefan-Boltzmann law. At higher chamber pressures regenerative cooling has to be applied. For very large heat fluxes regenerative cooling is at its limit. On the other hand transpiration cooling becomes increasingly efficient with large diameters. The application range of transpiration cooled rocket engines is therefore in the upper right corner of Figure 3.

Due to replacing of the conventional cooling channels with transpiration cooling, the hydrogen which is supplied to the injector cannot be heated by the chamber cooling process. Cold hydrogen with temperatures below \( T \approx 50 \, \text{K} \) often leads to instability in the combustion. In an engine with an open cycle it would therefore be necessary to somehow heat the hydrogen, which is supplied to the injector (e.g. by adding a heat exchanger to the gas generator exhaust). In a staged combustion cycle, the hydrogen has already been routed through the preburner, before being fed to the main chamber. Thus in staged combustion engines the main chamber injector is provided with warm hydrogen, irrespective of the chamber cooling method. The transpiration cooling therefore in this case does not require further modifications or added components.

2.3.1. Staged combustion cycle engines

Staged combustion cycle engines utilize the entire propellant mass flow for propulsion. They were first developed in the late 1950s in the Soviet Union \(^7\) and are characterized by high chamber pressures and high efficiency. This comes at the cost of an increased engine mass and complexity.

The staged combustion engine is highly suitable for application of transpiration cooling. High chamber pressures and (depending on the desired thrust level) high chamber diameters keep the required coolant fraction low. Since the propellants used in the injector have passed through the preburners, they reach the injector at elevated temperatures, even if there is no preheating during chamber scaling, due to the substitution of the regenerative chamber cooling. The combustion efficiency of the main chamber therefore, as discussed in the previous section, is not influenced by the application of transpiration cooling. The staged combustion cycle thus allows utilization of the benefits of transpiration cooled CMC wall material (namely increased life time, reduced manufacturing cost and reduced chamber weight) without significant disadvantages. Figure 4 shows schematically a staged combustion engine’s cycle, with fuel-rich preburners and main combustion chamber transpiration cooling.
3. Numerical Analysis

3.1. Chamber geometry

Figure 5 illustrates the contour parameters used to define the axis-symmetric combustion chamber.

Important values for comparison of thrust chamber geometries are the so-called expansion ratio ($\epsilon$) and contraction ratio ($\epsilon_c$) as defined below:

$$\epsilon = \frac{A_e}{A_t} = \left(\frac{d_e}{d_t}\right)^2$$  \hspace{1cm} (1)

$$\epsilon_c = \frac{A_k}{A_e} = \left(\frac{d_k}{d_e}\right)^2$$  \hspace{1cm} (2)

3.2. Normalised transpiration-massflow

Different sources each use different approaches to normalise the coolant mass flow. In this paper the normalisation by the total mass flow is used. In the following sections the definition from Eq. (3) therefore applies.

$$\tau = \frac{\dot{m}_k}{\dot{m}_e} = \frac{\dot{m}_k}{\dot{m}_0 f_u + \dot{m}_0 o_x + \dot{m}_k}$$  \hspace{1cm} (3)

The engine mixture ratio thus relates to the injector mixture ratio as follows:

$$R_e = \frac{R_0 (1 - \tau)}{1 + \tau R_0}$$  \hspace{1cm} (4)

The coolant ratio ($\tau$) directly impacts on the specific impulse of the thrust chamber due to the imperfect mixing of coolant mass flow with the hot gases. Part of the coolant mass flow is not adequately accelerated in the thrust chamber and therefore does not contribute to the total thrust. Due to the mass balance the engine's specific impulse (as defined in Eq. (5)) is therefore reduced by transpiration cooling. Another effect which needs to be considered is the difference between combustion chamber and total engine mixture ratios. Transpiration cooling with fuel reduces the total engine mixture ratio in relation to the injection mixture ratio. Low mixture ratios, while favorable in terms of specific impulse, are usually avoided in order to prevent excessively large hydrogen tanks, to optimize the launcher system as a whole. The overall design goal is therefore to minimize the required coolant ratio $\tau$ in order to maximize system performance.

$$I = \frac{F}{\dot{m}} = \frac{F}{\dot{m}_0 f_u + \dot{m}_0 o_x + \dot{m}_k}$$  \hspace{1cm} (5)

3.3. Description of models

To estimate the stationary wall temperature a model validated on measurements conducted in the framework of the German research program Transregio 40 (TRR40) is used. The model was validated in a square hot gas channel with flat samples and for hot gas temperatures of up to 540 K, pressures of 0.1 MPa and Mach numbers between 0.3 and 0.7.\(^8\) The application of this model to combustion chamber conditions is discussed in other publications.\(^9,10\) The focus of this paper is the inclusion of heat transfer correlations for convergent combustion chamber geometries.

3.3.1. Hot wall temperature calculation

The model is based on the heat balance of the transpiration cooled wall given in Eq. (6). The hot wall temperature is given by $T_w$.

$$\dot{q} = \alpha (T_{aw} - T_w) = \frac{\dot{m}_k}{\dot{m}_e} c_{ph} (T_w - T_k)$$  \hspace{1cm} (6)

It is assumed, that the coolant enters the wall at the coolant temperature ($T_k$) and leaves the chamber wall at the hot wall temperature ($T_w$), thus assuming thermal equilibrium between coolant and solid. This assumption is valid for large enough volumetric heat exchange coefficients of the porous wall material. Experiments in TRR40
indicate that this is the case for the currently used CMC materials. The heat transfer coefficient (α) is dependent on hot gas conditions as well as blowing ratio (F) and is determined via the Stanton number. The method for determination of the reduced heat transfer coefficient is given by Kays et al. and consists of Eq. (7) and Eq. (10).

\[
\frac{St}{St_0} = \frac{b_h}{e^{b_h} - 1} \quad (7)
\]

with the blowing ratio:

\[
F = \frac{\dot{m}_k/Oh}{\dot{m}/A} \quad (8)
\]

and the Stanton number:

\[
St = \frac{\alpha}{c_p \rho \nu} \quad (9)
\]

and \( b_h \):

\[
b_h = \frac{F}{St_0} \left( \frac{c_{p0}}{c_p} \right)^{0.6} \quad (10)
\]

The model therefore needs a Stanton number without cooling, \( St_0 \), and predicts a reduced Stanton number dependent on blowing ratio and the ratio of heat capacities of hot gas and coolant. The Stanton number without any cooling is crucial in this model and has to be determined from correlations. For the test campaigns of the TRR40 research program, the Dittus-Boelter relationship yielded good results. For combustion chambers a correlation developed by Bartz is usually applied.

\[
\alpha = 0.026 \frac{d^2}{\mu} \left( \frac{P_0}{P_{\infty}} \right)^{0.2} \left( \frac{P_{\infty}}{P_0} \right)^{0.8} \left( \frac{d_t}{r_{sub}} \right)^{0.1} \left( \frac{A_t}{A} \right)^{0.9} \sigma \quad (11)
\]

with the characteristic velocity \( c^* \)

\[
c^* = \frac{A_t p_{\infty}}{\dot{m}} \quad (12)
\]

\( \sigma \) in Eq. (11) accounts for the change of hot gas properties over the boundary layer and is given by Bartz as

\[
\sigma = \frac{1}{\left[ 0.5 \frac{T_w}{T_0} \left( 1 + \frac{\kappa-1}{2} Ma^2 \right) + 0.5 \right]^{0.8}} \left[ 1 + \frac{\kappa-1}{2} Ma^2 \right]^{\frac{3}{2}} \quad (13)
\]

with \( \omega \), the temperature exponent of the viscosity equation, set to 0.6 and \( \kappa \) the ratio of specific heats. Bartz gives an applicability of the correlation for contraction angles \( (\theta_{sub}) \) between 15° and 45° and ratios of throat diameter to curvature radius \( (d_t/r_{sub}) \) up to 3.

According to the model described by Eq. (6) through Eq. (13) transpiration cooling has two effects. First it reduces the Stanton Number and thereby the hot gas sided heat transfer coefficient due to the injection of the coolant in the boundary layer. Secondly the heat capacity of the coolant absorbs the steady state heat flux (cf. Eq. (6))

### 3.3.2. Description of algorithm

The code used in this paper has been implemented in Python. The algorithm for determination of a wall temperature with transpiration cooling is shown in Figure 6. First, Chemical Equilibrium with Applications (CEA) is used to determine the hot gas properties. For the Bartz equation (Eq. (11)), the stagnation point properties of heat capacity, Prandtl number, viscosity, density and temperature are needed. Local values for a specific location in the chamber are needed for pressure, Mach number and the ratio of heat capacities. Given coolant reservoir pressure and temperature are used to calculate mean coolant properties for chamber and reservoir values. The specific mass flow of the coolant is adjusted along the x-axis to get a near constant wall temperature distribution. Since the heat transfer coefficient is dependent on the wall temperature, the final wall temperature is iterated. In each iteration the Bartz equation is solved and a heat transfer coefficient along the chamber axis is calculated. Next, for each point the reduced Stanton number is determined using the model of Kays et al. The reduced heat transfer coefficient taken from this calculation is then used in the heat balance given in Eq. (6). The new wall temperature is compared to the wall temperature previously used in the Bartz equation. If the difference is below a certain threshold, the iteration ends.

### 3.3.3. Code validation

Since heat flux measurements with transpiration cooled chambers are difficult, no direct validation of the model has been performed yet. The implementation of the Bartz equation (Eq. (11)) used in this paper, was compared with published data from sub scale tests. Depending on the application of frozen or shifting equilibrium transport properties, the calculated heat flux matches experimental data. Figure 7 shows the contour and measured heat flux from the experiments of Dexter et al. as well as the
calculated heat flux.

Calculations for fully frozen and fully shifting equilibrium are shown. Especially the specific heat capacity \(c_{p,h}\) which linearly affects the heat transfer coefficient \(\alpha\) changes substantially. For the case described by Dexter et al. fluid properties 67% frozen and 33% shifting equilibrium led to a good accordance of measured and calculated data. Depending on the chamber pressure the applicable ratio shifts. For high pressure cases the fitting solution trends towards shifting equilibrium. For low pressure cases it trends towards frozen equilibrium. For the application of the model and the calculations presented in the following section, shifting equilibrium data was always applied. As can be seen in Figure 7, this constitutes a positive margin. The estimations of the required coolant mass flow are therefore a conservative estimate.

It also has to be noted that the heat load in the cylindrical chamber is overestimated. The initiation of the combustion and the associated rise in heat flux, starting from very low levels near the injector, is not part of the model discussed in this paper. Instead a constant heat flux in the cylindrical chamber is assumed. The resulting integration for calculation of required coolant mass flows therefore also constitutes a conservative estimate.

For a better understanding of the model the reduced heat flux, calculated for wall temperatures of \(T_w = 800\) K is plotted in Figure 7 as well. The reduced heat flux based on the methods of Kays et al.\(^{11}\) is used in the calculations presented in the following section. Comparison with experimental data from DLRs transpiration cooled CMC thrust chamber test campaign WS1b has been performed. Figure 8 shows the temperature evolution during a low mixture ratio test with the ceramic thrust chamber. Position 1 was a thermocouple 2 mm inside the wall at an angular position of 135°. Due to an unsymmetrical flame formation and associated heat flux distribution in the chamber, the measured temperatures are similar, although they were registered at different distances from the inner wall. The temperature measured in the coolant reservoir is shown for comparison. Those thermocouples were positioned at the same axial position, at -87.5 mm, near the end of the cylindrical part of the chamber. Table 1 list some test parameters of this test run. After 8 s of test duration the transpiration cooling was shut down to switch to the uncooled part of the test. Until that time steady state was not reached. It can however be seen that the calculated steady state wall temperature for the hot gas conditions is well above the final temperatures. The wall temperature calculation has been performed with equilibrium transport properties for the hot gas condition. This is another indication for the margins of the conservative calculations presented in this paper.

Fig. 7. Calculated heat flux on chamber wall of approx. 800 K.

For a better understanding of the model the reduced heat flux, calculated for wall temperatures of \(T_w = 800\) K is plotted in Figure 7 as well. The reduced heat flux based on the methods of Kays et al.\(^{11}\) is used in the calculations presented in the following section. Comparison with experimental data from DLRs transpiration cooled CMC thrust chamber test campaign WS1b has been performed. Figure 8 shows the temperature evolution during a low mixture ratio test with the ceramic thrust chamber. Position 1 was a thermocouple 2 mm inside the wall at an angular position of 135°. Due to an unsymmetrical flame formation and associated heat flux distribution in the chamber, the measured temperatures are similar, although they were registered at different distances from the inner wall. The temperature measured in the coolant reservoir is shown for comparison. Those thermocouples were positioned at the same axial position, at -87.5 mm, near the end of the cylindrical part of the chamber. Table 1 list some test parameters of this test run. After 8 s of test duration the transpiration cooling was shut down to switch to the uncooled part of the test. Until that time steady state was not reached. It can however be seen that the calculated steady state wall temperature for the hot gas conditions is well above the final temperatures. The wall temperature calculation has been performed with equilibrium transport properties for the hot gas condition. This is another indication for the margins of the conservative calculations presented in this paper.

Table 1. Test conditions for test WS1b-H11 (Nov. 2012).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chamber diameter</td>
<td>50 (mm)</td>
</tr>
<tr>
<td>Throat diameter</td>
<td>20 (mm)</td>
</tr>
<tr>
<td>Mixture ratio</td>
<td>2</td>
</tr>
<tr>
<td>Chamber pressure</td>
<td>4.5 (MPa)</td>
</tr>
<tr>
<td>Coolant ratio</td>
<td>2.7 %</td>
</tr>
</tbody>
</table>

4. Coolant Requirements

By a set of equations the required mass flow to keep the wall temperature below a specific operational temperature limit can be estimated. Taking the specific ceramic wall temperature limits into account, this estimation, in combination with the previously discussed associated per-
formance losses, allows to establish the operational boundaries of a transpiration cooled rocket engine.

4.1. Sample case

The calculations presented in this section use the data listed in the following subsections for sample calculations.

4.1.1. Scaled chamber geometry

The calculations are made for five different geometries. The parameters defining the contours are listed in Table 2 (cf. Figure 5 on page 4). Figure 9 shows the superimposed contours. The point of origin is the center of the throat. The abscissa therefore lists negative positions for the subsonic chamber geometry.

![Fig. 9. Chamber contours used for heat transfer calculations.](image)

Table 2. Chamber geometries used for diameter scaling.

<table>
<thead>
<tr>
<th>(d_{c} ) [mm]</th>
<th>50</th>
<th>100</th>
<th>200</th>
<th>440</th>
<th>1000</th>
</tr>
</thead>
<tbody>
<tr>
<td>(d_{t} ) [mm]</td>
<td>20</td>
<td>50</td>
<td>113</td>
<td>290</td>
<td>725</td>
</tr>
<tr>
<td>(d_{e} ) [mm]</td>
<td>141</td>
<td>354</td>
<td>799</td>
<td>205</td>
<td>5127</td>
</tr>
<tr>
<td>(\epsilon ) [-]</td>
<td>50</td>
<td>50</td>
<td>50</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>(\epsilon_{c} ) [-]</td>
<td>6.25</td>
<td>4</td>
<td>3.13</td>
<td>2.3</td>
<td>1.9</td>
</tr>
<tr>
<td>(\theta_{sub} ) [deg]</td>
<td>25°</td>
<td>25°</td>
<td>25°</td>
<td>25°</td>
<td>25°</td>
</tr>
<tr>
<td>(l_{cyl} ) [mm]</td>
<td>138</td>
<td>204</td>
<td>226</td>
<td>230</td>
<td>63</td>
</tr>
<tr>
<td>(l_{sub} ) [mm]</td>
<td>41</td>
<td>76</td>
<td>143</td>
<td>289</td>
<td>616</td>
</tr>
</tbody>
</table>

The chamber diameters are arbitrarily selected in order to represent typical engine diameters. The throat diameter is scaled according to the text book method by Humble et al., as shown in Eq. (14). Here the throat diameter is normalized by \(d_{a, norm} = 10 \) mm. The chambers are scaled to a characteristic chamber length of \(l^* = 1 \) m. The characteristic chamber length \(l^*\) defined as the ratio of chamber volume to throat area ratio (Eq. (15)), is related to the stay time of the propellants in the combustion chamber. For the propellant combination LOX/LH2 a choice of \(l^* = 1 \) m is a conservative value.

\[
\epsilon_{c} = 8 \left( \frac{d_{e}}{d_{a, norm}} \right)^{-0.6} + 1.25 \quad (14)
\]

\[
l^* = \frac{V_{c}}{A_{t}} \quad (15)
\]

4.1.2. Fluid properties: hot gas

The fluid properties of the hot gases were calculated for equilibrium conditions by use of CEA.\textsuperscript{14} The option for Finite Area Combustor (FAC) was used, in order to differentiate between the varying contraction ratios. The injector mixture ratio was set to \(R_{0} = 6\). Calculations were made for four different total pressures (\(p_{0} = 2 \) MPa, 6 MPa, 10 MPa, 20 MPa). Table 3 lists fluid properties for the initial conditions as calculated by CEA.

Table 3. Calculated initial conditions for hot gas fluid properties.

<table>
<thead>
<tr>
<th>(p_{c} )</th>
<th>(T )</th>
<th>(\kappa )</th>
<th>(c_{p} )</th>
<th>(\rho )</th>
<th>(Pr )</th>
</tr>
</thead>
<tbody>
<tr>
<td>2 MPa</td>
<td>3433 K</td>
<td>1.1308</td>
<td>12.33</td>
<td>0.92</td>
<td>0.507</td>
</tr>
<tr>
<td>6 MPa</td>
<td>3577 K</td>
<td>1.1374</td>
<td>10.23</td>
<td>2.67</td>
<td>0.523</td>
</tr>
<tr>
<td>10 MPa</td>
<td>3642 K</td>
<td>1.1405</td>
<td>9.37</td>
<td>4.40</td>
<td>0.532</td>
</tr>
<tr>
<td>20 MPa</td>
<td>3727 K</td>
<td>1.1448</td>
<td>8.33</td>
<td>6.25</td>
<td>0.545</td>
</tr>
</tbody>
</table>

4.1.3. Fluid properties: coolant

The used fluid data were taken from the National Institute of Standards and Technology (NIST) chemistry web-book.\textsuperscript{17} The data are evaluated temperature and pressure dependent.

4.1.4. Wall material properties

For the method presented in this paper local thermal equilibrium between coolant and chamber wall is assumed. The steady state chamber wall temperature, as a function of the coolant mass flow is calculated. Based on those boundary conditions no actual material properties are required for the calculations. The wall material is assumed to be a chemically resistant CMC material with high allowable operating temperatures. Recent tests during the campaign WS1a in October 2012 showed promising results with C/SiCN material.

4.2. Model results for combustion chambers/coolant requirements

The models described in the preceding sections are applied to the scaled chambers cases given in Table 2. Particular attention is paid to the scaling effects occurring when increasing the chamber diameter.

With the models described in the preceding section, the operational domain of a ceramic thrust chambers of various inner diameters can be evaluated. The required area specific coolant mass-flow is constant with increasing chamber diameter, meaning the coolant mass flow scales linear with chamber diameter. At the same time, the hot gas
mass flow with constant pressure and velocity is scaling with the cross section area (i.e. the square of the diameter). By increasing the diameter of a combustion chamber, the percentage of coolant mass flow of the total mass flow is decreasing. This is illustrated in Figure 10 and Figure 11. In this figures, for a given allowable CMC wall temperature and different chamber diameters, the resulting required percentage of coolant mass flow to total mass flow (cf. $\tau$ in Eq. (3)) can be seen. For comparison tests of the most recent campaign WS1 are plotted. The initial mixture ratio was $R_0 \approx 5.5$. The minimum coolant ratio applied in this campaign without damage to the chamber wall was $\tau = 7.05\%$. The thrust was not measured during the test campaign. It is calculated based on measured mass flow and chamber pressure, as well as CEA calculation of fluid properties and velocity.

![Figure 10. Required coolant mass flow for different chamber diameters and a maximum wall temperature of $T_w = 800\, K$. Test results of the campaign WS1.](image1)

As can be seen the scaling curve for the chamber diameter $d_w = 50\, mm$ (which is the chamber diameter used during the test campaign) roughly fits test conditions for the lowest coolant ratio test. With current chamber materials, scaling behavior as plotted in Figure 10 is therefore expected. As can also be noted the required coolant ratio is extremely large for small diameter, low pressure applications. For chamber diameters of more than $d_w \geq 400\, mm$ it becomes feasible to operate the chamber with coolant ratios around $\tau \approx 1\%$. If the mean hot wall temperature can be increased to $T_w = 1200\, K$ by further material development, the required coolant ratio can be decreased by roughly $\frac{1}{2}$ compared with the demand at $T_w = 800\, K$ wall temperature. Applications in the thrust range above $F_{vac} \geq 100\, kN$, with coolant ratios below $\tau < 2\%$ are moderately efficient and therefore worth of consideration. The most efficient applications of transpiration cooling, with coolant ratios below $\tau < 0.5\%$, lie however in the thrust range well above $F_{vac} \geq 1000\, kN$.

![Figure 11. Required coolant mass flow for different chamber diameters and a maximum wall temperature of $T_w = 1200\, K$. Test results of the campaign WS1.](image2)

5. Validation Experiment

In previous test campaigns an exact correlation between coolant mass flow and wall temperature could not be established. The previously used wall material C/C is very temperature resistant but sensitive to oxidation. Furthermore the matrix of this CMC is prone to erosion. Another effect inherent with transpiration cooling is the possibility of near wall combustion. This may occur when the injected propellants are not sufficiently mixed by the injector. The oxidizer may then react with the fuel which is fed through the wall as coolant, even though there is a surplus of fuel available at the injector. For those reasons in previous test campaigns the walls were usually overcooled in order to avoid damage.

In order to validate the numerical methods, described in the previous section, experiments are planned to be performed as part of the upcoming test campaign end of 2013. Figure 12 shows schematically the test configuration for the planned campaign. The first four segments will use a dump cooling as test for regenerative cooling of CMC materials. The fifth segment will be the principal test segment for transpiration cooling. The mixture ratio will be set below $R = 2$ to keep wall temperatures below $T_w = 2000\, K$. Thermocouples will be directly attached to the inner wall. Test goal is to reach steady state operation for validation of the developed numerical models.

6. Conclusion and Outlook

Concerning propulsion performance, transpiration cooling imposes disadvantages on the engine in terms of achievable specific impulse. On the other hand, porous fiber ceramics allow substantial savings in manufacturing
The transpiration cooling concept, coupled with low thermal expansion wall material, is expected to lead to increased engine life time and robustness. Concerning combustion stability and system performance the staged combustion cycle provides an inherent benefit for the application of transpiration cooling.

Thermodynamic considerations indicate that full scale transpiration cooled ceramic combustion chambers, with inner diameters larger than 400 mm, require low amounts of transpiration coolant ($\tau \leq 1\%$). Future work includes further experimental and theoretical studies, in order to validate and improve the model used for the calculations presented in this paper.

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