Nascar Restrictor Plate Exhaust Manifold Design Strategies

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by

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ABSTRACT

This paper presents the results of a study on exhaust manifold design for a NASCAR Restrictor plate internal combustion engine. A computer simulation model was developed using Ricardo WAVE software. WAVE is a computer-aided engineering code developed by Ricardo to analyze the dynamics of pressure waves, mass flows and energy losses in ducts, plenums and the intake and exhaust manifolds of various systems and machines. [1] The model was validated against experimental data from a current NASCAR Winston Cup restrictor plate motor. The parameters studied have been exhaust manifold diameters and lengths. A response surface analysis of the simulation output followed.

The analysis of results shows the design parameters of the existing exhaust manifold are not optimized. The findings from these studies are used to derive exhaust system design guidelines which define optimum exhaust system geometry to maximize average Brake Horsepower over a given powerband for a restrictor plate NASCAR engine.

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LIST OF ABBREVIATIONS

CA – Crank Angle

HP – Horsepower

IMEP – Indicated Mean Effective Pressure

FMEP – Frictional Mean Effective Pressure

MEP – Mean Effective Pressure

PMEP – Pumping Mean Effective Pressure

RSM – Response Surface Methods

TDC – Top Dead Center

CHAPTER ONE: INTRODUCTION

The exhaust manifold geometry for internal combustion engines has a significant influence on the dynamic behavior of the exhaust flow. Therefore, it has a large effect on the gas exchange process parameters, such as volumetric efficiency, residuals and back flow or short circuit phenomena.

Computer simulation has been used extensively in the development of intake and exhaust systems. Considerable effort can still be required to identify an optimum design. Response Surface Methodology (RSM) is a collection of statistical and mathematical techniques useful for developing, improving, and optimizing processes. RSM can investigate a performance criterion, and also account for interactions between different control factors. It also provides an alternative to the scientific method, which does not account for interactions between control factors. Using RSM in combination with a validated engine simulation model offers cost savings, time savings, and a quicker path to finding an optimum for a given performance criterion.

CHAPTER TWO: EXHAUST GAS EXTRACTION

The most important mechanism for extracting residual exhaust gas from the combustion chamber at the end of the exhaust cycle is to utilize the kinetic energy of the outgoing exhaust gases to produce a compression wave followed by an expansion wave in which the gas pressure is reduced to a depression in the exhaust port region of the exhaust system. The high-pressure gas from the cylinder expands to the exhaust port rapidly upon exhaust valve opening events. The exhaust gas attains a high flow velocity in the primary exhaust port/pipe. The high-pressure wave travels outwards; the leading compression side raises the pressure while the trailing expansion side reduces its pressure in the exhaust pipe. [1, 2, 3]

By the time the piston has moved up to TDC at the beginning of the induction stroke and the end of the exhaust stroke, the compression wave will have reached the end of the pipe. The speed of the pressure wave pulse greatly exceeds the gas discharge speed through the exhaust port and pipe, caused by the upward moving piston pushing the exhaust gases out of the cylinder and into the exhaust port. [2] Therefore, the exhaust gas on the trailing side of this expansion wave becomes less dense, which causes a corresponding drop in exhaust port pressure, making it negative. This depression, created during the valve overlap period, considerably helps to draw residual exhaust gases out of the combustion chamber and into the exhaust port, while at the same time pulling the fresh charge from the induction port to fill this evacuated space. [1, 3]

Inducting the maximum air mass at wide-open throttle is the primary goal of the gas exchange process. Volumetric efficiency is the parameter that determines this overall mass of intake charge able to be inducted into the combustion chamber/cylinder. The port in the head and the valves themselves make up the majority of the friction losses in the entire system. [2] Therefore, there is less to gain at other locations. However, intake and exhaust manifold tuning still have a significant overall effect on volumetric efficiency of a given internal combustion engine.

The pulsating flow from each cylinder's exhaust process sets up pressure waves in the exhaust system. These pressure waves propagate at the local speed of sound relative to the moving exhaust gas. The pressure waves interact with the pipe junctions and ends in the exhaust manifold and pipe. [1] These interactions cause pressure waves to be reflected back toward the engine cylinder. In multi-cylinder engines, the pressure waves set up by each cylinder are transmitted through the exhaust and reflected from the end of the exhaust pipe - or a significant change in cross-sectional area - can interact with each other. These pressure waves may aid or inhibit the gas exchange processes. When they aid the process by reducing the pressure in the exhaust port toward the end of the exhaust process, the exhaust system is said to be tuned.

The time varying inlet flow to the cylinder causes expansion waves to be propagated back into the inlet manifold. These expansion waves can be reflected back to the open end of the manifold, causing positive pressure waves to be propagated toward the cylinder. If the timing of these waves is appropriately arranged, the positive pressure wave will cause the pressure at the inlet valve at the end of the intake process to be raised above the nominal inlet pressure. This will increase the inducted air mass, and hence the volumetric efficiency. This is also referred to as a tuned intake system.

Frictional flow losses increase as the square of engine speed. At higher engine speeds, the flow into the engine during at least part of the intake process becomes chocked. Once this occurs, further increases in speed do not increase the flow rate significantly, so volumetric efficiency decreases sharply. Intake and exhaust tuning can increase the volumetric efficiency only over a specific engine speed range. [1]

The exhaust gas mass flow rate and the properties of the exhaust gas vary significantly during the exhaust process. The exhaust gas temperature, for example varies substantially through the exhaust process, and decreases due to heat loss as the gas flows past the exhaust valve and through the exhaust system. [1] Average exhaust gas temperatures are usually measured with a thermocouple. Thermocouple-averaged temperatures are close to time-averaged temperatures.

If the exhaust manifold only has short branch pipes before they merge together, there will be insufficient time for the compression wave to leave behind it a depression capable of pulling out the stagnant gas so that the fresh charge arriving at the inlet port is prevented from entering the combustion chamber in the early part of the induction period. Conversely, if the pipe length is very long the flow resistance may become excessive, creating its own back pressure, which will also slow down the scavenging and filling process.

The exhaust gas speed can be calculated knowing the following:

D - piston Diameter

d – port diameter

S – piston stroke

Vp – mean piston speed

Vg – mean gas speed

N – crankshaft speed

$$V_p = \frac{2SN}{60 \times 1000}(m/s)$$

$$swept \quad cylinder = V_p \times \frac{\pi}{4} D^2 (m^3/s)$$

$$Gas \quad disch\arg e \quad volume = V_g \frac{\pi}{4} d^2 \; (m^3/s)$$

$$V_g \frac{\pi}{4} d^2 = V_p \frac{\pi}{4} D^2$$ Therefore:

$$V_g = \frac{SN}{30000}\left(\frac{D}{d}\right)^2 (m/s)$$

This formula only provides a very rough calculation of gas speed since it does not take into account the varying exhaust valve lift. [1]

The study of exhaust gas scavenging depends on being able to estimate the velocity at which sound travels through the exhaust gas; the following calculations are therefore provided. The velocity of a sound wave in a gas is given by:

$$C = \sqrt{\gamma RT}\,(m/s)\,[1]$$

γ = Ratio of molar heat capacities

R =Gas Constant (kJ/kg K)

T = Absolute temperature (K)

The exhaust gases entering the exhaust port are approximately 800°C, but this drops to about 150°C at the tail pipe. Thermocouple data from both the intake and exhaust side of the Dodge

NASCAR Restrictor plate engine was used as duct and junction boundary conditions for the WAVE simulation model.. Every time the exhaust valve opens towards the end of the power stroke a compression wave is released into the exhaust port. This positive pressure-wave pulse travels to the open end of the exhaust pipe where it is expelled into the atmosphere leaving a rarefaction behind, that is, a momentary drop in density of the surrounding air at the pipe exit. The elasticity of the surrounding air will make it rebound towards the pipe exit thus causing a negative wave to be reflected all the way back to the exhaust port. [1,2,3]

When the pulse reaches the exhaust port it will again be reflected towards the pipe outlet as a positive wave. Once again, as it reaches the open end of the pipe a wave will be reflected inwards. This cycle of events will continue indefinitely with decaying amplitude, if time permits, before the next exhaust period discharge takes place.

Discharge coefficient:

$$C_D = \frac{actual\ mass\ \ flow}{ideal\ mass\ \ flow}$$

For best results, the exhaust pipe length should be chosen such that a pressure-wave will travel from the exhaust valve, to the pipe exit and back again during a crankshaft interval 'θ_t' of about 120 ^{o}CA at a given engine speed. This will ensure that the first reflected negative wave is at its lowest pressure when the piston has just passed TDC at the end of the exhaust period. Under these conditions the residual exhaust gas can readily be pulled out (scavenged) from the combustion chamber. However, at lower and higher engine speeds, compared with the tuned exhaust pipe length, the first negative reflected wave will shift relative to the exhaust closure point. Thus, the depression created by the exhaust pulse in the exhaust port will not be able to extract the residual exhaust gases and induce the fresh charge to enter the combustion chamber. In fact, the positive part of the primary or secondary reflected waves may become partially aligned with the exhaust valve closure point and will therefore prevent the expulsion of the residual gases from the chamber. [1]

To take full advantage of the pressure-wave pulse it must be timed so that the first negative reflected pressure-wave reaches TDC towards the beginning of the induction and the end of the exhaust period at its peak negative amplitude. To obtain the correct phasing of the depression wave relative to the closure of the exhaust valve, it is essential to be able to estimate the time it takes the pressure wave to travel through the exhaust gas column from the exhaust valve exit to the end of the exhaust pipe and for this wave to be reflected and returned to its starting point at the exhaust valve exit.

The same principles apply as for induction wave ram cylinder charging, that is, the time taken to travel the exhaust pipe length and back again is equal to the distance the pulse moves from the exhaust valve to the end of the pipe and for it to return to its original starting point, divided by the speed that sound moves through the gas media operating under average working temperature conditions.

$$L = \frac{\theta_t C}{0.012N}$$

L, the total exhaust tract length from the exhaust valve to the exhaust pipe to maximize the wave scavenging effect at a given engine speed. This characteristic length is obviously longer for exhaust gas scavenging compared to induction wave charging due to the higher speed of sound in exhaust gases compared to intake mixture.

Exhaust gas compression wave interference between cylinders by utilizing an idler pipe can be beneficial in producing a depression wave in the exhaust port when the piston is in the TDC region with the exhaust valve still open. [12]

When the exhaust valve opens, the compression wave released travels from the exhaust port to the junction, the increased flow area then causes a sudden expansion of the exhaust gases. This produces a rarefaction that sends a reflected wave back to the open port, thus subjecting the exhaust valve passageway to a slight vacuum. The original compression wave also travels around the forked junction to the blanked end of the idler pipe, and here it is reflected as a compression wave back to the junction, its wave font then divides with one wavefront moving back through the branch pipe to the open exhaust port as the other part of the wavefront travels downstream to the downpipe exit.

The net result is that the negative pressure wave at the open exhaust port is delayed so that it occurs during the TDC valve overlap period, with the piston at approximately TDC. It thereby extracts the residual exhaust gases from the cylinder and induces the fresh charge to enter the cylinder. Likewise, a second cylinder branch pipe with its exhaust valve closed can be considered to be the equivalent to the idler interference pipe. Therefore, similar depressions in the TDC region during valve overlap can be obtained when pairs of branch pipes such as cylinders numbers 1-4 and 2-3 merge into two downpipes, provided the correct length between the port and junction is chosen.

With a two-plane crankshaft there is some unevenness in exhaust port discharge intervals due to the firing sequence. It would be ideal to have each cylinder bank discharge at intervals of 180^o of crankshaft rotation for paired exhaust ports. [1,3]

Computer simulations are extremely useful in identifying key controlling variables to provide guidelines for more rational and therefore less costly experimental development efforts. The behavior of intake and exhaust systems is important because these systems govern the airflow into the engines cylinders. Inducting the maximum airflow at full load at any given speed and retaining that mass within the engines cylinders is a primary design goal. The higher the airflow, the larger the amount of fuel that can be burned and the greater the power produced. If manifold flows are the primary focus, then the models that adequately describe the unsteady gas-flow phenomena, which occur, are required. Then simple models for the in- cylinder phenomena usually suffice to connect the intake and exhaust processes. The valves and ports, which together provide the major restriction to the intake and exhaust flow, largely decouple the manifolds from the cylinders.

Simulation models for exhaust and intake systems are sufficiently advanced because they offer real benefits over traditional design methods. Application of time domain simulation methods and recent increases in computational speed has enabled complex models to be applied to more detailed studies.[5]

Complex flow paths are often difficult to represent accurately within these types of simulation models due to the inherent one-dimensionality of the calculation. Careful consideration and understanding of the flow paths together with suitably flexible modeling elements facilitates simulation of complex flow paths with reasonable accuracy. Wave software has previously shown to be an ideal basis for approaching engine performance, noise levels and sound quality.[5]

Additional sources have shown other simulations that accurately compute the exhaust pressure diagrams and the performance characteristics over a large engine speed Many SAE papers discuss the use of commercially available codes that are available for computation of thermodynamic and gas dynamics behavior for simulation of engine performance. Most are based on the same 1-D conservation equations that WAVE software utilizes. Two and three-dimensional effects can be important and can be modeled with multidimensional gas dynamic flow models. [6,7,8] This may be useful for with future experimentation.

Ricardo WAVE software is a detailed multi-cylinder reciprocating engine simulation code. Its various sub-models require a number of input parameters related to combustion chamber geometry, valve flow, manifold configuration, etc. It also provides a fully integrated treatment of time-dependent fluid dynamics and thermodynamics by means of a one-dimensional finite-difference formulation incorporating a general thermodynamic treatment of working fluids including air, air-hydrocarbon mixtures, products of combustion and liquid fuels. Below is a typical application of how ducts, and junctions would be defined in a WAVE simulation to appropriately define a real world flow path. [9,10]

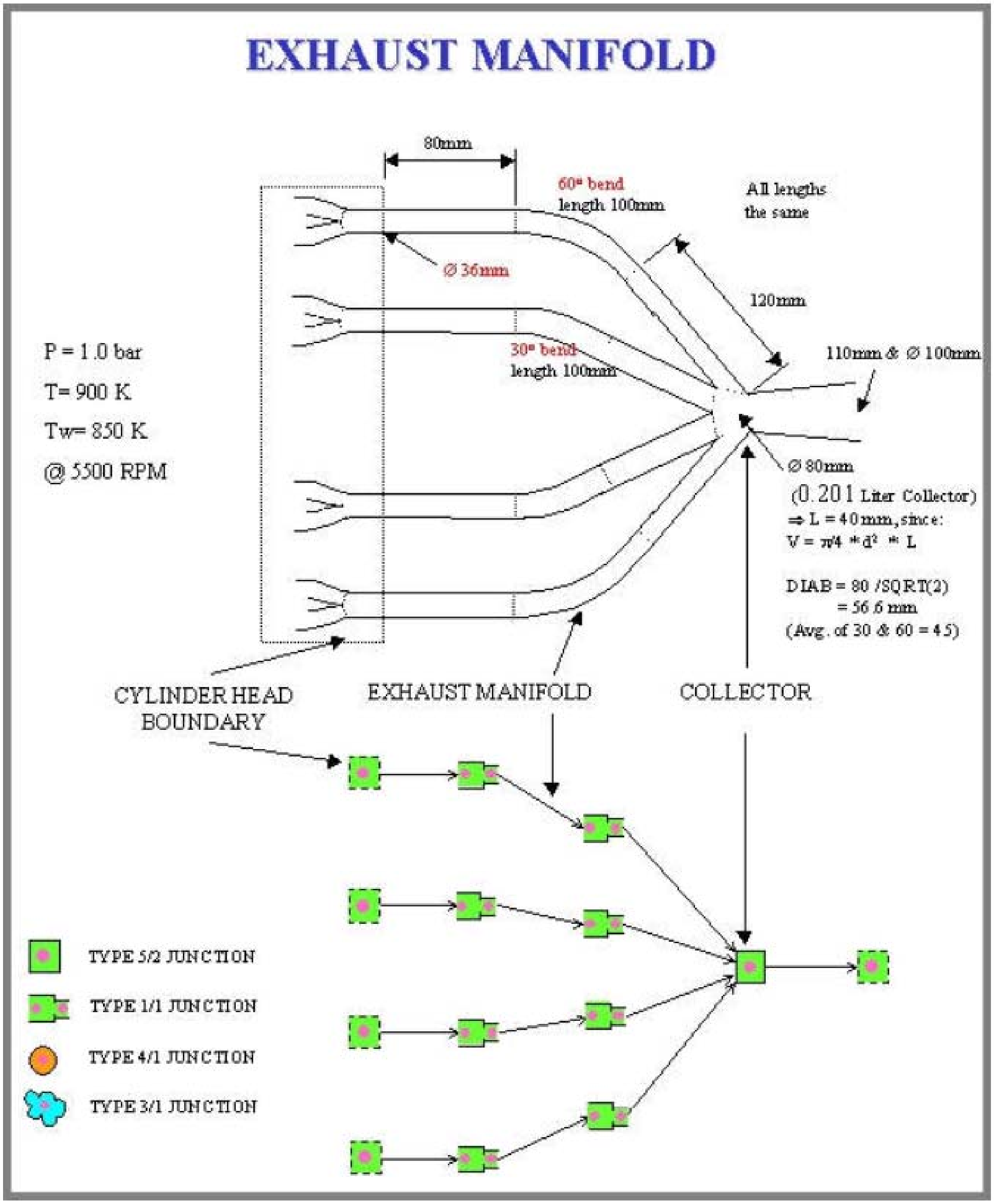

Figure 1: Wave exhaust manifold diagram

The data list below contains items that are either necessary or very helpful to successfully construct and validate a WAVE engine model. [9, 10]

Table 1: Required WAVE input data

Parameter Name	Units
Bore	(mm)
Stroke	(mm)
Connecting rod length, center to center	(mm)
Piston pin offset (positive toward major thrust side)	(mm)
TDC combustion chamber volume	(m3)
Compression ratio	
Number of cylinders	
Firing order	
Firing interval	(°CA)
Two or four stroke	
Rocker arm ratio (if cam lift is prescribed)	
Intake piping and manifold geometry	
Exhaust piping and manifold geometry	
EGR circuit geometry	
Profile of lift vs. crank (or cam) angle	
Valve/cam timing events	
Dynamic valve data (e.g. valve event phase shift vs. engine rpm)	
Tappet type (hydraulic/fixed)	
Valve lash (hot)	(mm)
Rocker arm ratio (if cam lift is prescribed)	
Inner seat diameter (D)	(mm)
Maximum valve lift	(mm)

The overall graphical user interface for the Dodge NASCAR restrictor plate motor is show in figure 2.

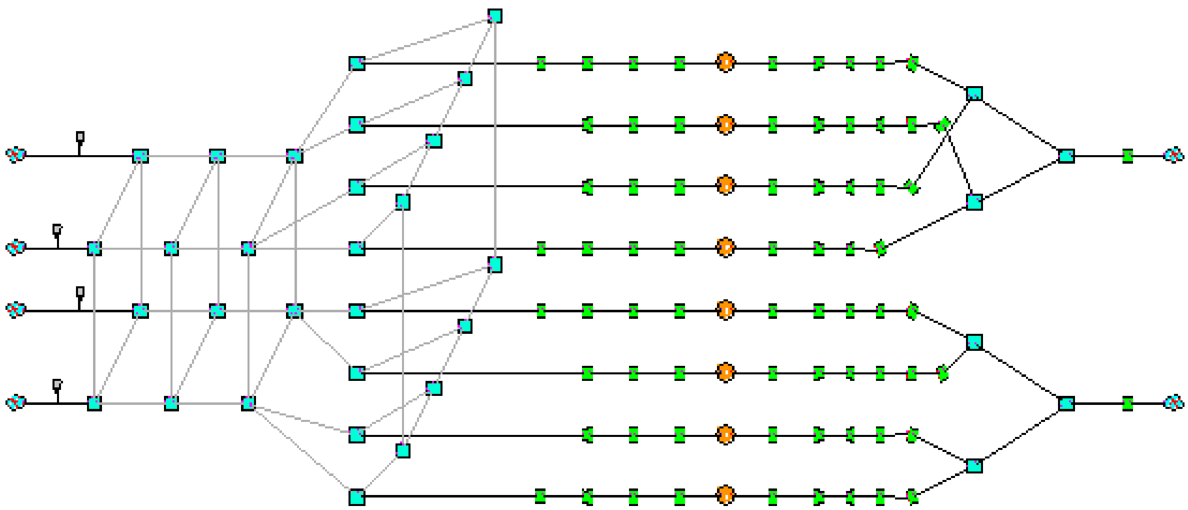

Figure 2: WAVE model overall GUI

The exhaust manifold side of the WAVE model is shown below in figure 3.

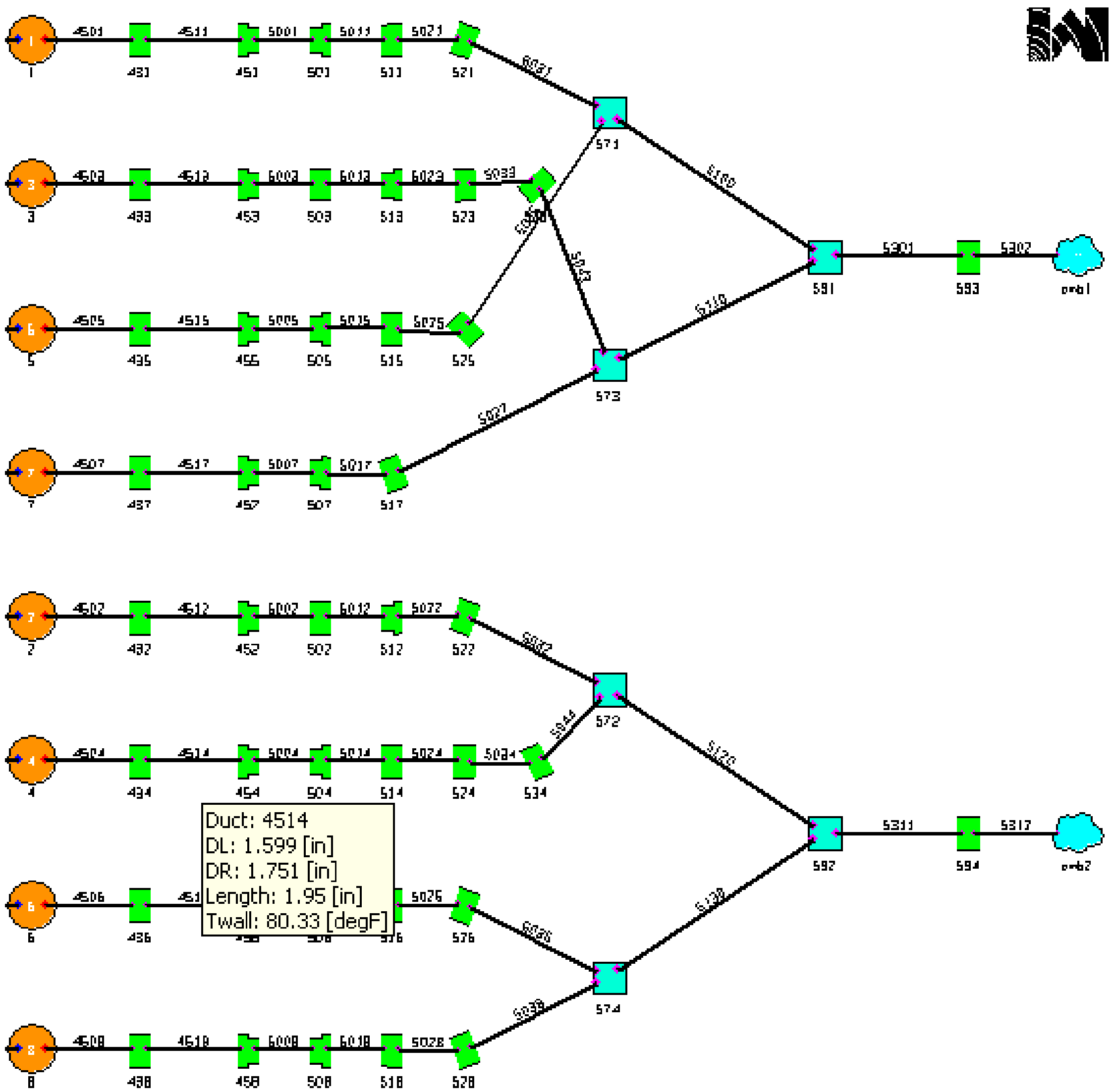

Figure 3: Wave Model exhaust manifold GUI

General engine parameters were measured, along with in cylinder pressure, brake horsepower, and inlet and exhaust manifold temperatures

CHAPTER THREE: METHODOLOGY

The WAVE model had to be accurate to predict the results of modifications to both the exhaust, and the intake side of the engine. Three different setups were used to test the simulation model developed in WAVE. A summary of these setups is shown below in Table 2.

Table 2: Model Validation - WAVE setup descriptions

	Description	
	Restrictor Plate Size	Exhaust manifold
Setup A	29/32	Baseline setup
Setup B	7/8	4" length removed from secondary pipe
Setup C	29/32	4" length removed from secondary pipe

Finally, in order to validate the model with a high degree of precision, it is important to have as much engine test data as possible. This was discussed in the previous section. The goal with the WAVE model was to be accurate within 1% of the available data from the dynamometer runs. After multiple iterations and changes to the WAVE model itself, average IMEP over the 1200 rpm range (6000 – 7200 rpm) and corresponding dynamometer data is show below in Table 3.

Table 3: Model Validation Output

	Experimental Dyno Data IMEP (psi)	Wave Simulation Predicted Output IMEP (psi)	% difference
Setup A	155.60	157.76	1.39%
Setup B	146.61	148.99	1.62%
Setup C	155.21	157.53	1.49%

The overall goal of a Response Surface Methodology is to optimize a response or responses. [11] In this case, the goal is to optimize brake Horsepower output from the WAVE simulation model over a RPM range of 6000 – 7200 RPM.

In general, RSM has a few basic steps, shown below:[11]

- Screening experiments – eliminates factors that are statistically insignificant to the overall model
- Find an 'area' of optimum settings for factors that are significant to the model – steepest ascent techniques
- Collect enough data to fit quadratic terms
- Find an optimum setting – stationary point/canonical analysis
- Identify variability of the response

The initial design consisted of 27 variables to describe diameters and length of each individual section of each bank of the exhaust manifolds.

Below is a diagram and corresponding list of control factors.

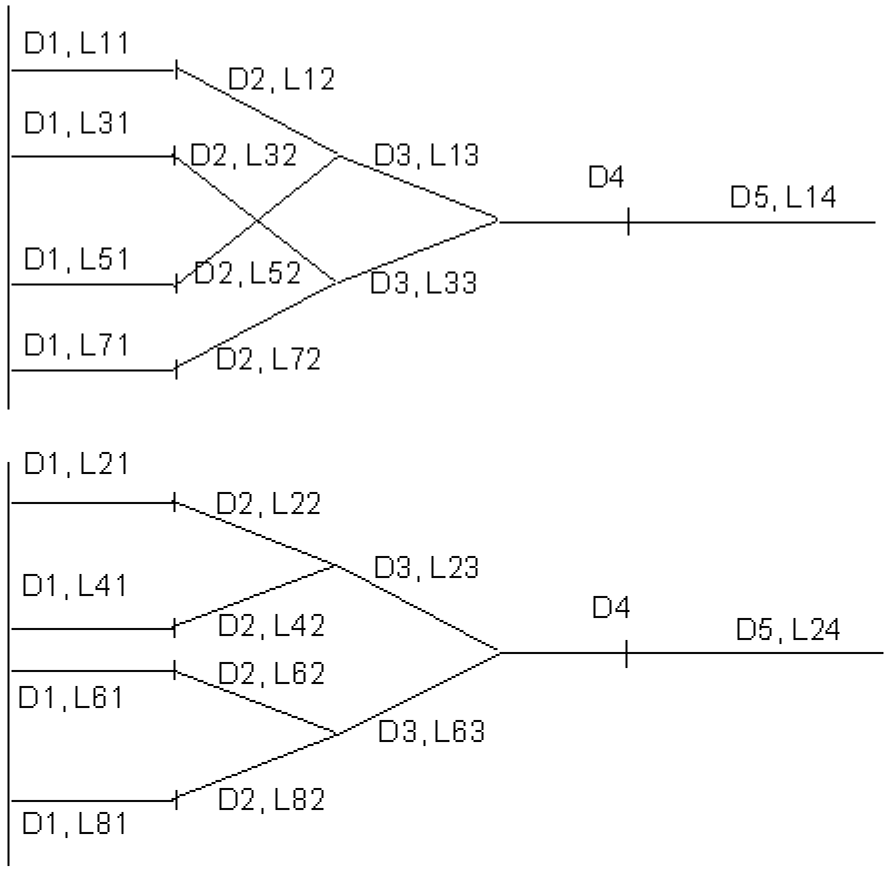

Figure 4: Potential Control Factor Layout

Table 4: Potential Control Factor List

Control Factor	Variable
Primary Runner Diameter (before step)	D1
Primary Runner Diameter (after step)	D2
Secondary Runner Diameter	D3
Choke Diameter	D4
Tertiary Runner Diameter	D5
Primary Runner Length (before step) Cylinder #1	L11
Primary Runner Length (before step) Cylinder #3	L31
Primary Runner Length (before step) Cylinder #5	L51
Primary Runner Length (before step) Cylinder #7	L71
Primary Runner Length (before step) Cylinder #2	L21
Primary Runner Length (before step) Cylinder #4	L41
Primary Runner Length (before step) Cylinder #6	L61
Primary Runner Length (before step) Cylinder #8	L81
Primary Runner Length (after step) Cylinder #1	L12
Primary Runner Length (after step) Cylinder #3	L32
Primary Runner Length (after step) Cylinder #5	L52
Primary Runner Length (after step) Cylinder #7	L72
Primary Runner Length (after step) Cylinder #2	L22
Primary Runner Length (after step) Cylinder #4	L42
Primary Runner Length (after step) Cylinder #6	L62
Primary Runner Length (after step) Cylinder #8	L8\2
Secondary Runner Length Cylinder #1/5	L15
Secondary Runner Length Cylinder #3/7	L37
Secondary Runner Length Cylinder #2/4	L24
Secondary Runner Length Cylinder #6/8	L68
Tertiary Runner Length Bank 1	L14
Tertiary Runner Length Bank 2	L24

A 2 level full factorial, designed experiment would have 2^k Designs. In this case with k = 27, total number of experimental runs would be 134,217,728.to estimate all of the quantitative parameters in the model. [11] Even with a substantial amount of processor power, computer simulation time in WAVE would be substantial to develop output for a full factorial design. A ½ factorial design would still require $2^{(27-1)} = 67,108,864$, and a ¼ fraction would require $2^{(27-2)} = 33,554,432$.[11] It was determined that the control factor list could be simplified. Even though a design with independent lengths for each port may be desirable, at this point in time, analyzing the current design and looking for an optimum range of these settings will be the focus.

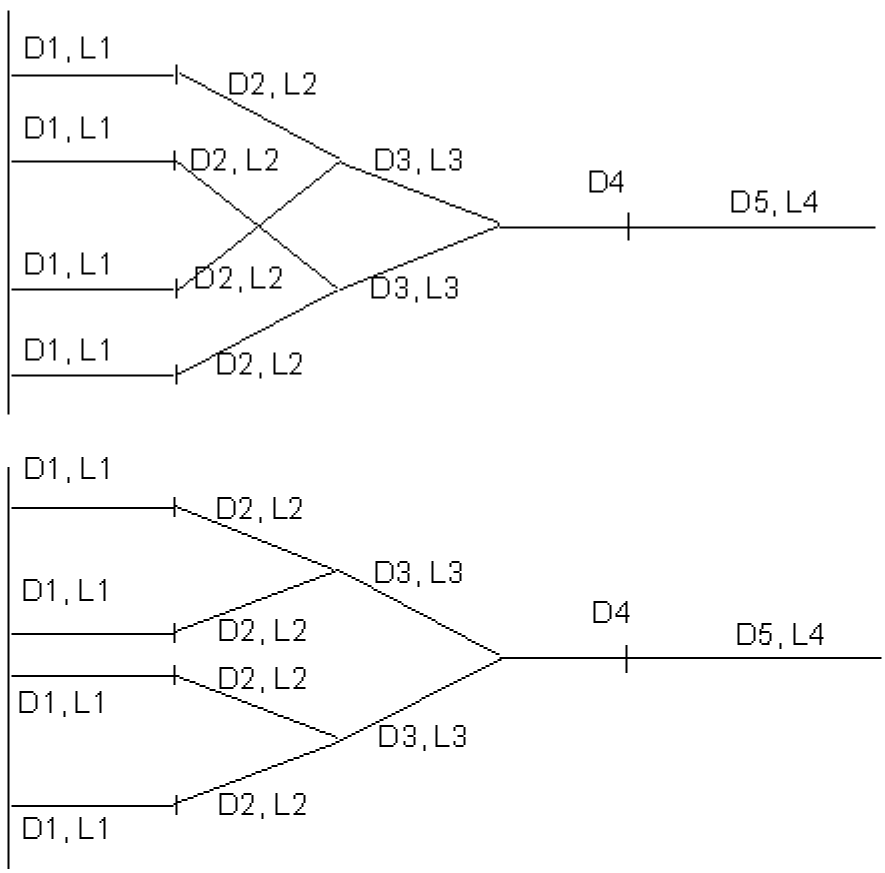

Figure 5: Revised Control Factor Layout

Each bank of the exhaust manifold is considered as nearly symmetric, with each runner having an equal length between junctions. Figure 5 indicates the revised control factor diagram. The control factor range for a 2 level factorial design are as follows in table 5.

Table 5: Control Factor List - Factorial Design

Control Factor	Variable	-1 level	+1 level
Primary Runner Diameter (before step)	D1	1.515	1.770
Primary Runner Diameter (after step)	D2	1.640	1.935
Secondary Runner Diameter	D3	1.935	2.185
Choke Diameter	D4	2.310	2.560
Tertiary Runner Diameter	D5	3.185	3.685
Primary Runner Length (before step)	L1	2.270	12.270
Primary Runner Length (after step)	L2	7.260	17.260
Secondary Runner Length	L3	0.500	10.500
Tertiary Runner Length	L4	12.750	52.750

The high and low settings were chosen from the next larger/smaller size commercially available 304SS/321SS tubing. Initial baseline settings for the exhaust manifold diameters and lengths are as shown in table 6.

Table 6: Baseline Control Factor Settings

Variable	Initial settings
D1	1.640
D2	1.770
D3	2.035
D4	2.370
D5	3.375
L1	7.270
L2	12.260
L3	5.500
L4	32.750

With the baseline settings, WAVE output for average brake Horsepower between the 6000 –

7200 RPM range was 419.6 HP. Upon completion of the regression analysis, it was determined that a linear model adequately approximated the true response, brake Horsepower. The following shows the resulting model, and corresponding type I error rate for each parameter.

$$\hat{y} = 158.50 - 0.908\mathrm{D}_1 - 0.601\mathrm{D}_2 + 0.745D_4 + 0.025L_1$$
$$+ 0.016L_2 - .004L_4$$

$\hat{y}$ - Predicted WAVE output (HP)

D_1 - $\alpha<0.001$

D_2 - $\alpha<0.007$

D_4 - $\alpha<0.005$

L_1 - $\alpha<0.001$

L_2 - $\alpha<0.014$

L_4 - $\alpha<0.001$

This 2 level factorial design determines the factors, D_5 and L_3, can be eliminated from the design, their type I error exceeded the critical value of 0.10, and therefore did not significantly help to predict the response. Additionally, even though the tertiary exhaust length, L_4 was included as a control factor in the experiment; additional physical constraints require that it must also be removed from the design.

From the located optima, the goal is to move outside of the initial design region to a point where response is improved. In the first order model, the direction is parallel to the slopes/coefficients of the approximated linear polynomial response function.

Table 7: Steepest Ascent Summary

Step		D1	D2	D3	D4	D5	L1	L2	L3	L4
	Base	1.64	1.79	2.06	2.44	3.44	7.27	12.26	5.50	32.75
		0.08	0.12	0.03	0.06	0.00	-0.31	-0.17	-1.22	-0.29
1	Base +	1.72	1.91	2.09	2.49	3.44	6.96	12.09	4.28	32.46
2	Base + 2	1.80	2.02	2.13	2.55	3.44	6.64	11.92	3.05	32.17
3	Base + 3	1.89	2.14	2.16	2.61	3.44	6.33	11.76	1.83	31.87
4	Base + 4	1.97	2.26	2.20	2.67	3.44	6.01	11.59	0.61	31.58
5	Base + 5	2.05	2.38	2.23	2.73	3.44	5.70	11.42	-0.62	31.29

Following the steepest ascent, a new full central composite design (CCD) was constructed to explore the region of interest and potentially locate the point of optimality. The upper and lower limits of the control factors are shown in Table 8

Table 8: Control Factor Settings - CCD

Control Factor	Variable	-1 level	+1 level
Primary Runner Diameter (before step)	D1	1.754	1.854
Primary Runner Diameter (after step)	D2	1.973	2.073
Secondary Runner Diameter	D3	2.078	2.178
Choke Diameter	D4	2.504	2.604
Primary Runner Length (before step)	L1	6.140	7.140
Primary Runner Length (after step)	L2	11.424	12.424

A six factor CCD consisted of 45 different exhaust configurations. A simulation of each configuration was performed to determine the brake horsepower output from WAVE. The results were then statically fitted to approximate the true response. Upon completion of the regression analysis, it was determined that a quadratic model adequately approximated the true response minimizing mean squared error. The following shows the resulting polynomial approximation. (for uncoded variable units)

$$\hat{y} = -54.693 + 214.82\mathrm{D}_1 + 88.122D_2 + 76.770D_3 + 71.604D_4 + 4.052L_1 + 0.431L_2 - 55.240D_1^2$$
$$-1.486D_1L_2 - 18.516D_2D_3 - 19.514D_2D_4 - 14.732D_3D_4 - 1.974D_3L_1 + 1.115D_3L_2$$

A canonical analysis was performed on the remaining control factors to determine the configuration, which optimizes brake Horsepower within the design region. These settings for the control factors were then used in WAVE to generate brake Horsepower output from the model. Results are shown in Table 9.

Table 9: Optima Determined by Canonical Analysis

	Initial Settings	Optimal Settings
D1	1.64	1.77
D2	1.77	1.90
D3	2.04	2.25
D4	2.37	2.67
D5	3.38	3.44
L1	7.27	7.83
L2	12.26	13.11
L3	5.50	3.04
L4	32.75	32.17
WAVE response - Brake HP	**419.60**	**421.70**

CHAPTER FOUR: FUTURE WORK AND CONCLUSIONS

Even though optimum brake Horsepower is only 2.1 Horsepower greater than the baseline settings, it would be worthwhile to further investigate the potential increase by fabricating an exhaust manifold representative of the optimum settings. Since dynamometer engine testing is repeatable within 1 HP, a potential 2.1 HP increase would be worth the cost and time to fabricate an additional exhaust manifold for dynamometer testing. These findings are in opposition with previous restrictor plate exhaust work performed by Dr. Todd Dvorak. [12] A similar restrictor plate engine benefited from smaller diameter primary exhaust pipes in the 6000 – 7000 RPM range. This would lead to conclusions that there is a large amount of interactions between exhaust manifold design and intake manifold design. Additionally, dynamometer testing with a larger diameter primary exhaust manifold would also provide additional validation data for the WAVE model.

The designed experiment can still be considered a work in progress. Many different parameters can be modified easily with the WAVE model. Interactions between intake manifold designs, exhaust manifold designs, and valve events would be a new direction to experiment considering the WAVE model accurately represents the inlet and exhaust port and pipe dimensions.

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