Single Cylinder Four Stroke Four Valve Engine Optimization

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ABSTRACT

Our team chose to design our 250 cc single cylinder four-stroke motorcycle engine using performance data from a simulated race track comparable to the Austrian Grand Prix. This track program runs an engine and motorcycle configuration through a forward-looking driving cycle that applies throttle or braking based on upcoming corners. Performance of different engine designs were simulated in GT-POWER, and the models were integrated through the use of modeFRONTIER. Optimizations on various intake, exhaust, and cam design parameters were carried out to identify high performing engine configurations for the simulated race track. A linear multi-criteria decision-making model was used to prescribe three cases with different lap time versus fuel consumption weightings. The case that balanced 75% lap time and 25% fuel consumption was adjusted for physical packaging and manufacturing constraints. Specifications and engine performance curves for this configuration are provided. Our approach incorporates realistic, vehicle dynamics in the design of race engines. These effects are not apparent in the analysis of a fixed set of steady-state operating points.

INTRODUCTION

When analyzing data from engine simulations it is difficult to determine which configurations are optimal without knowing the driving cycle the vehicle will undergo. This is only made more difficult by the fact that the engine performance will itself determine how a vehicle will be operated. We chose to address this issue with an approach that was designed for use in the 2014 Formula Hybrid design competition, in which the University of Idaho's team won 1st place overall. [1] When developing the vehicle for this competition it was realized that no commercially available product met the vehicle simulation needs of the team. As such, a highly configurable track simulation model was developed in TK Solver. It is this models versatility that allowed us to modify it for use in motorcycle applications. Though this software model was developed in TK Solver we were able to integrate it with modeFRONTIER through an Excel translator. This track program, when given the inputs from a 1D model of our engine configuration made in GT-Power, was sufficient information to decide the optimality of an engine design. The integration of these solvers, as well as the inputs and output of interest in our design process are shown in Figure 1.

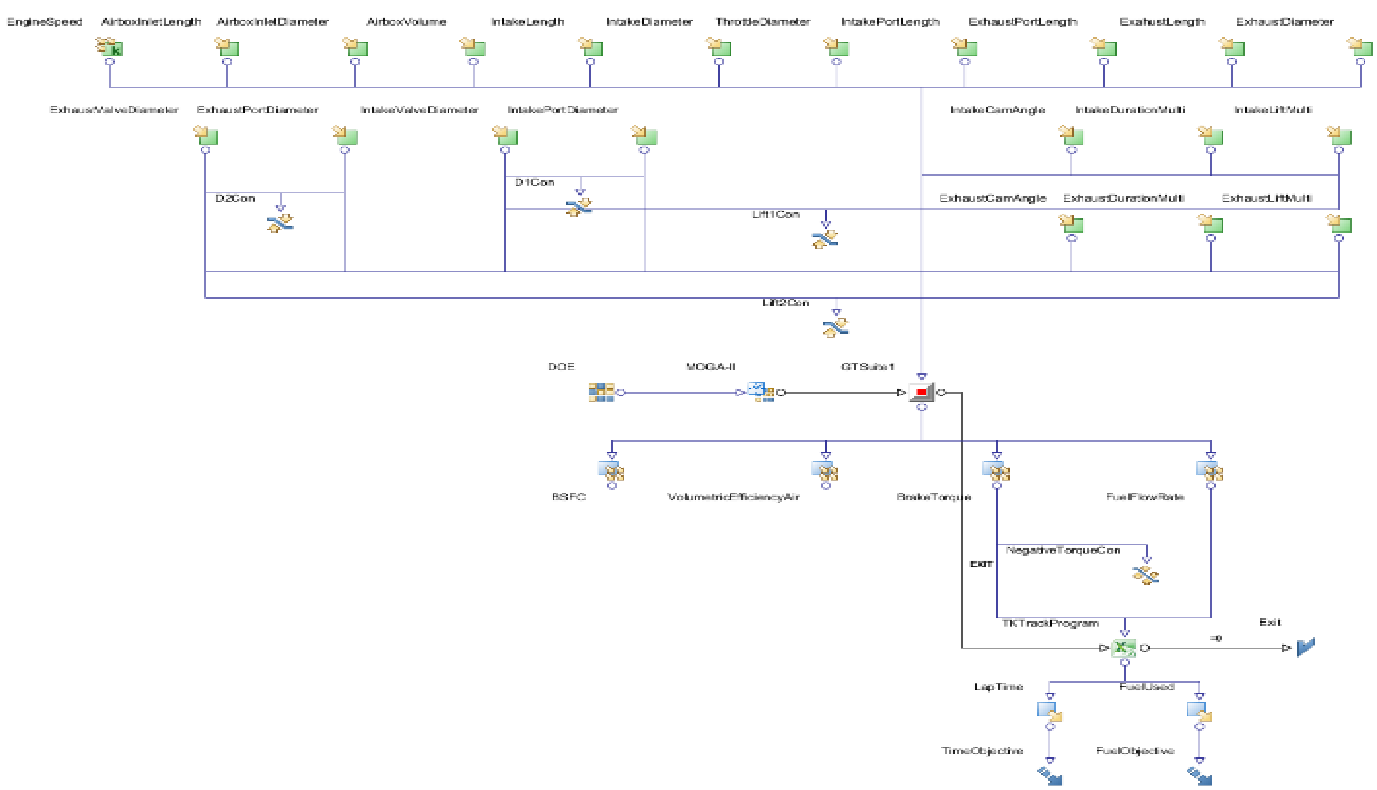

Figure 1. Workflow diagram and integration of engineering solvers in modeFRONTIER.

Our methodology simplifies the design process, and allows us to consider thousands of designs with minimal effort. Once a sufficient number of designs has been evaluated a suitable one can be easily chosen, and any trends in the input variables analyzed for future exploration.

METHODOLGY

TRACK SIMULATION

MODIFICATION OF THE HYBRID ENDURANCE TRACK MODEL

The simulated test track used in this optimization was created with modifications from the original Hybrid Endurance program. [2] The first part of the modifications was to lengthen the track to match our chosen track. The original program has a limit to the on the size of the lists that plot the track layout. This was originally planned to give TK Solver an exit in the loop if there was an error in the length inputs or values of torque and rpm were transferred incorrectly. The solution to this problem was to lengthen the amount of iterations that the program can perform before it should forcefully exit the loop.

The second modification made to the original code was the removal of the hybrid components used in its original form. The solution to this was to first remove the hybrid components used in the torque calculations and the vehicles velocity profile. Then the heavy side step function was modified to only use positive values of torque.

The third modification made to the original track program was the addition of fuel consumption. This step gave the ability to predict the amount of fuel consumed during one lap of the simulation. This was accomplished by adding an interpolation table that was fed fuel consumption rates from GT-Power for the different RPM values determined. Already programed in the code is a look up function for the values of torque at different RPM ranges. The fuel consumption was then tagged along in each location where torque was interpolated. This allowed for simplification of code changes and fuel consumption to be calculated at every tenth of a second.

The fourth modification made was the velocity at which the vehicle entered the radius of the corners. The previous velocity was set at a lower value, keeping the bike at a lower velocity as it travels through the corner. To figure out the maximum velocity that the bike was allowed to travel through the corner we choose to use the following equations.

$$G_{force} = V_{r,ent}/32.2 \tag{1}$$

Equation one shows the “G” force being calculated as a function of gravity. We choose to use a G force of 1.7 this left the velocity of the bike entering the radius of the corner at 54.74 feet/second.

$$V_{R,ent} = R * w \tag{2}$$

Equations two shows the Velocity entering the corner radius set to equal the radius of the comer times the constant omega equaling 1.01. Solving this relationship will set the radius of the corner on the track in feet. The above equation relations define the corner radius needed to keep the bike at 1.7 G’s as it travels through the corner.

$$V_{R,ent} = R * w^2 \tag{3}$$

Equation three shows the relationship used to calculate the lateral acceleration of the bike around the corner in Feet per second squared.

The fifth modification made to the original track program were the input variables. These included weight, max velocity, tire diameter, final drive ratio, max crank rpm, counter shaft rpm, Corner radius, velocity at corner radius, primary gear reduction, 1^{st}-5^{th} gear ratio, and the interpolation tables for torque, rpm, and fuel consumption.

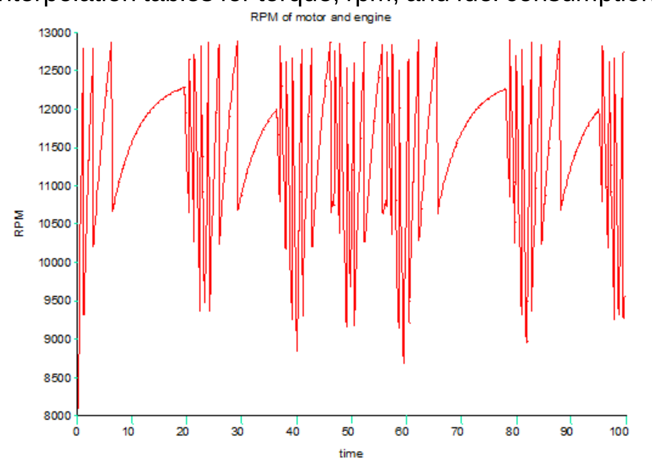

Figure 2: This figure shows the RPM ranges of the engine in one lap.

The last modification made to the original program was the addition of drag in the acceleration calculations. The original program was not concerned with the drag because the slow velocity of the Formula Hybrid car. Without drag the sport bike was accelerating to maximum velocity freely without air resistance. Seen in Figure 2 is the engine RPM during one lap. Looking at this graph the shift ranges can be seen. After drag was added the shift ranges lessen as the selected gear increases, this is a strong indication that the gearing ratios are correct and there is an appropriate load on this engine configuration.

VALIDATION OF THE MODEL AGAINST COLLECTED DATA

To validate the method that we choose to use for the optimization we considered a few different variables. All values collected in Table 1 are based on the Austrian Grand Prix and were taken from Joan MIR, Leopard Racing’s fastest lap in 2016, placing him in 1st place for that race.

Table 1: Side by side comparison of the Austrian Grand Prix and the Simulated Test Track.

	Austrian Grand Prix	**Simulated Test Track**
Length	4.31 km	4.3 km
Max Velocity	219 km/h	210 km/h
# of Major Turns	6	6
# of Straights	6	6
Longest Straight	626 meters	1066 meters
Lap Time	96.228 Seconds	102 Seconds

Looking at Table 1 the results of the Simulated Test Track and the Austrian Grand Prix can be seen. This table provides a side by side comparison of the different measurable values of a track and the racers performance. The most important difference shown in the table is the final lap time for each race. This is still a valid comparison because there are limitations presented when using TK Solver as the equation solver. One of those limitations include each turn radius is identical whereas on the Austrian Grand Prix they vary. This limits the test track bike to slowing down in each corner to a speed of 54 feet/sec, whereas the Austrian Grand Prix there are corners where close to maximum speed can be held.

Figure 3 shows the Simulated Test Track. In this figure the track is shown that the track is largely linear. This is a hindrance to the overall speed and realistic paths that can be taken by a rider to lower lap times.

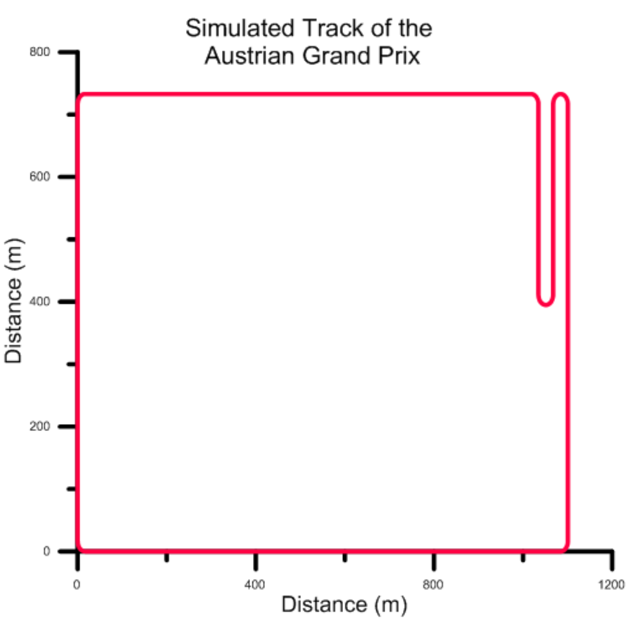

Figure 3. The FinalFormula Test Track with a total length of 4.31 km and an overall speed of 210 km/h.

The first measured value in the comparison is the length of the two tracks. The overall lengths vary by 0.01 km which can be neglected in this case. This difference is due the linear nature of the Simulated Test Track.

The second measured value is the maximum velocity. The max velocity for the Austrian Grand Prix was taken from the fastest lap time in the 2016 Moto3 season. Whereas the Simulated Test Track max speed is set by the user as an input parameter. The input parameter must be input into the program as a value of miles per hour. It will then be changed into feet per second. In the final simulation it was selected at 210 km/h, this is 9 km/h less than that of the fastest lap. However, the test track bike does not reach a speed faster than 210 km/h during its traveled path. This is due to the assumed drag coefficient applied to the model.

The third and fourth measured value is number of turns and straight always in each track. For the Austrian Grand Prix the turns were only considered if they were a major turn. This was because the test track program is limited in a linear fashion and is not capable of having bends and slight turns. The track program only has curves of 90 degrees. If only considering the major straights and turns the two tracks are equal. However, the Austrian Grand Prix has one smaller straight in between turns six and seven, this is not shown in the test track due to the methods used to map the hairpin turn. The constraints set by the code will not allow for the extra straight section to be added.

The fifth measured value is the longest straight section. There is a difference in the lengths because between turn one and three on the Austrian Grand Prix there is a small bend considered as a turn. The test track models that section of track as a true straight. This is the difference in the overall longest straight section of the two tracks.

The sixth measured value is the overall lap time of each course. The fastest lap in 2016 Moto3 at the Austrian Grand Prix was by Joan MIR at 96.288 seconds. The fastest lap conducted in the simulation was 102 seconds.

After the side by side comparison of the two different tracks, and taking in the considerations of the limitations of TK Solver we believe that the results are valid and an accurate representation.

GT-MODEL

OVERVIEW

The GT-POWER model used to simulate the performance of our engine configurations was made as simple as possible in order to reduce runtime, and to ensure that we had an understanding of how the various components interact with each other. We initially wanted to see which variables were important, and which could be left at fixed values for all engine configurations. Based on previous research [3] we fixed the parameters in the SIWiebe combustion model to be as shown in Table 2, and fixed the cylinder wall temperatures shown in Table 3 to values suggested in the GT-SUITE manual.

Table 2. SIWiebe combustion model parameters.

Anchor Angle	8 [degrees]
Duration	25 [degrees]
Wiebe Exponent	2

Table 3. Cylinder wall temperature parameters.

Head Temperature	550 [K]
Piston Temperature	590 [K]
Cylinder Temperature	450 [K]

Engine friction reference model and parameters were copied from GT-POWER example four-stroke engine simulation. We also decided to fix the air-fuel ratio at stoichiometric. Because we are running our engine through a race driving cycle we also fixed the throttle at fully open. We then looked at how to most simply model the air box, as it was the most complicated component in the intake system. At the suggestion of the GT-POWER manual we neglected the resistance of the air filter in our simulation. From subsequent research [4] we found that the air box intake length, air box volume, and intake manifold length were the most important factors, and thing such as tappers and bell mouths are of secondary importance. As such the intake was modeled as a simple collection of pipes and volumes. The full engine model in shown in Figure 4.

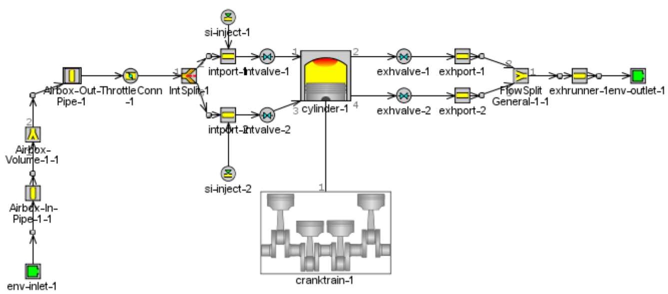

Figure 4: GT-POWER engine model as used for final simulation.

By comparing the results of this engine model to the dyno measurements taken in previous years by the Formula Hybrid team [1] we were able to verify that the output were reasonable for a 250cc motorcycle engine.

GT VALVE

As the intake and exhaust systems were the primary focus of our optimization efforts, and the cam design has a large effect on these systems we elected to use the VT-DESIGN tool to aid in the production of a camshaft design for use in our simulations.

To increase the performance and reduce the BSFC of the engine, various camshaft lifts and durations were tested. The initial optimization varied the critical values needed to produce the camshaft and valve train model in VT DESIGN. The input values selected for optimization were: intake cam angle, intake cam duration, intake valve lift, exhaust cam angle, exhaust cam duration, and exhaust valve lift. Table 4 shows the ranges assigned to each value. These ranges were determined by examining the data in the Megacycle cam catalogue [5], and finding the absolute minimum and maximum values used for any existing camshaft design.

Table 4. Camshaft input values

Variable	Min Value	Max Value
Intake Cam Angle	250 [Crank Degrees]	350 [Crank Degrees]
Intake Cam Duration	240 [Crank Degrees]	350 [Crank Degrees]
Intake Valve Lift	6 [mm]	15 [mm]
Exhaust Cam Angle	80 [Crank Degrees]	160 [Crank Degrees]
Exhaust Cam Duration	300 [Crank Degrees]	350 [Crank Degrees]
Exhaust Valve Lift	6 [mm]	15 [mm]

Using these values, an optimization was run in modeFRONTIER on our GT-SUITE model utilizing a multiplier applied to each variable. We then examined the results in the multi-criteria decision maker, chose a design that was a good trade-off between lap time and fuel usage, and input those values into VT-DESIGN

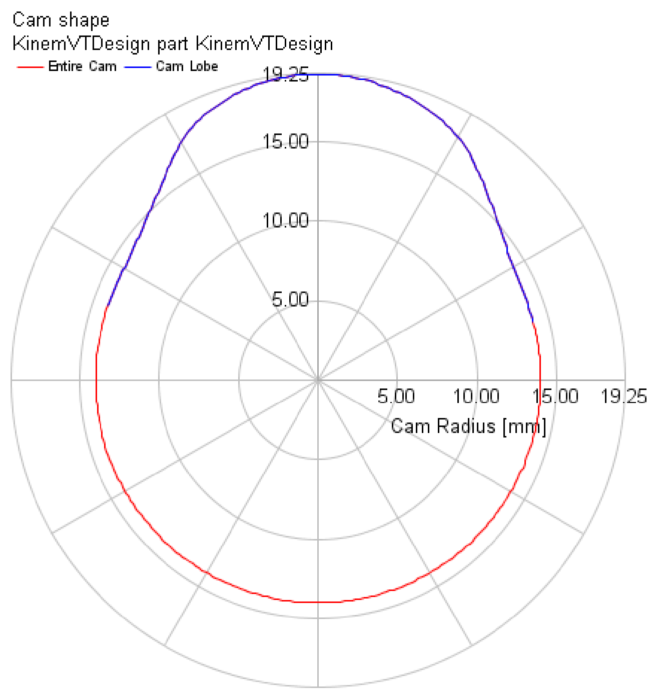

Figure 5: Intake camshaft profile

A polynomial lobe profile was chosen for the VT-DESIGN model due to the need to generate a complete lift profile and the simplicity allowed by a reduced number of user input variables [6,7]. The camshaft base radii were configured to minimize both camshaft mass and concavity of the cam profile. This choice allows manufacturability by reducing the need for small radius stones used to grind the camshaft, as well as reducing contact stress. The cam profile for the intake is shown in Figure 5. The same base radius was chosen for both intake and exhaust profiles to allow interchangeability of followers. Valve train reciprocating mass could be reduced by further optimization of the camshaft and follower parameters.

One of the key design choices made early in the optimization was the use of a mechanically adjusted roller finger follower valve train. This valve train was chosen due to the decreased engine friction and the need to operate at high engine RPM where HLA systems are incapable of operating. The tradeoff for this was the number of variables affecting the lift and valve train dynamics. To reduce design time, it was decided to fix values for many of the variables in the valve train. Further optimizations could be carried out to improve valve train geometry if such an engine were to be produced.

Valve-piston interference was also calculated using the integrated tool in VT-DESIGN. The geometry was created as shown in Table 5. The calculation results are shown in Figure 6. VT-DESIGN generated 2-D model is shown in Figure 7.

Table 5. Valve Piston-Interference calculation input values

Mechanism Geometry	**Value**
Valve Side Arm Length	.04 [m]
Cam Side Arm Length	.02 [m]
Roller or Contact Radius	.01 [m]
Valve Side Contact Radius	.007 [m]
Pivot-Cam Ctr. Distance	.03 [m]
Valve Side Arm Angle	0 [deg]
Valve Side Contact Offset	0 [m]
Valve-Piston angle	-15 [deg]
Valve-Piston offset	-.018 [m]
Piston Top to Pin distance	.025 [m]
Piston Pin Offset	0 [m]
V-P Clearance at TDC	.005 [m]

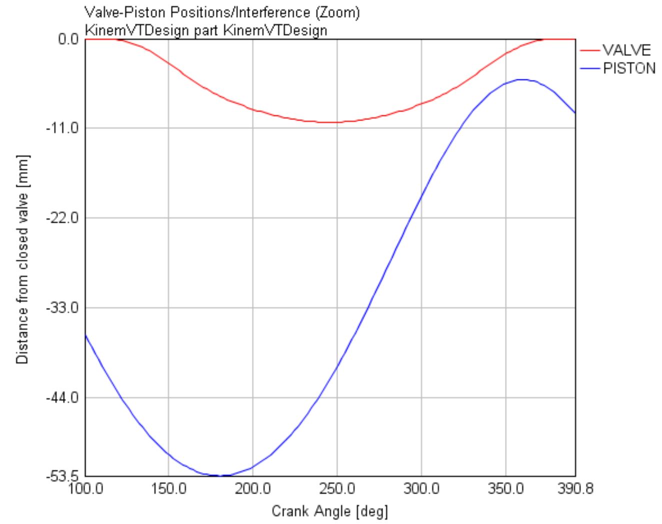

Figure 6. Valve-Piston Interferance

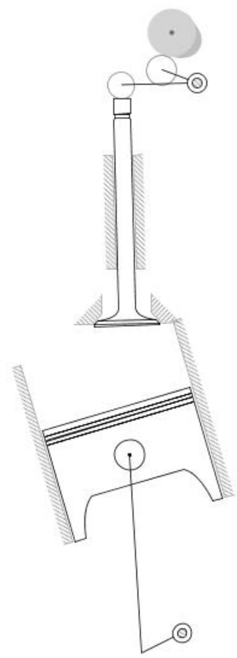

Figure 7. 2-D VT_Design Intake Model

A GTM file was generated and prepared however, due to time constraints was not integrated into the model. Further research could be done to determine friction coefficients for the model and optimize values for the follower and valve train geometry.

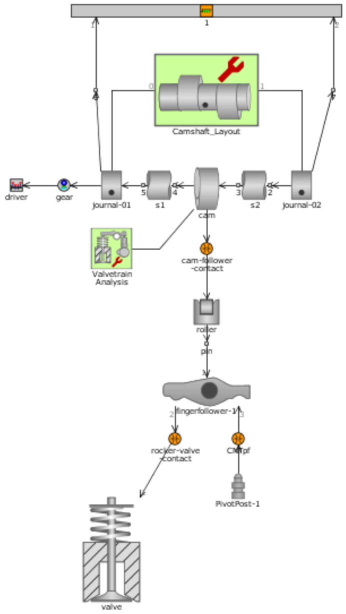

Figure 8. GTM Model

MODEFRONTIER

Due to the limitations of both the track program and GT-POWER several constraints were implemented in modeFRONTIER to maximize the number of feasible designs produced. The primary constraints implemented were on the port diameters, and the valve lifts. These constraints limited our engine model to operate within the range of the existing flow coefficients in the GT-POWER tables. With more experimental data the range of this limit could be increased. A method to measure valve flow coefficients, and obtain this experimental data is detailed in Blair's book [8]. The second constraint we implemented was to exclude designs where the engine output torque dropped below zero. This was done because the track program maintains the same shift points regardless of whether the engine is able to actually reach the desired operating RPM. These designs are technically able to be run through the track simulation; however, they will be very poor options due to the incompatibility between their power curve and the gearing choice.

Due to the large number of variables involved in our problem it is essential that a suitable optimization algorithm be chosen. After looking at the data provided in the modeFRONTIER manual it became clear that this problem was well suited to a genetic algorithm such as MOGA-II so long as we had time to run a sufficient number of trials. After choosing this algorithm we decided that given the time remaining and our computers power we would run an optimization 50 generations long with 100 initial starting points. We used the ISF algorithm to generate our DOE as we wanted to ensure a wide spread of initial designs. Despite our best efforts to constrain input variable only about 2000 of the designs were both real and feasible. However, this proved to be enough for a basic analysis.

TEST TRACK COMBINDATION AND MODEFRONTIER

To use the simulated test track in conjunction with modeFRONTIER we had to use the excel node. Excel must be used in this case because there is not a direct TKSolver node in Mode Frontier. This work flow was then combined with the GT Suite node where a list of 7 rpm, torque, fuel consumption numbers were passed to excel and then onto TKSolver. TKSolver then returned the lap time and fuel consumed in liters to excel. Mode Frontier plotted each case sent to TKSolver vs the amount of fuel consumed and its corresponding lap time. The reasoning for this extra link was to compare our different engine simulations not by pure power curves but with realistic lap times and fuel consumption.

DECISION MAKING

It is a difficult task to pick between the 2000+ designs that we generated. In order to simplify this process we utilized modeFRONTIER's MCDM tool. We chose a linear model, as it is the simplest, and while we do not have knowledge of the proper weighting anyone who would produce such an engine would have some idea of the relative performance they are attempting to achieve. Figure 9 shows this tool, and its use on our data set.

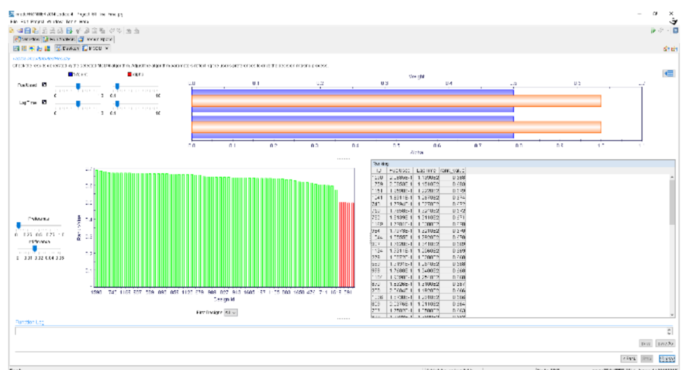

Figure 9. Linear Multi-criteria Decision Making model as applied to our data set. Shown with equal weighting of lap time and fuel usage.

Early in the design process we also used Pearson Correlation Matrix, Figure 10, in order to aid in the process of deciding which variables could be fixed. From this we were able to see which variables had a clear ideal value, and which were dependent on the interaction of the entire system.

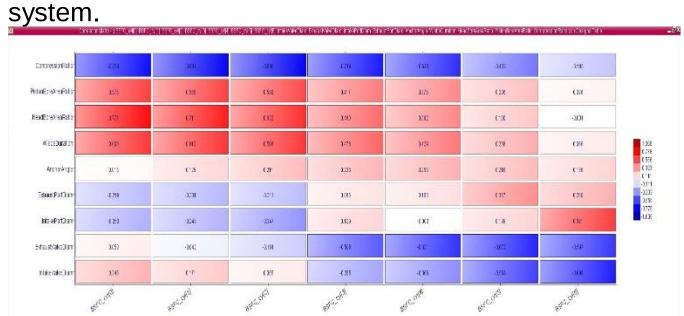

Figure 10. Pearson Correlation Matrix for preliminary study of variable interaction.

RESULTS

The results of our simulation and decision making process are shown below in Figure 11, and the data in Table 6 highlights the characteristics of our three design choices.

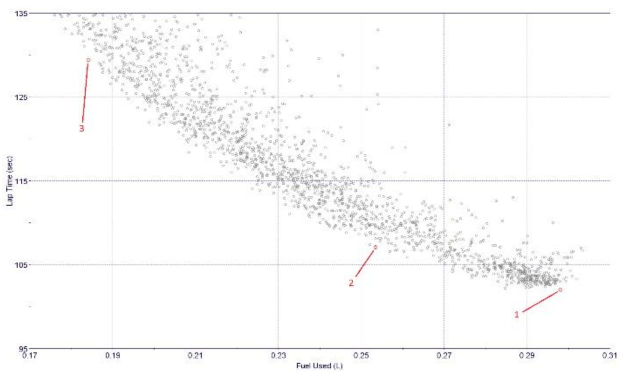

Figure 11. Selection of optimal cases.

Table 6. Basic results of three selected cases

Case Number	Max Torque and RPM	Fuel Consumption [Liters]	Lap Time [sec]
1	23.4 Nm at 11500 RPM	0.298	102.0
2	23.9 Nm at 7500 RPM	0.253	107.1
3	15.8 Nm at 5500 RPM	0.184	129.4

Case 1 is the fastest time we were able to obtain. Case 2 is a balance of 25% fuel consumption 75% lap time, and Case 3 is a 50/50 balance between the design objective. As most real life designs will be a compromise between these factors we chose Case 2 as our final design. The graph shows that it is only a slight compromise in speed for a significant difference in fuel consumption. The performance plots from GT-POWER for this case are shown in Figure 12, and Figure 13.

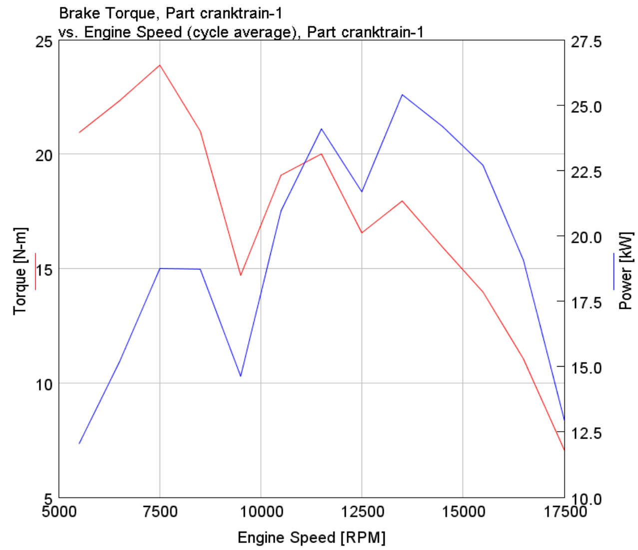

Figure 12. Case 2 torque and horsepower graphs.

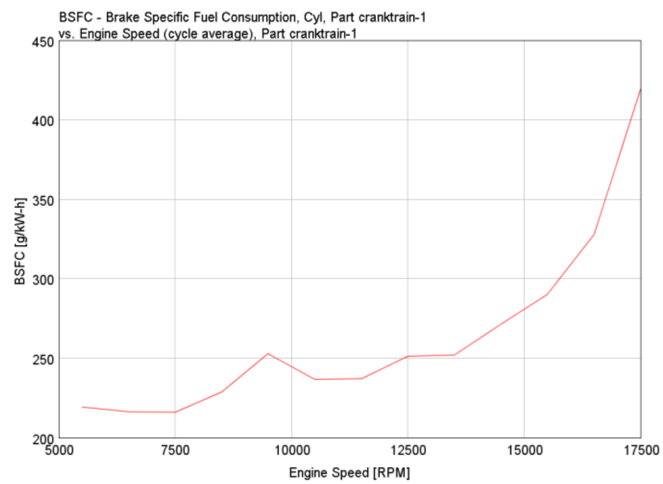

Figure 13. Case 2 BSFC output.

As seen in the plots above this configuration has good output in the 11000-15000 RPM operating range where the majority of the track is run at. However, despite the suitability of this designs torque and fuel usage there are some practical considerations that must be taken into account. Some lengths in this design proved far too long for packaging in a motorcycle design. In order to remedy this the resonant frequencies of the intake and exhaust were doubled based on calculations in Blair's [8]. This resulted in shorter lengths while maintaining similar operating characteristics. Due to the lack of dynamic effects considered in these calculations there was some shift in ramming peaks and troughs, but the operating region remained mostly unchanged. The outputs for this modified configuration are shown in Figure 14 and Figure 15 below, and Table 7 shows the final design parameters chosen.

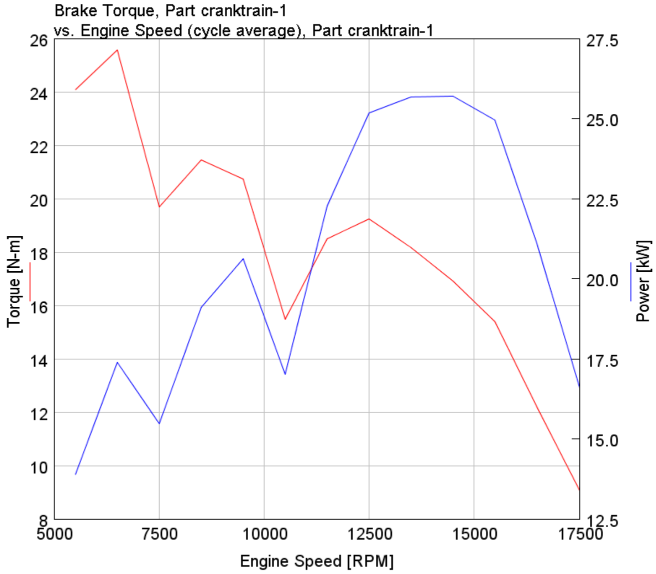

Figure 14. Final configuration torque and horsepower graphs.

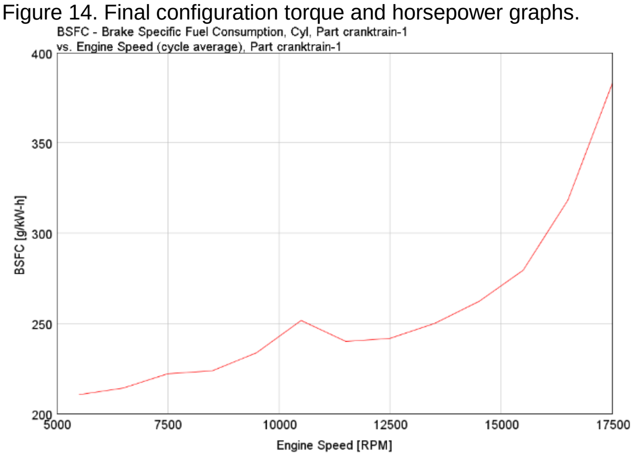

Figure 15. BSFC of final design.

Table 7. Final Design Parameters

Air box inlet diameter	68 [mm]
Air Box Inlet length	469 [mm]
Air box volume	2.66 [liters]
Intake length	428 [mm]
Intake diameter	60 [mm]
Throttle diameter	47 [mm]
Intake port diameter	49 [mm]
Intake port length	90 [mm]
Intake valve diameter	34.1 [mm]
Intake cam duration	257.7 [degrees]
Intake cam lift	7.5 [mm]
Intake cam angle	327.4[degrees]
Compression ratio	15.8:1
Exhaust cam duration	298.0 [degrees]
Exhaust cam lift	8.0 [mm]
Exhaust cam angle	103.1 [degrees]
Exhaust valve	27.4 [mm]
Exhaust port length	105 [mm]
Exhaust port	29.5 [mm]
Exhaust length	698 [mm]
Exhaust diameter	51 [mm]

CONCLUSIONS

Further revisions of this design would be necessary in order to arrive at an engine for production, but the general concept of a track based optimization approach has shown to be effective. Looking at the final engine design and previous parametric studies the only variable that appears to be outside the normal range is the intake port diameter. However, as long as this value was above a certain threshold the simulation results were mostly unchanged. Because of this it would be best to run another optimization to determine the minimum acceptable size. Other features that could be analyzed in the future would include variable length intake, or a variable volume air box. This approach would be very useful tor not only determine what parameter values to use, but also how to control these type of systems. Finally, with experimental data things like air filter restriction, and the effects of bends and similar restrictions could be added to the model in order to improve its accuracy.

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