The design and testing of a can-type combustor in a compressor test rig.

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ABSTRACT

In order to increase the power of a blowdown, turbine driven compressor test rig, a combustor was designed to insert in the flow before the turbine. The combustor was required to provide 800K output temperature in order to achieve this. The combustor was designed for two conditions, 1) a start-up regime and 2) a target design point. Using the NREC preliminary combustor design method, a can-type combustor was designed. During the design process, the various method of temperature prediction resulted in 3 different outlet plane temperatures. In order to enable future confidence in the design method used, an experiment was setup to determine which models more closely represent reality. The temperature at the outlet plane was measured using a thermocouple rake during the running of the combustor in the test rig. It was found that the AFT Graph temperature predictions were the closest to the experimental data however; the results depend on the method for predicting the efficiency. It is recommended that a future experimental study be performed in order to confirm the best efficiency model using emissions measurements.

Keywords: Combustor; outlet temperature; performance

NOMENCLATURE

AFT   Adiabatic flame temperature
Av.   Average
Calc.   Calculated
Comp.   comparison
Cond   Condition
Exp.   Experimental
g   gauge
LHV   Lower heating value
No.   Number
NREC   Northern Research and Engineering Corporation
param   parameter
pf   Pattern factor
Pred.   Predicted
pt.   point
Pz   Primary zone
Rz   Recirculation zone
SUN   Stellenbosch University
Unc.   Uncertainty

Symbols

ϕ Equivalence ratio or diameter
α Blade stagger angle
β Airflow turning angle
η Efficiency
θ Theta parameter [1]
ΔP Pressure drop
$\dot{m}$ Mass flow rate
A   Area
AFR   Air/fuel ratio
d   Liner diameter
D   Casing Diameter
FAR   Fuel/air Ratio
FARi   Fuel/air Ratio calculation terms [2]
K   Swirler blade type factor
L   Length
p   Static pressure
P   Total pressure
q   Dynamic pressure
T   Temperature at the outlet to the combustor

Subscripts

3   Station at the inlet to the combustor
4   Station at the outlet to the combustor
f   Fuel
ft   Flame tube/Combustor liner
gas   Combustion gasses inside the combustor liner
Pz   Primary zone
ref   Combustor reference parameters, usually place of maximum cross-sectional area
sw   Swirler

1.0 INTRODUCTION

In order to test the compressor designs during the development process of micro-gas turbines before they are integrated into the engine, a compressor test rig at Stellenbosch University (SUN) is used. The test rig initially consisted of compressed air tanks which were blown down through a turbine, which in turn spun the compressor. A throttling valve was used after the compressor to regulate the compressor running speed. Subsequently, a new engine design has commenced for a 1000N engine which requires a compressor with a higher performance specification. This higher pressure design in turn requires a larger power input in the order of 350kW to attain the higher performance specifications. It was determined that the compressor test rig as it is with an atmospheric temperature and 10 bar blowdown capability, could not produce enough power to drive the designed compressor at the required operating condition. It was determined that adding a combustor into the blowdown facility, before the turbine, would provide the most practical solution to obtaining the additional power required to be able to drive and test the larger compressors.

This study involves the design of the combustor liner using first principles and the design methods of the NREC series [1,3]. This method is implemented using a combination of Excel and Matlab environments in order to produce a preliminary design to initiate the combustor development, testing and application. The combustor was designed, as far as possible, to allow for the use of some off the shelf components such as tubing and injectors and standard, non-specialized material such as Stainless Steel SS304. Based on the test rig requirements, two different operating regimes were considered: 1) a start-up point with a mass flow rate of 0.32 kg/s and 2) a design point with an air mass flow rate of 1 kg/s. The combustor was required to provide 800K output temperature at the design point and be able to start-up within the same blowdown system.

Before manufacture, the preliminary combustor design should be theoretically/semi-empirically evaluated to determine its likely performance [1]. This was done using the proprietary code Holeflow [4]. Holeflow is a post-design analysis tool that incorporates mostly a variation of the NREC combustor analysis methods as well as some methods from fundamental engineering texts to analyse the likely performance of the combustor design. Holeflow was also used to evaluate the effects of some minor changes that were applied to enable easier manufacture and to ensure that a detrimental effect was not expected.

During the design process, there were some uncertainties uncovered w.r.t the various methods used to predict the outlet temperature. Further, during the initial tests, the fuel flow rate values devised using the preliminary design process seem to potentially be too high which is related to the uncertainties in the outlet temperature prediction methods. For these reasons, experimental data was required to help validate the methods used during the design process. Experimental measurements were performed in the blowdown compressor test rig to compare the predicted design performance and the actual performance of the combustor liner.

2.0 DESIGN PROCESS

The requirement of the combustor was to provide an outlet temperature, T_4, of 800K to the turbine at the testing mass flow rate of 1kg/s. Previous experiments performed on the test rig indicated that 0.32kg/s was the minimum controllable mass flow rate of the blowdown rig which was then used to design a start-up condition. For the start-up point, a $\dot{m}_f$ was chosen to get an overall AFR within the range found in the literature [5,6] to be typical for a high altitude relight condition of between 33 and 45. An AFR of 45 was chosen, due to a lower value causing a primary zone equivalence ratio, $\phi_{Pz} > 1.5$ which is unfavourable in terms of smoke creation. The input preliminary design conditions are given in **Table 1**.

In order to choose the fuel mass flow rate for the initial iteration, equations to estimate the Fuel/air Ratio (FAR) from Walsh and Fletcher [2] were used. These equations are shown below in Equation 1 to 4.

$$FAR1 = 0.10118 + 2.00376 \times 10^{-5} \times (700 - T_3) \qquad \dots (\quad 1 \quad)$$

$$FAR2 = 3.7078 \times 10^{-3} - 5.2368 \times 10^{-6} \times (700 - T_3) - T_4 \\ \times 5.2632 \times 10^{-6} \qquad \dots (\quad 2 \quad)$$

$$FAR3 = 8.889 \times 10^{-8} \times |T_4 - 950| \qquad \dots (\quad 3 \quad)$$

$$FAR = \frac{FAR1 - \sqrt{FAR1^2 + FAR2} - FAR3}{\eta_{34}} \qquad \dots (\quad 4 \quad)$$

These equations provide estimates within 0.25% accuracy with kerosene fuel which has an LHV of 43124 kJ/kg [2]. Using this method, the initial estimate for the required $\dot{m}_f = 0.0133$ kg/s to get T$_4$ = 800K with η_{34} = 95% for the design point.

Table 1
The initial input values for the preliminary combustor design

Condition No.:		1	2	
	Condition:	**Start-up**	**Design point**	**Unit**
Compressor outlet data	p$_3$	221627	548506	[Pa]
	P$_3$	222000	550000	[Pa]
	T$_3$	295	300	[K]
	$\dot{m}_3$	0.32	1	[kg/s]
	$\dot{m}_f$	0.0071	0.0133	[kg/s]
Turbine inlet data	T$_4$	-	800	[K]
	pf	20	20	[%]
Combustor	$\frac{P_{34}}{P_3}$	0.053	0.053	-
	$\frac{P_4}{P_3}$	0.947	0.947	-
	η_{34}	0.80	0.95	[%]
	ϕ_{34}	0.3287	0.20	-
	FAR_{34}	0.02222	0.01327	-
	AFR_{34}	45	75	-

The first iteration of preliminary design was analysed using the combustor analysis tool Holeflow [4] which indicated a lower than required outlet temperature of 775K. An increase in fuel mass flow rate was therefore required. A $\dot{m}_f$ value of 0.0155 kg/s was settled on using Holeflow to investigate the required value. It was decided to choose a value of T$_4$ = 840K as the target to allow some leeway due to Holeflow assuming 90% efficiency at the exit to the secondary zone, which might be optimistic. This resulted in a new set of input conditions which are shown in **Table 2**. The adiabatic flame temperature graphs in NREC indicated T$_4$ = 877K at the new chosen $\dot{m}_f$ for the design point and T$_4$ = 965K at the start-up point.

These values were then re-applied into the NREC design process for a second iteration. A new combustor diameter was chosen that the combustor sizing process indicted would allow 80% combustor efficiency during the start-up condition and >95% combustor efficiency for the design point, according to the Odgers-carrier method. This diameter also satisfied the theta parameter at the design point as well as the Brag criterion and the Aerodynamic considerations at both the considered operating conditions.

These efficiency recommendations were obtained from Lefebvre and Ballal [7] who indicated that the minimum efficiency requirement for ignition conditions is 80% to enable spool acceleration while still maintaining a reasonable combustor size. Further they indicate that attempts to aim for efficiency >90% should be applied.

After the design was completed and the wall thicknesses of 0.9 mm chosen, the casing inner diameter was 0.126 m. Upon investigation, the closest off-the shelf pipe that could be used for the casing had an inner diameter of 0.128 m. This variation was analysed in Holeflow which indicated that this variation would pose little to no effect on the performance of the combustor.

Table 2
The revised input values for the preliminary combustor design

Condition No.:		1	2	
	Condition:	Start-up	Design point	Unit
Compressor outlet data	p_3	221627	548506	[Pa]
	P_3	222000	550000	[Pa]
	T_3	295	300	[K]
	$\dot{m}_3$	0.32	1	[kg/s]
	$\dot{m}_f$	0.0071	0.0155	[kg/s]
Turbine inlet data	T_4	965	877	[K]
	pf	20	20	[%]
Combustor	$\dfrac{P_{34}}{P_3}$	0.053	0.053	-
	$\dfrac{P_4}{P_3}$	0.947	0.947	-
	η_{34}	0.80	0.95	[%]
	ϕ_{34}	0.3287	0.2293	-
	FAR_{34}	0.02222	0.0155	-
	AFR_{34}	45	65	-

For the swirler design, the recommended range for the blade stagger angle, α_{sw}, which is assumed to be equal to the turning angle of the airflow, β_{sw}, is between 45 and 70°. A value of 60° was chosen arbitrarily based on past experience in similar combustor designs. According to NREC [1], ±50% of the air from the Pz holes goes to the Rz, 1/3rd of the total Rz air should be admitted through the swirler thus half of the Pz hole Rz air amount enters through the swirler. Since it was designed that the Pz receives 22% of the total air flow, $\dot{m}_3$, thus $\dot{m}_{sw}$ = 5.5% of $\dot{m}_3$. For ease of manufacture thin straight blades were chosen thus K_{sw} = 1.3 [1]. Using equation 5 below from NREC [1], the area of the swirler opening, A_{sw}, was calculated.

$$A_{sw} = \frac{A_{ref}}{\sqrt{\left(\frac{\left(\frac{\Delta P_{sw}}{q_{ref}}\right)}{K_{sw} * \left(\frac{\dot{m}_{sw}}{\dot{m}_3}\right)} + \left(\frac{A_{ref}}{A_{ft}}\right)^2\right) * cos^2(\beta_{sw})}} \quad \dots (\ 5\ )$$

Assuming an injector diameter of 0.015 m, an inner swirler wall of 0.0015 m and 10 blades with a blade front face thickness of 0.001 m, the inner diameter of the outer swirler ring was calculated to be 0.026 m. In NREC it states "For satisfactory performance, both the blade stagger angle α_{sw} and the blade length should be made large enough that one can no longer see through the swirler" [1]. The swirler length was thus calculated to achieve this giving L_{sw} = 0.005 m.

For this swirler design, a Swirl number, Sn, of 2.15 is predicted, which is much larger than the minimum of 0.6. Thus, good recirculation is expected.

The dome shape of the combustor is not specified by NREC thus typical shaping can be selected. A rounded dome follows the "magic circles'" outline better, however, a truncated cone dome was chosen over a rounded dome for ease of manufacture. The cone edges were calculated to be tangential to the "magic circle" zones.

In order to specify an Igniter, a method needed to be followed to specify the expected required energy. Bane et al. [8] indicated that there was definite kerosene ignition between 2 & 10mJ but their experiments were performed in Kerosene-air at 60°C (333.15K). The inlet temperatures from the blowdown rig are in the range of 290K to 300K, thus to compensate for the low temperatures in the SUN compressor test rig, Figure 5.35 in Lefebvre & Ballal [7] was used: the ignition energies required were thus more likely to lie between 10 to 20mJ.

An off the shelf injector was specified that could deliver both the start-up and the design point fuel mass flow rates and that had a solid cone spray and not just a hollow cone pattern. There was not a solid cone injector that could supply the full mass flow rate ranges thus a semi-solid spray cone was specified as adequate.

3.0   DESIGN RESULTS

The resultant combustor design is a can type combustor shown in Figure 1 and Figure 2. The dimension values for the combustor design are given in **Table 3** and **Table 4**.
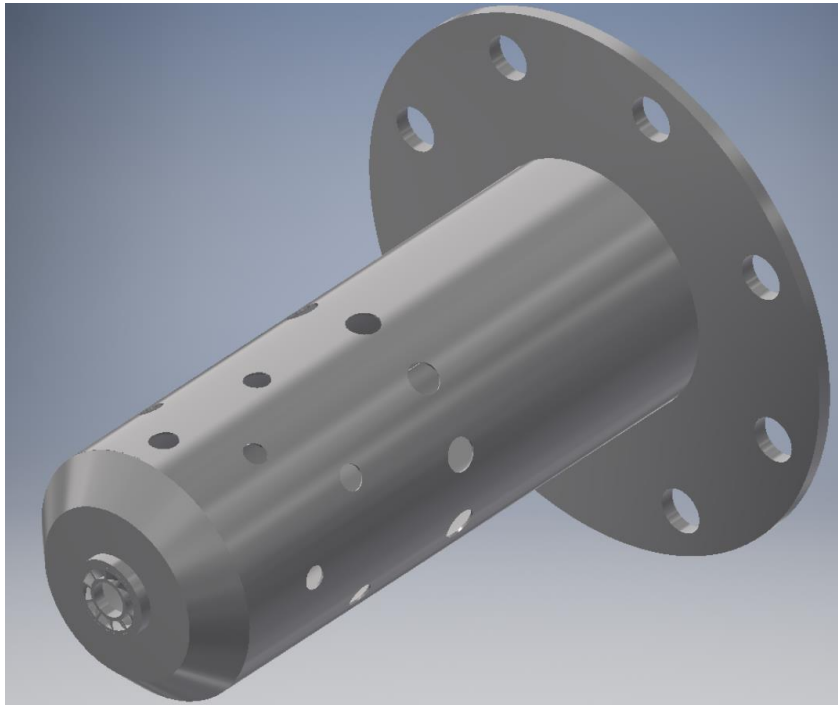

Figure 1 A CAD model of the designed combustor liner

Table 3
The designed combustor dimensions

Parameter	Value before wall thickness considered	Value after wall thickness considered	Unit
A_{ft} =	0.0085		[m^2]
d_{ft} =	0.104	0.104	[m]
A_{ref} =	0.0121	0.0125	[m^2]
D_{ref} =	0.124	0.126	[m]
Pz air =	22.00		[%]

Table 4
The designed combustor zone lengths and air mass flow splits

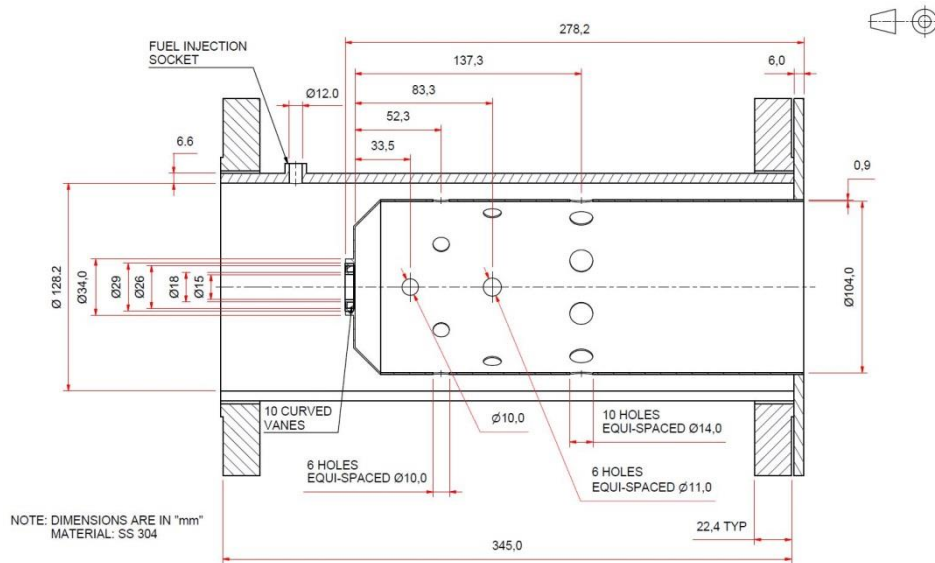| Zone	% air [%]	L [m]
Recirculation Zone	-	0.052
Primary Zone	22	0.078
Secondary Zone	19	0.052
Dilution Zone	59	0.142
Cooling Air	0	-
Totals	**100**	**0.272**

Figure 2 The details of the designed combustor for manufacture

Figure 3 shows the graphs for the gas temperature along the length of the combustor, calculated using three different methods: Holeflow [4], NREC equations and NREC AFT graphs [1]. It can be seen on this figure that the predicted outlet temp has some variation, specifically T_4 = 840K, 588K and 877K respectively. These values will be measured in the experimental test rig for validation.
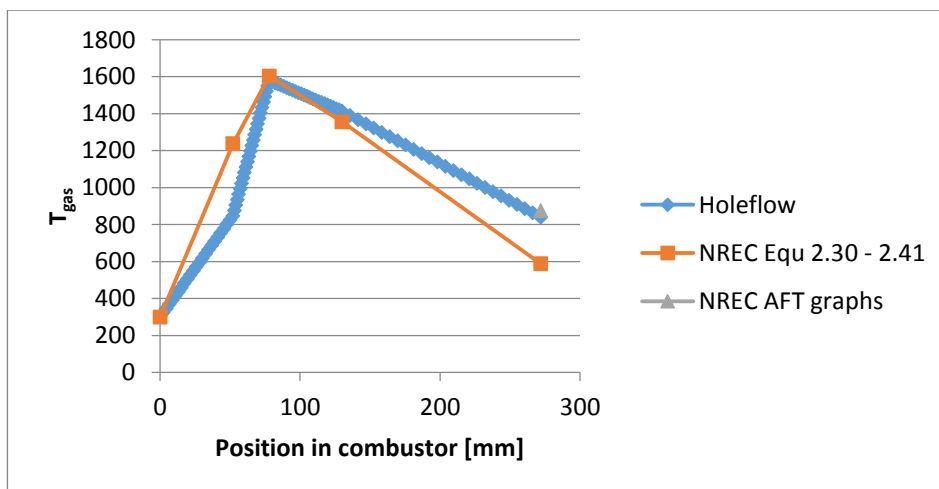

Figure 3 The predicted gas temperature, T_{gas}, along the length of the combustor using three tools: Holeflow [4], NREC equations and NREC AFT graphs [1]

The Holeflow analysis results on the final design for the design point and the start-up point are shown in **Table 9** and **Table 10** respectively. The results indicate that at the design point conditions, the theta parameter method predicts a 95% efficiency. A pattern factor of 0.1 is predicted. For the start-up condition, Holeflow indicates a theta parameter efficiency of 93% and a pattern factor of 0.1.

The pressure drop over the combustor is slightly above the recommended value for a can type combustor at $\frac{\Delta P_{34}}{P_3} = 5.7\%$ instead of 5.3 at design point.

4.0   EXPERIMENTAL SETUP

The compressor test rig, in which this combustor is intended to be used, is shown in Figure 4 and the manufactured combustor is shown in Figure 5. The test rig runs at or just below atmospheric temperature, with a 10 bar blowdown capability. The operating conditions provided from previous tests are given in **Table 5**.
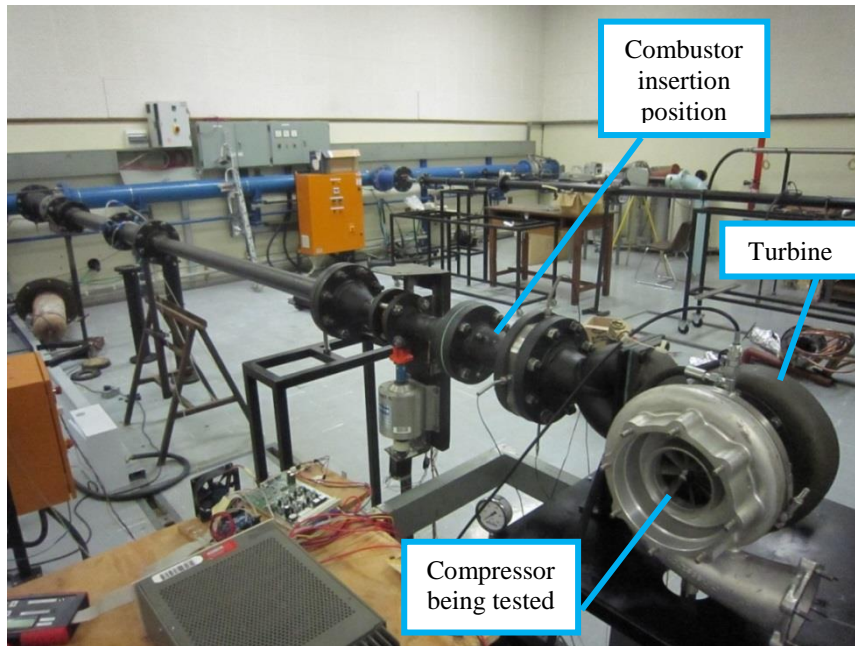

Figure 4 The pipes, turbine and compressor section of the blowdown compressor test rig at Stellenbosch University

Figure 5 The manufactured combustor ready for initial testing

Table 5
Blowdown compressor test rig operating points

Speed	Turbine inlet pressure (g)	Mass flow
[rpm]	[kPa]	[kg/s]
40 000	122	0.32
60 000	149	0.49
70 000	185	0.65
80 000	250	0.76
101 000	554	1.26

The combustor design was manufactured and placed in a small fan driven combustor test rig for initial testing. After the completion of these tests, the combustor was moved to the blowdown compressor test rig for further testing.

Before the combustor could be placed in the compressor test rig with the other components, tests were performed on the combustor alone in the blowdown facility. The purpose of this was both to test the combustor functionality before inserting it into the test rig with other components in the future and secondly to enable greater accessibility to the combustor in order to perform combustor outlet temperature measurements. The disadvantage to this setup is that, due to the missing turbine, the higher inlet pressures could not be reach.

In order to compare the design method temperature calculations to the experimental outlet temperature, an experimental set point that the test rig was capable of reaching in this open configuration was then chosen to use as an experimental comparison point. The experimental input values were then measured and fed into the design evaluation equations in order to compare the experimental and calculated values at a single set point. The chosen set point was the point at which the highest mass flow rate could be attained while maintaining stable combustion with the flame inside the combustor can. The properties at this operating condition are shown in **Table 6**.

Table 6
The values obtained experimentally for the experimental comparison set point

Condition No.:		3	
	Condition:	Exp. Comp. pt.	Unit
	p_3	126147	[Pa]
	P_3	130428	[Pa]
Combustor inlet data	T_3	285	[K]
	$\dot{m}_3$	0.85	[kg/s]
	$\dot{m}_f$	0.0071	[kg/s]

The chosen set point was obtained at a fuel mass flow rate equivalent to the start-up condition; however, the air mass flow rate was able to be increased up to 0.85kg/s which is almost at the design point air mass flow rate.

At this chosen set point, the temperatures at the outlet plane were measured using a thermocouple rake during the running of the combustor. The thermocouple rake, which consists of 5 thermocouples placed radially in a single radius of the combustor outlet, was placed at 2 azimuthal positions: 0° and 180°. 3 sets of measurements were taken at these points and the average was used to calculate an average outlet temperature for comparison.

The test rig in operation while collecting temperature data is shown in Figure 6

Figure 6 The test rig burning during a test

5.0   EXPERIMENTAL RESULTS

The experimental measurements were used to compare to the predicted design performance and the actual performance of the combustor liner. The measurement that was used for this comparison was the exit temperature. Other parameters being measured were the inlet temperature, air mass flow rate and fuel mass flow rate.

Both the temperature and efficiency parameter values calculated in the design phase as well as the temperature from the experimental measurements are shown in **Table 7**. The uncertainty of the temperature measurements is of the order of ±7.5K.

Table 7
The calculated temperature and efficiency parameters and measured temperatures

	Condition No.:	1	2	3	
	Condition:	Start-up	Design point	Exp. Comp. pt.	
Property	Method				Unit
Av. T_4	Holeflow	1052	840	608	[K]
	NREC Equations	864	588	337	[K]
	NREC AFT Graphs	965	877	552	[K]
	Experimental	-	-	485	[K]
η	Odgers-Carrier	80	97	53	[%]
	θ-parameter	100	92-96	40-55	[%]
	Holeflow	95	94	41	[%]
	NREC Equations	68	48	16	[%]
	Experimental	-	-	60	[%]

It was found that the NREC AFT Graphs temperature predictions were the closest to the experimental data; however the predicted temperature was still 15% higher than the measured value. The reason for this is likely due to the efficiency of the actual combustor at this set-point being low and thus a lower temperature was achieved. This was tested by using the various methods of predicting the efficiency to determine the predicted temperature using the AFT Graphs. These results can be seen in **Table 8**. The predicted temperature in the experimental column it the temperature predicted using the AFT graphs and the efficiency calculated using the experimentally measured values. It was found that the temperature was best predicted using the Odgers-carrier method for predicting the efficiency. The θ-parameter method predicts a range for the efficiency rather than a single value. If the highest predicted efficiency was used, the predicted temperature was close to the measured value; however there is no guideline as to which efficiency value in the range should be chosen.

Table 8
A comparison of the predicted temperature using the various efficiency prediction methods and the AFT graphs to the calculated and measured values from the experiment

η pred. method:	Assumed during design	Odgers-Carrier	θ-parameter			Hole flow		Exp	Unit
Pred. η:	0.8	0.53	0.40	0.45	0.55	0.41	**Calc. η:**	0.6	[%]
AFT ΔT:	334	334	334	334	334	334	**AFT ΔT:**	334	[K]
Pred. T_4:	552	460	419	435	469	422	**Pred. T_4:**	484	[K]
							Meas. T_4:	485	[K]

6.0 CONCLUSION AND RECOMMENDATIONS

Throughout the design process, the Odgers-Carrier method of predicting the efficiency of the combustor liner has been more conservative than the Theta parameter method. The Odgers-Carrier method has been used as the guiding predictor thus far due to it being similar to the combustor loading factor which is used by other combustor design methods such as Mattingly et al. [**9**].

It was expected that the temperature predictions using Holeflow and the NREC AFT graphs would prove to be a more accurate representative value for the outlet temperature than the NREC equations used to calculate the gas temperature. The temperature result was best predicted using the AFT graphs with the Odgers-Carrier method to calculate the efficiency.

It is recommended in the future to also compare the efficiency values to those predicted using the various models to those measured experimentally using emission measurements. Further, these experiments should be repeated at the actual design points in order compare the results at the required operating conditions of the test rig.

ACKNOWLEDGMENTS

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APPENDIX

Table 9
The analysis results of the combustor for the design point using Holeflow [4]

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Combustion Chamber Flow Distribution/Heat Transfer Simulation
INPUT DATA SUMMARY

Case name / description .......... : CRCD17.HFI

Combustor type .......... (CCtype) :  CAN
Total number of hole sets ........ =     4 [-]

Set  Type         Cd    Number  Diameter  Xpos    LOD      COD
                  [-]    [-]     [mm]      [mm]    [mm]     [mm]
 1   Swirler      0.56    1      17.49      0.0    73.00    126.00
 2   Plain        Calc    6      10.00     52.0   104.00    126.00
 3   Plain        Calc    6      11.00     83.0   104.00    126.00
 4   Plain        Calc   10      14.00    137.0   104.00    126.00

Total air mass flow rate ... (mdot) =    1.000 [kg/s]
Overall air/fuel ratio ...... (AFR) =   65.00 [-]
Stagnation pressure   ........ (P0) =  550.00 [kPa]
Stagnation temperature ....... (T0) =  300.00 [K]
Stagnation temperature ....... (T0) =   26.85 [øC]
Specific heat of gas ......... (Cp) = 1006.00 [J/kgK]
Specific heat ratio ....... (gamma) =    1.40 [-]
Fuel atomic C/H ratio ........ (CH) =   10.00 [-]
Fuel lower calorific value .. (LCV) =   42.00 [MJ/kg]
Heat transfer/liner cooling model . =  UNCOOLED LINER
Flame radiation model ............. =  SIMPLIFIED
Ambient air temperature  (Tambient) =   25.00 [øC]
Liner inner emissivity ...... (eLi) =    0.40 [-]
Liner outer emissivity ...... (eLo) =    0.80 [-]
Casing inner emissivity ..... (eCi) =    0.80 [-]
Casing outer emissivity ..... (eCo) =    0.80 [-]
Liner thermal conductivity ... (kL) =   15.00 [W/mK]
Casing thermal conductivity .. (kC) =   15.00 [W/mK]
Liner wall thickness ......... (tL) =    0.90 [mm]
Casing wall thickness ........ (tC) =    3.00 [mm]




Combustion Chamber Flow Distribution/Heat Transfer Simulation
CALCULATION RESULTS

Pressure drop across combustion chamber .......... =    31.0 [kPa]
Percentage pressure drop ......................... =     5.6 [%]

 # Set#  Type     #holes Diameter   Cd     Ag      Ag%     va       ma
ma%
                         [mm]      [-]    [mm^2]   [%]    [m/s]    [kg/s]
[%]
 1   1  Swirler      1    17.49    0.56   240.25   8.52   54.22   0.08307
8.31
 2   2  Plain        6    10.00    0.57   471.24  16.70   52.01   0.15567
15.57
 3   3  Plain        6    11.00    0.58   570.20  20.21   54.94   0.19930
19.93
 4   4  Plain       10    14.00    0.60  1539.38  54.57   57.28   0.56196
56.20

Set  Type         angle   YRZ     YRP     Ysec    Ydil
                   [ø]     [-]     [-]     [-]     [-]
 1   Swirler        -      1.00    0.00    0.00    0.00
 2   Plain        68.80    0.50    0.50    0.00    0.00
 3   Plain        73.85    0.00    0.00    1.00    0.00
 4   Plain        83.21    0.00    0.00    0.00    1.00

Mass flow rate in primary recirculating zone ...... =   0.1609 [kg/s]
Mass flow rate in remainder of primary zone ....... =   0.0778 [kg/s]
Mass flow rate in secondary zone .................. =   0.1993 [kg/s]
Mass flow rate in dilution zone ................... =   0.5620 [kg/s]

Primary zone (recirc.) equivalence ratio .. (phiRZ) =    1.41 [-]
Primary zone (remainder) equivalence ratio (phiRP) =    0.95 [-]
Secondary equivalence ratio ............... (phiSZ) =    0.52 [-]
Dilution zone equivalence ratio ........... (phiDZ) =    0.23 [-]

Theta parameter .......................... (theta) =   76.53 x 10^6 [-]
Combustion efficiency, NREC Figure 2.2 .. (CombEta) =   95.11 [%]
Emmision index : CO .................... (EI_CO) =  183.26 [g CO/kg
fuel]
Emmision index : HC .................... (EI_HC) =   82.37 [g HC/kg
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fuel]
Emmision index : H2 ...................... (EI_H2) =     5.04 [g H2/kg
fuel]
Combustion efficiency ................. (CombEta2) =    85.94 [%]

Pattern factor .............................. (PF) =     0.10 [-]

Flametube flow area ........................ (Aft) =     0.01 [m^2]
(Flametube flow area)/(Reference area) . (Aft/Aref) =    58.35 [%]
Reference area ............................ (Aref) =     0.01 [m^2]
Reference diameter ........................ (Dref) =     0.13 [m]
Reference density ...................... (rhoaREF) =     6.37 [kg/m3]
Reference velocity ........................ (uref) =    12.55 [m/s]
Reference velocity head ................... (qref) =   502.27 [Pa]
Pressure difference 3 to 4 ................ (dp34) = 31032.62 [Pa]

                    dP3-4/p3 [%]    dP3-4/qref    mT3^0.5/(Aref*p3)
    CAN (typical)       5.3             40              3.0E-3
    CAN (actual)        5.7             62              2.5E-3
```

Table 10
The analysis results of the combustor for the Start-up point using Holeflow [4]

```
Combustion Chamber Flow Distribution/Heat Transfer Simulation
INPUT DATA SUMMARY

Case name / description ........... : CRCD18.HFI

Combustor type .......... (CCtype) :  CAN
Total number of hole sets ........ =    4 [-]

Set   Type      Cd    Number  Diameter  Xpos    LOD     COD
               [-]     [-]     [mm]     [mm]    [mm]    [mm]
 1    Swirler   0.56    1      17.49     0.0    73.00   126.00
 2    Plain     Calc    6      10.00    52.0   104.00   126.00
 3    Plain     Calc    6      11.00    83.0   104.00   126.00
 4    Plain     Calc   10      14.00   137.0   104.00   126.00

Total air mass flow rate ... (mdot) =     0.320 [kg/s]
Overall air/fuel ratio ...... (AFR) =    45.00 [-]
Stagnation pressure  ........ (P0) =   222.00 [kPa]
Stagnation temperature ....... (T0) =   300.00 [K]
Stagnation temperature ....... (T0) =    26.85 [øC]
Specific heat of gas ........ (Cp) =  1006.00 [J/kgK]
Specific heat ratio ....... (gamma) =     1.40 [-]
Fuel atomic C/H ratio ........ (CH) =    10.00 [-]
Fuel lower calorific value .. (LCV) =    42.00 [MJ/kg]
Heat transfer/liner cooling model . = UNCOOLED LINER
Flame radiation model ............. = SIMPLIFIED
Ambient air temperature  (Tambient) =    25.00 [øC]
Liner inner emissivity ...... (eLi) =     0.40 [-]
Liner outer emissivity ...... (eLo) =     0.80 [-]
Casing inner emissivity ..... (eCi) =     0.80 [-]
Casing outer emissivity ..... (eCo) =     0.80 [-]
Liner thermal conductivity ... (kL) =    15.00 [W/mK]
Casing thermal conductivity .. (kC) =    15.00 [W/mK]
Liner wall thickness ......... (tL) =     0.90 [mm]
Casing wall thickness ........ (tC) =     3.00 [mm]




Combustion Chamber Flow Distribution/Heat Transfer Simulation
CALCULATION RESULTS

Pressure drop across combustion chamber ........... =     7.9 [kPa]
Percentage pressure drop ......................... =     3.5 [%]

 # Set#  Type    #holes Diameter  Cd      Ag      Ag%     va      ma
ma%
                        [mm]     [-]    [mm^2]    [%]    [m/s]   [kg/s]
[%]
 1   1  Swirler    1    17.49    0.56   240.25    8.52   42.95   0.02657
8.30
 2   2  Plain      6    10.00    0.57   471.24   16.70   41.17   0.04984
15.58
 3   3  Plain      6    11.00    0.58   570.20   20.21   43.50   0.06379
19.93
 4   4  Plain     10    14.00    0.60  1539.38   54.57   45.37   0.17979
56.19

Set   Type      angle    YRZ    YRP    Ysec    Ydil
               [ø]      [-]    [-]    [-]    [-]
 1  Swirler      -      1.00   0.00   0.00   0.00
```

```
2   Plain            68.81    0.50    0.50    0.00    0.00
3   Plain            73.85    0.00    0.00    1.00    0.00
4   Plain            83.21    0.00    0.00    0.00    1.00

Mass flow rate in primary recirculating zone ...... =   0.0515 [kg/s]
Mass flow rate in remainder of primary zone ....... =   0.0249 [kg/s]
Mass flow rate in secondary zone .................. =   0.0638 [kg/s]
Mass flow rate in dilution zone ................... =   0.1798 [kg/s]

Primary zone (recirc.) equivalence ratio .. (phiRZ) =    2.03 [-]
Primary zone (remainder) equivalence ratio  (phiRP) =    1.37 [-]
Secondary equivalence ratio ............... (phiSZ) =    0.75 [-]
Dilution zone equivalence ratio ........... (phiDZ) =    0.33 [-]

Theta parameter ........................... (theta) =   62.15 x 10^6 [-]
Combustion efficiency, NREC Figure 2.2 .. (CombEta) =   93.85 [%]
Emmision index : CO ....................... (EI_CO) =  183.26 [g CO/kg
fuel]
Emmision index : HC ....................... (EI_HC) =   82.37 [g HC/kg
fuel]
Emmision index : H2 ....................... (EI_H2) =    5.04 [g H2/kg
fuel]
Combustion efficiency .................. (CombEta2) =   85.94 [%]

Pattern factor ............................... (PF) =    0.10 [-]

Flametube flow area ......................... (Aft) =    0.01 [m^2]
(Flametube flow area)/(Reference area) . (Aft/Aref) =   58.35 [%]
Reference area ............................. (Aref) =    0.01 [m^2]
Reference diameter ......................... (Dref) =    0.13 [m]
Reference density ....................... (rhoaREF) =    2.57 [kg/m3]
Reference velocity ......................... (uref) =    9.95 [m/s]
Reference velocity head ................... (qref) =  127.52 [Pa]
Pressure difference 3 to 4 ................ (dp34) = 7863.81 [Pa]

                     dP3-4/p3 [%]      dP3-4/qref     mT3^0.5/(Aref*p3)
     CAN (typical)       5.3               40            3.0E-3
     CAN (actual)        3.5               62            2.0E-3
```