ENGINE MODIFICATIONS FOR THE FORMULA SAE RACE CAR

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Abstract

A group of students at the University of Manitoba are involved in the design and construction of a small formula race car to be entered in the Formula SAE student engineering competition. This thesis deals with the modifications made to the engine of this vehicle.

The modifications include building a tuned intake manifold, changing camshafts, increasing the compression ratio and charge cooling. All the modifications are meant to increase the performance of the engine.

The underlying theory for each modification is presented, then the specific modification made to the engine is discussed. Background theory is also given.

Actual dynomometer test results are presented for each modification, in order to prove or disprove the theory. Test results for the compression ratio change are not presented, as this modification has yet to be done.

It was found that many of the modifications resulted in increasing the performance at one engine speed, while decreasing the performance at other engine speeds.

Acknowledgements

I would like to express my sincere appreciation to all those who helped make this thesis possible. I would firstly like to thank Professor W. D. Alexander, my thesis advisor, for his assistance. I would also like to thank Professor J. R. Cahoon for his many suggestions on intake manifold construction and his assistance during dynomometer testing. Also, I wish to thank fellow team mate Mark Budzey who took time away from his own thesis to help in dynomometer testing. Finally, I wish to express graditude to my freinds Doug Hilderman and Lyle Roberecki who both sacrificed a Saturday night to help me in dynomometer testing. Thanks to all.

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A	area
ABC	after bottom dead center
B	bore
BBC	before bottom dead center
BDC	bottom dead center
BHP	brake horse power
bmep	brake mean effective pressure
BTC	before top dead center
cfm	cubic feet per minute
Cp	constant pressure specific heat constant
D	diameter
E	internal energy
f	residual gas fraction
$\mathcal{f}$	Fanno friction factor
F	fuel air ratio
F_r	relative fuel air ratio
h	convective heat transfer coefficient
H	enthalpy
HC	hydro carbons
HP	horse power
imep	indicated mean effective pressure
J	Joule constant
k	conduction heat tranfer coefficient
°K	degrees Kelvin

L	length
m	molecular weight
M_a	air mass flow rate
M_f	fuel mass flow rate
mmep	mechanical mean effective pressure
N	engine speed in rpm
Nu	Nusselt number
η	thermal efficiency
P	pressure
Pr	Prantl number
ρ	density
q	heat flow rate
q"	heat flux rate
Q	volumetric flow rate
r	compression ratio
rpm	revolutions per minute
°R	degrees Rankin
Re	Reynolds number
S	stroke
T	temperature
TDC	top dead center
μ	viscosity
v	velocity
v.e.	volumetric efficiency
w	work
x	distance
Δ	delta, change in

Chapter 1

Introduction

1.1 Preliminary Remarks

In the fall of 1986 three fourth year mechanical engineering students became involved in the design of a small formula style race car to be entered in the Formula SAE student engineering competition. During the fall term of 1986 a spaceframe and suspension were designed for the car. During the spring term of 1987 the frame and suspesion designs were upgraded and revised. This thesis deals with the modifications made to a motor to be used for the vehicle.

1.2 Objectives

The objectives of this thesis are to present the modifications made to an engine to be used for the Formula SAE race car, give the reasons for the modifications and present the underlying theory involved. Also, actual dynomometer test results will be presented and discussed for each modification. The modifications are intake manifolding, changing camshafts, increasing compression ratio and charge cooling.

1.3 Rules of the Competition

There are many rules for the Formula SAE competition, however, there are only a few that pertain to the engine. The rules state the engine must be of the four-stroke variety with a maximum displacement of 610 cubic centimeters or 305 if it is a rotary engine. The engine must run on gasoline, that is, no alcohol or nitro methane is allowed. No nitrous oxide injection is allowed. The rules also state the engine must use a single carburetor and have a 20 mm intake restriction placed between the engine and carburetor. Superchargers and turbochargers are allowed only if placed between the restriction and the engine.

1.4 The Engine

The engine chosen to be used in the Formula SAE race car is a 1984 Honda motorcycle engine. It is 500 cc displacement, has a V-4 cylinder arrangement and is liquid cooled. The engine has 4 overhead camshafts operating 4 valves per cylinder. Stock compression ratio is 11.0 to 1. Engine bore is 60 mm and stroke is 44 mm. Rated torque is 30.4 lb ft at 10500 rpm and rated horsepower is 66 at 11500 rpm. The engine uses one 32 mm carburetor per cylinder and contains an integral transmission. Ignition is by a crank triggered capacitive discharge system with a fully electronic advance.

Initial advance is 15° at 1700 rpm and full advance is 37° at 3000 rpm.

1.5 The Race Engine

Contrary to popular belief, the first priority of a race car engine is not to make maximum power. The first priority of a race engine is to be reliable and durable. Afterall, you cannot win a race if you cannot finish it first.

The second priority of a race car engine is to fit within the rules of the race. Obviously you cannot win a race if you are disqualified for nonconformance with the rules.

The third priority of a race engine is, of course, power output. "Anything for a horsepower" is a motto of many racers. Afterall, it is often only 1 HP that can separate the winner from second place.

Fuel econonmy is a priority, but a much lesser one. Exhaust emissions are of least priority since few races care what comes out of the tail pipe.

In order to convert the production Honda engine into a race engine, several modifications are done. Since the rules require a single carburetor, and the stock engine has one carburetor for each cylinder, an intake manifold is constructed to link the single carburetor to the 4 cylinders. Camshafts are changed in order to increase power, and compression will be increased. Also, charge cooling is used to increase power. Modifications that would reduce the reliability or durability of the engine, such as lightening of the reciprocating mass, were not considered.

Chapter 2

The Engine as an Air Pump

2.1 Introduction

The internal combustion engine can be compared to a simple air pump. It draws in fresh air (and fuel) and pumps out spent air and fuel (exhaust gases). The efficiency of the engine can be directly related to the efficiency of the air pump. That is, the more efficiently an engine can pump air through it, the more efficiently the engine runs. Also, the more air the engine pumps, the greater the power output of the engine since there will be more air available at the time of combustion.

2.2 The Intake Stroke

Consider the intake stroke. The intake valve opens and the piston begins the downward stroke. Since the volume of the cylinder is increasing at a relatively constant temperature, the ideal gas law requires the pressure drop. The pressure differential between the cylinder and the outside atmosphere creates a potential for flow, and the air will try to flow into the cylinder. The air first flows through the carburetor where it picks up fuel, then through an intake manifold, and past the intake valve to finally arrive in the cylinder.

The carburetor, intake manifold and intake valve each represent a resistance to the air flow, but the intake valve restriction is by far the dominant resistance. Thus, any improvement in reducing the valve's air flow resistance will result in large increases of air flow into the cylinder, and similar power increases. The camshaft plays a very important part, since the camshaft design determines the valve motion. The effects of camshaft design will be further discussed in the camshaft chapter.

Finally, at the end of the intake stroke, the intake valve is closed so no air is pumped out during the compression stroke.

2.3 Volumetric Efficiency

At the end of the intake stroke the cylinder is filled with a certain amount of air fuel mixture. A measure of this amount is called volumetric efficiency, and it is usually expressed as a percentage. Volumetric efficiency is defined as the weight of air fuel mixture taken into a cylinder during a normal intake stroke divided by the weight of gas that would fill the cylinder if it were left open to standard conditions of atmospheric pressure and temperature.

Volumetric efficiency changes continually, and is dependent on many factors. Intake manifolding, camshaft design, carburetor size, engine speed, engine size and rod length to stroke ratios are some of the factors influencing volumetric efficiency. In any case, any increase in v.e. will result in increased engine efficiency and increased power output, since at the time of combustion, more air fuel mixture will be available to combust.

Volumetric efficiency can be easily estimated for a specific engine. By doing a combustion chart analysis, the power available could be calculated. Then volumetric efficiency is simply the observed or rated power divided by the calculated power. For the engine used in the Formula SAE race car, the stock volumetric efficiency is estimated to be 79% at 11500 rpm from Appendix A.

2.4 The Exhaust Stroke

Now consider the exhaust stroke. At the end of the combustion stroke the cylinder is filled with combustion residue. The exhaust valve opens and the piston begins its upward travel. Since the volume of the cylinder is decreasing at a constant temperature, the ideal gas law requires the pressure in the cylinder increase. Again there is a pressure differential between the cylinder and the outside atmosphere, and a potential for flow from the cylinder to outside exists. The exhaust valve represents the greatest resistance to this flow. Therefore, any decrease in the exhaust valve's flow resistance will result in increased exhaust flow.

Exhaust flow is important. If exhaust flow is inadequate, some exhaust gas will remain in the cylinder after the exhaust stroke and during the intake stroke. The exhaust gas will not burn again so it effectively dilutes the incoming air fuel charge. Charge dilution results in less effective combustion, lower engine effeciency and reduced power output.

Therefore, any increase in intake volumetric efficiency must be accompanied with an increase in exhaust flow capacity so charge dilution will not increase.

2.5 Real Engines

In the ideal internal combustion engine the intake valve opens instantaneously at exactly TDC and closes instantaneously at exactly BDC. Similarly the exhaust valve opens at exactly BDC after the power stroke and closes exactly and instantaneously at TDC.

Since in the ideal engine instantaneous opening and closing of valves would require infinite accelerations, and infinite forces, in the real engine finite time is given for the opening and closing of valves.

Now consider the intake stroke again. If the intake valve *begins* to open at TDC some amount of time will pass before the valve is far enough off its seat for flow to occur. During this time the piston moves. So some amount of piston travel and some part of the intake stroke is wasted since the valve is not open far enough for flow to occur. Similarly, if the intake valve closes exactly at BDC the last part of the intake stroke is wasted.

But if the intake valve begins to open before TDC, then it will have a head start on the process, and at TDC when the piston begins its travel downward , the intake valve would be open far enough for flow to occur. Similarly if the intake valve closes after BDC the last part of the intake stroke is still useful.

The exhaust valve will similarly open before BDC and close after TDC in the real engine.

2.6 Overlap

Now there is a point at TDC where both the intake and exhaust valves are open at the same time. This period of time is called the valve overlap period, and several things can occur during this period. The events that take place will be further explained in the camshaft chapter.

Chapter 3

Camshaft

3.1 Objectives

Camshaft design can easily be a thesis in itself. Therefore, in order to preserve the continuity of this thesis, only the basic camshaft and valve aspects and events, and their influence and relationship to airflow and engine performance will be discussed. Lastly, the specific camshafts used in the Formula SAE race car engine will be covered.

3.2 Introduction

The camshaft is the brain of the engine. It has the greatest influence on engine performance and dictates to a large extent, engine behavior. Stock camshafts give good idle characteristics and have good low rpm torque. Racing camshafts typically give great high rpm power and little low rpm torque. Racing camshafts usually have very erratic idle characteristics.

3.3 Lift

Net valve lift is the lift as seen by the valve. Valve lift is related to cam eccentricity (cam lift) by the geometry of the valve train mechanisms between the valve and the cam face. Typically, valve lift is a simple multiple of cam lift.

Valve lift affects net flow into, and out of the cylinder. The greater the valve lift, the greater the valve opening, the greater the flow area as seen by the air, the lesser the resistance to flow. Therefore, any increase of valve lift will result in increased volumetric efficiency, up to the point of where the flow is choked in the intake port. For now, refer to this lift as choked flow lift.

In racing engines lifts greater than choked flow lift are frequently used. Because the valve is not instantaneously opened to maximum lift, held there, and then instantaneously closed, but rather opens slowly, reaches a peak lift, and then is closed gradually, average lift becomes important. Once the average lift reaches the choked flow lift, further increases in lift will no longer result in further gains in volumetric efficiency. Therefore, even if peak lift is much greater than choked flow lift, average lift may still be less than choked flow lift.

Peak valve lift is often limited by other considerations. Valve spring bind and piston to valve clearance are the major ones.

3.4 Duration

Duration is defined as the total amount of time a valve is off its seat, and is usually measured in crankshaft degrees. Duration can either be overall duration or effective duration. Overall duration is the time from when the valve *begins* to open to the time when the valve is finally fully closed. Effective duration is the time from when a valve is off its seat far enough for flow to occur to when it is closed far enough for flow to stop. Typically the starting and stopping points of effective duration are at .050 inch valve lift.

3.5 Valve Opening and Closing Rates

The rates at which a valve opens and closes affect the average valve lift. Figure 3.5.1 shows a typical stock cam profile and a typical racing cam profile. The flank of the

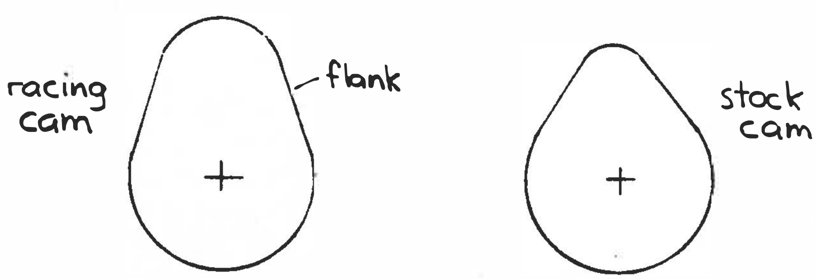


FIGURE 3.5.1 CAMSHAFT PROFILES

racing cam is much steeper, giving much quicker opening and closing rates of the valve. Also the racing cam has a more flat peak causing the valve to dwell longer at peak lift. The overall effect of steeper flanks and a flatter peak is to increase the average valve lift without increasing the peak lift, as well as increase effective duration.

3.6 Valve Events

3.6.1 Intake Valve Opening

Opening the intake valve before TDC causes some problems. When the intake valve opens, the exhaust stroke is just finishing, so the pressure in the cylinder will tend to be greater than atmospheric. Since in a normally aspirated engine, the maximum pressure available in the intake manifold is atmospheric, when the intake valve opens combustion gases will try to flow out of the cylinder into the intake manifold thereby disrupting the intake flow. This phenomenon is called reversion, and it occurs during the first part of the intake stroke. The effect of reversion is to dilute the intake charge, and to reduce average intake vacuum.

Reversion is mainly a problem at low engine speeds. At higher engine speeds, as the flow velocity in the intake port begins to increase, the inertia of the intake charge will tend to dampen the reversion pulse, and the intake flow will be disrupted less. At low engine speeds with very long duration camshafts, reversion causes very erratic idling, poor throttle response and generally poor engine performance.

3.6.2 Intake Valve Closing

Intake valve closing is the most important event in camshaft design. The point at which the intake valve closes determines. to the greatest extent, the engine rpm where peak volumetric efficiency will occur.

Since the intake valve closes after BDC, the piston is moving upward when the intake valve closes. If the intake valve is open far enough at BDC, when the piston starts its upward travel it will push some of the intake charge back out of the cylinder. At higher engine speeds, the inertia of the incoming charge will be enough at some point to continue cylinder filling even though the piston is moving upward. Intake manifold port size determines at what rpm the maximum flow velocity, and maximum inertia, will occur. The intake valve closing event should then be delayed until the maximum inertia is just enough to continue cylinder filling before the intake valve closes. This would maximize cylinder filling and maximize volumetric efficiency.

At lower engine speeds, when intake charge velocity and inertia is much less, some of the intake charge will be pumped back out of the cylinder, resulting in lower volumetric efficiency, and lower average intake vacuum. So if low speed performance is important, a camshaft with shorter duration (earlier intake closing event) should be used. If low speed performance can be sacrificed for maximum high speed power, a longer duration camshaft (later closing intake event) should be used.

3.6.3 Exhaust Valve Opening

Early exhaust valve opening has advantages and drawbacks. If the exhaust valve is opened too early before BDC some of the power stroke will be lost, and engine performance will suffer. However, if the exhaust valve is opened during the last part of the power stroke when cylinder pressure is still high there will be an initial high velocity burst of gas through the exhaust valve. This high velocity gas will possess high inertia and will tend to pull exhaust gas behind it, aiding in the exhaust flow.

3.6.4 Exhaust Valve Closing

Late exhaust valve closing can result in exhaust gas reversion. If the exhaust valve is closed late after TDC when the piston has begun its downward travel for the intake stroke, some of the exhaust gas may be drawn back into the cylinder. At higher engine speeds, when the inertia of the exhaust gas in the exhaust port is high, exhaust gas will tend to be drawn out of the cylinder. and reversion becomes less of a problem. However other problems arise during the overlap period.

3.7 Valve Overlap

The time at TDC when both the intake and exhaust valves are simultaneously open is called the valve overlap period. and it is usually measured in crankshaft degrees. During this period 2 things can happen. If the cylinder pressure at the end of the exhaust stroke is higher than the intake manifold pressure, intake reversion will occur, and some of the exhaust gas will flow back into the inlet stream and dilute the intake charge. If the inertia of the exhaust gases in the exhaust port is high, some of the fresh intake charge may be pulled right through and out of the cylinder. This is called over-scavenging. Both of these 2 conditions result in reduced engine performance.

Typically, engines with long overlap camshafts produce peak volumetric efficiency at higher engine speeds when gas inertias are high enough to dampen reversion, and run poorly at low engine speeds.

3.8 Advancing and Retarding Camshafts

Advancing a camshaft typically improves low speed performance. This is predominantly because the intake valve closes earlier, and low speed cylinder filling is increased. However, advancing a camshaft can reduce high speed engine performance if the peak intake charge inertia is not used to benefit high speed volumetric efficiency.

Retarding a camshaft sometimes improves high speed cylinder filling because the intake valve closes later. If the intake closing event was not delayed enough to maximize inertial cylinder filling initially, retarding the camshaft will delay the intake closing event and high speed cylinder filling will increase. However, retarding a camshaft almost always reduces low speed performance because more low inertia intake charge will be pumped back into the intake manifold when the intake valve closes later.

3.9 The Camshafts for the Formula SAE Race Car

The Honda 500 cc engine uses 4 overhead camshafts. One intake camshaft and one exhaust camshaft is used for each cylinder bank. The stock intake camshaft's specific valve lifts are given in Table 3.9.1. The stock exhaust camshaft is given in Table 3.9.2. Maximum intake valve lift is .292 in. and effective intake duration is 220 crankshaft degrees. The maximum exhaust valve lift is also .292 in. and effective exhaust duration is also 220 degrees. Total valve overlap is 70°.

Table 3.9.1 Stock Intake Camshaft

Time (crank degrees)	Lift (inches)	Time	Lift
20 BTC	.006	110	.2915
15	.0165	115	.290
10	.028	120	.287
5	.044	125	.283
0 TDC	.061	130	.278
5 ATC	.0775	135	.272
10	.0935	140	.2645
15	.112	145	.256
20	.126	150	.2465
25	.142	155	.2355
30	.1573	160	.224
35	.1727	165	.211
40	.1872	170	.198
45	.201	175	.184
50	.214	180 BDC	.169
55	.2265	5 ABC	.153
60	.238	10	.1385
65	.248	15	.121
70	.2585	20	.1065
75	.2665	25	.0895
80	.274	30	.074
85	.280	35	.056
90	.285	40	.0405
95	.288	45	.026
100	.2905	50	.015
105	.292	55	.0075
		60	.002

Table 3.9.2 Stock Exhaust Camshaft

Time (crank degrees)	Lift (inches)	Time	Lift
115 ATC	.0005	85 ABC	.289
120	.003	90	.285
125	.009	85 BTC	.281
130	.0175	80	.275
135	.0295	75	.268
140	.044	70	.2595
145	.059	65	.250
150	.075	60	.240
155	.092	55	.228
160	.108	50	.216
165	.126	45	.203
170	.142	40	.189
175	.1575	35	.174
180 BDC	.172	30	.1595
5 ABC	.187	25	.144
10	.200	20	.128
15	.213	15	.112
20	.224	10	.095
25	.235	5	.079
30	.246	0 TDC	.058
35	.255	5 ATC	.047
40	.264	10	.0335
45	.271	15	.022
50	.278	20	.014
55	.283	25	.0085
60	.287	30	.005
65	.290	35	.004
70	.2915	40	.003
75	.292	45	.002
80	.291	50	.001

Aftermarket high performance camshafts will be used in the Formula SAE race car engine. The intake specifications are given in table 3.9.3. Maximum lift is .303 in. and effective duration 230 degrees. Exhaust valve lift specifications are given in table 3.9.4. Maximum exhaust valve lift is also .303 in. and effective duration is also 230 dgrees. Total valve overlap is 55 degrees.

Figure 3.9.1 shows a comparison of the stock intake valve motion to the aftermarket camshaft's valve motion. It can easily be seen that with the aftermarket camshaft. intake valve closing occurs much later. The difference in maximum lift can also be seen.

Figure 3.9.2 shows a comparison between the stock exhaust valve motion and the aftermarket camshaft's valve motion. The longer duration and higher lift of the aftermarket camshaft can easily be seen.

The higher valve lift and the added duration of the aftermarket camshafts will increase the high rpm power of the engine. High rpm power is needed for racing.

Table 3.9.3 Aftermarket Intake Camshaft

Time (crank degrees)	Lift (inches)	Time	Lift
20 BTC	.001	125 ATC	.303
15	.0065	130	.301
10	.015	135	.298
5	.025	140	.294
0 TDC	.0375	145	.288
5 ATC	.052	150	.2825
10	.0665	155	.2755
15	.082	160	.2665
20	.099	165	.258
25	.1165	170	.246
30	.134	175	.235
35	.151	180 BDC	.223
40	.169	5 ABC	.209
45	.184	10	.195
50	.200	15	.1795
55	.214	20	.163
60	.227	25	.147
65	.239	30	.129
70	.251	35	.111
75	.261	40	.095
80	.270	45	.0795
85	.278	50	.060
90	.284	55	.048
95	.290	60	.034
100	.295	65	.022
105	.300	70	.012
110	.302	75	.005
115	.3035	80	.000
120	.3036		

Table 3.9.4 Aftermarket Exhaust Camshaft

Time (crank degrees)	Lift (inches)	Time	Lift
110 ATC	.002	75 ABC	.303
115	.008	80	.303
120	.015	85	.3015
125	.024	90	.298
130	.0355	85 BTC	.294
135	.049	80	.2885
140	.063	75	.283
145	.079	70	.275
150	.0955	65	.2665
155	.112	60	.257
160	.130	55	.2465
165	.1465	50	.234
170	.164	45	.222
175	.180	40	.2085
180 BDC	.1955	35	.1925
5 ABC	.210	30	.178
10	.223	25	.160
15	.236	20	.142
20	.2465	15	.126
25	.2565	10	.107
30	.2665	5	.090
35	.274	0 TDC	.0735
40	.2815	5 ATC	.058
45	.2875	10	.0435
50	.2925	15	.031
55	.2965	20	.020
60	.300	25	.0105
65	.302	30	.004
70	.3025	35	.000

FIGURE 3.9.1 INTAKE CAM COMPARISON

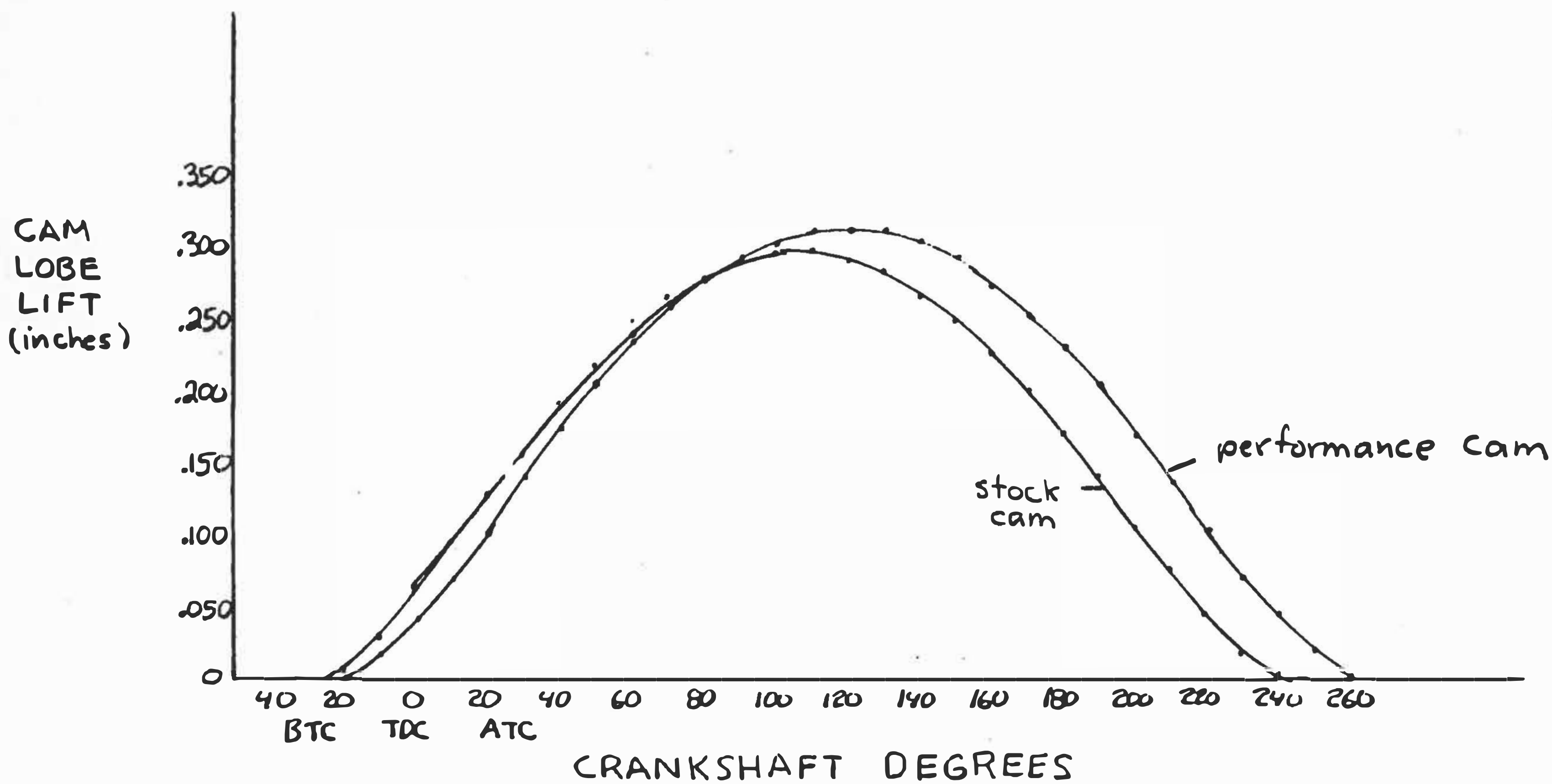
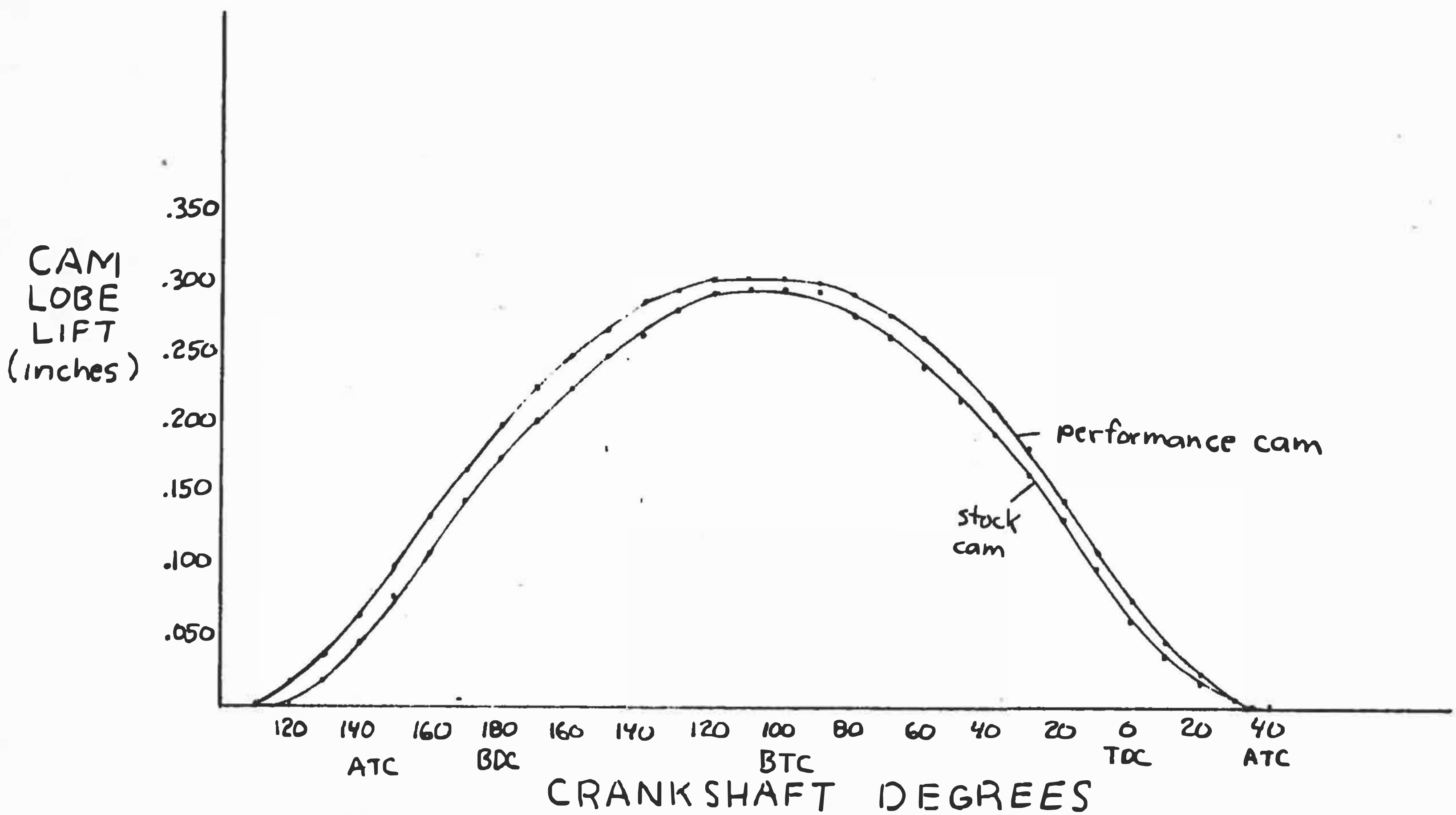


FIGURE 3.9.2 EXHAUST CAM COMPARISON



Chapter 4

Intake Manifold

4.1 Introduction

Because the rules of the Formula SAE competition specify only one carburetor can be used, and since the Honda engine originally had one carburetor for each cylinder, an intake manifold must be constructed to link the 4 cylinders to the single caburetor. Since the engine is to be used for racing, the intake manifold should, if possible, increase the power output of the engine.

There are 3 basic functions of an intake manifold. The manifold should provide equal air flow to all cylinders. prevent fuel from separating from the air stream, and if possible, to boost volumetric efficiency through intake "tuning".

4.2 Air Flow

The intake manifold is responsible for providing equal air flow to all cylinders. If each cylinder is considered as a separate engine, then unequal air flow to each cylinder will cause performance variations from cylinder to cylinder, and overall performance will suffer.

For an intake manifold to supply equal air flow to all cylinders, the intake manifold runners of each cylinder should be geometrically similar, and of equal size. Also, if all the seperate runners are collected into one common chamber, air flow can be more equalized for each cylinder. The common chamber is called a plenum chamber, and it acts as a resevoir of air fuel mixture for each cylinder runner to draw from.

4.3 Separation

Fuel and air have different mass. The air is relatively light in weight whereas the fuel is heavier. The mass differential between the fuel and air becomes a problem when the mixture has to change direction. Since the air is lighter, at a given velocity the air will possess less kinetic energy and inertia than the fuel, and it is easily accelerated around corners. The fuel however, is heavier and possesses more inertia, so it will respond more slowly to changes in direction. Figure 4.3.1 shows the paths that air and fuel might take through a turn in an intake manifold.

Higher velocities in the intake manifold increase turbulence which tends to aid in keeping the fuel suspended in the air stream. So intake manifolds with smaller port cross sectional areas have less fuel separation problems.

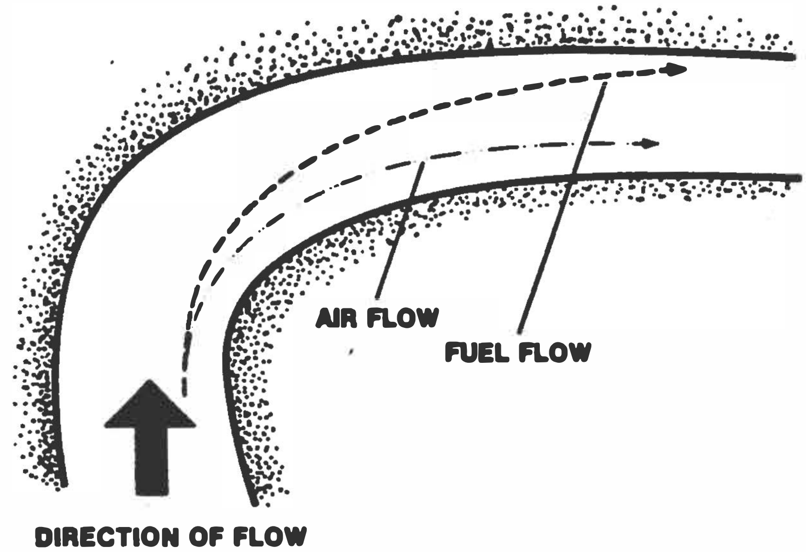


FIGURE 4.3.1 FUEL SEPARATION

Fuel separation directly affects the air fuel ratio that arrives at each cylinder. If there is fuel separation between the carburetor and the cylinders, some cylinders will receive a lean mixture, and performance will suffer. Rejetting the carburetor richer may correct the problem, but it does not elliminate it.

To reduce fuel separation, an intake manifold should be designed with runners geometrically similar, that have no sharp turns. Also, the intake manifold should have no large changes in runner area. If the area changes rapidly, air velocity will change rapidly, but fuel velocity will not. If, for example, the runner area increases rapidly, then air velocity will decrease rapidly, but the fuel will keep going and separate from the air.

4.4 Intake Manifold Tuning

Intake manifold runner size and length can be carefully designed to boost volumetric efficiency at specific engine speeds. The size can be designed for critical velocities and the length can be tuned to "organ pipe" frequencies.

4.4.1 Runner Length

Air fuel mixtures are elastic. When the intake valve opens and the intake stroke begins, the molecules of air nearest to the valve move first. Then the molecules just behind those move, and so on, until the low pressure signal finally reaches the plenum volume and then the carburetor. When the intake valve closes, a high pressure wave is transmitted back from the closing valve. It is similar to "water hammer" when a tap is closed quickly.

The plenum volume acts as a constant pressure resevoir, so when the high pressure wave hits the plenum volume, an expansion (or low pressure) wave is reflected back towards the valve.

This expansion wave can be taken advantage of. As the expansion wave travels towards the valve, it tends to draw intake air along with it. If the length of the intake runner is designed so its natural organ pipe frequency is some harmonic of a specific engine rpm, then at that rpm, the expansion wave's motion will always exactly correspond to an intake stroke, and volumetric efficiency will increase.

The primary engine rpm at which the organ pipe frequency tunes in, is given by:

$$rpm = v/(4L) \qquad (4.4.1\text{-}1)$$

where v = speed of sound (1560 ft/s)

L = length of intake runner'

The harmonic rpms will occur at some even fraction of the primary rpm.

4.4.2 Runner Diameter

The diameter of an intake port can be designed to give critical air velocities at specific engine speeds. Critical velocity is the maximum velocity that can occur in an intake

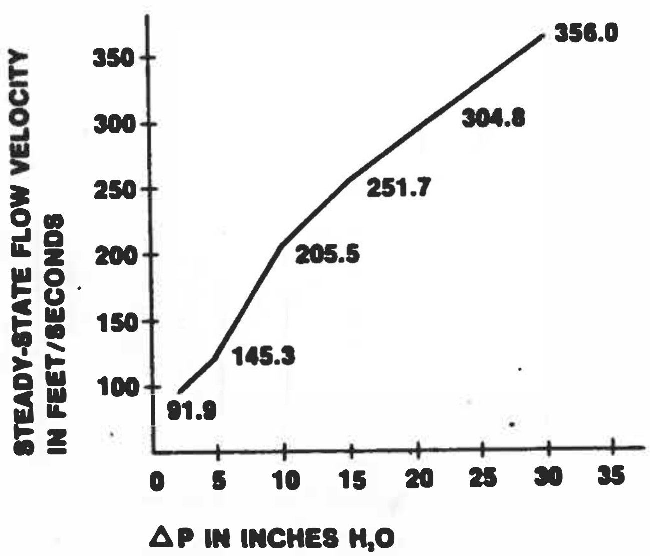


FIGURE 4.4.2.1 CRITICAL VELOCITY

runner. and is a function of cylinder vacuum. Figure 4.4.2.1 shows critical velocities for various amounts of vacuum. Engines with long duration camshafts typically have less vacuum, and tend to have slower critical velocities.

Critical velocity is important. It determines at what engine speed the intake air kinetic energy will be maximum. From the chapter on camshafts, it is known that the kinetic energy of the intake air charge can be used to continue cylinder filling after BDC when the piston is moving upwards. When the intake velocity is critical, this kinetic energy effect is at a maximum. Therefore, cylinder filling is at a maximum, volumetric efficiency is at a maximum, and a torque peak will be realized.

Intake air runner velocity can be found from:

$$v = v.e. \times N \times 2 \times d/A \qquad (4.4.2\text{-}1)$$

where v.e. = volumetric efficiency

N = engine speed

d = cylinder displacement

A = runner cross sectional area

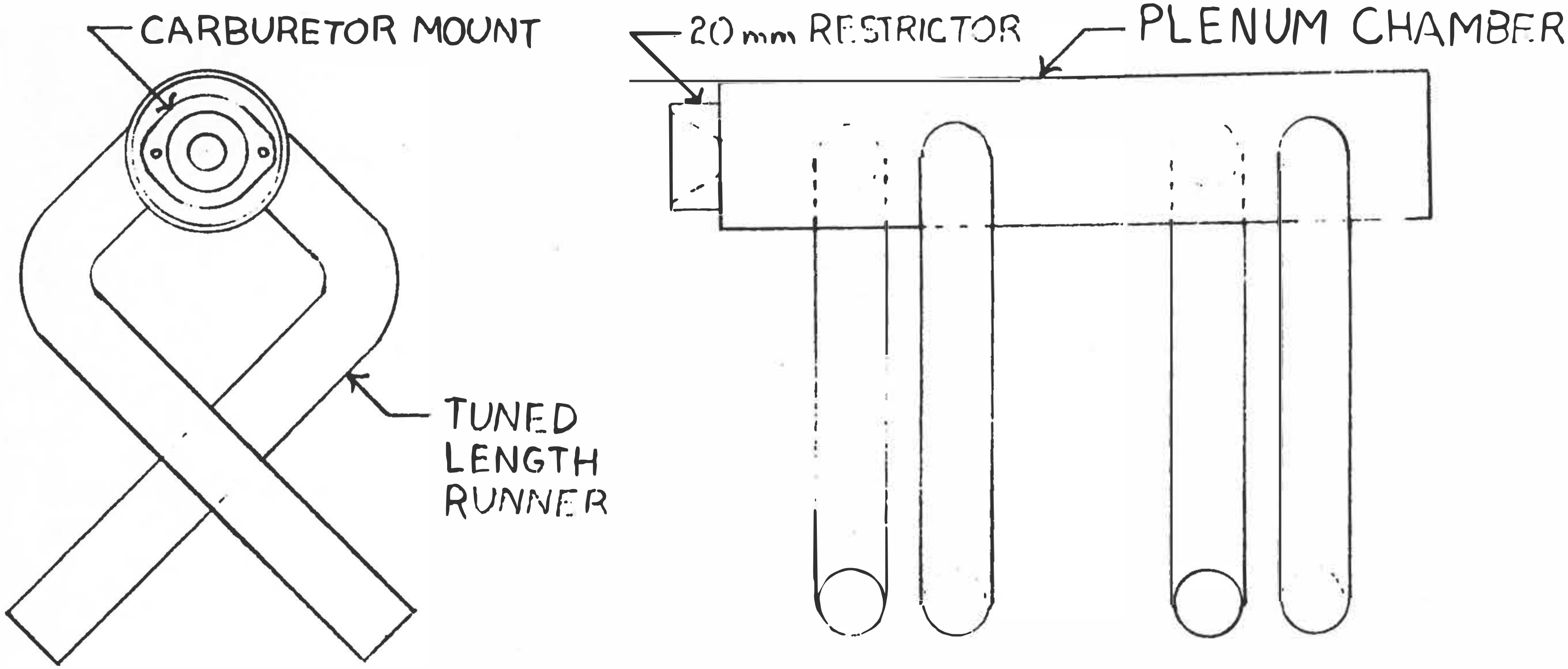
At engine speeds above the critical velocity engine speed, the time to fill the cylinder is shorter (the intake stroke is quicker), but since the velocity can no longer be increased, cylinder filling is reduced and volumetric efficiency drops.

4.5 The Intake Manifold for the Formula SAE Race Car

Figure 4.5.1 shows the intake manifold to be used for the Formula SAE race car. Four intake runners collect into one common plenum chamber. To equalize air flow and reduce fuel separation, the 4 runners are identical. Runner length is designed to be 14" which gives a secondary resonance at 7329 rpm. Runner diameter is 1.25" which gives critical flow velocity between 11000 and 12000 rpm, depending on engine vacuum resulting from the changing of camshafts. The intake also has removeable 1" sleeves for the runners to bring the critical velocity down to 6500 rpm if desired. The manditory twenty mm intake restrictor is located between the carburetor and the plenum chamber.

Since the fuel must make a sharp 90° turn from the carburetor into each intake runner, some fuel separation is anticipated. To correct the problem, carburetor jet size is increased from a 140 size to a 170 size.

FIGURE 4.5.1 INTAKE MANIFOLD



Chapter 5

Compression Ratio

5.1 Theory

Changing the compression ratio affects the efficiency of the thermal cycle. For a spark ignition internal combustion engine, the thermal cycle is the Otto cycle. Figure 5.1.1 shows the pressure volume diagram of the Otto cycle. Process 1 to 2 is compression, 2 to 3 is constant volume heat addition, 3 to 4 is expansion (work output), and 4 to 1 is heat rejection. The enclosed area is the net work output.

Since thermal efficiency is defined as the work output divided by the work input (where work input includes the work of compression and the heat addition), any increase in work output will increase the thermal efficiency. Also, any increase in efficiency will increase the work output for a given work input. Therefore, if thermal efficiency can be increased, more power will be produced for a given fuel input.

Increasing the compression ratio increases the thermal efficiency of the cycle. If, on Figure 5.1.1, compression is further increased from V_2' to V_2, the shaded area will be the increase in net work output.

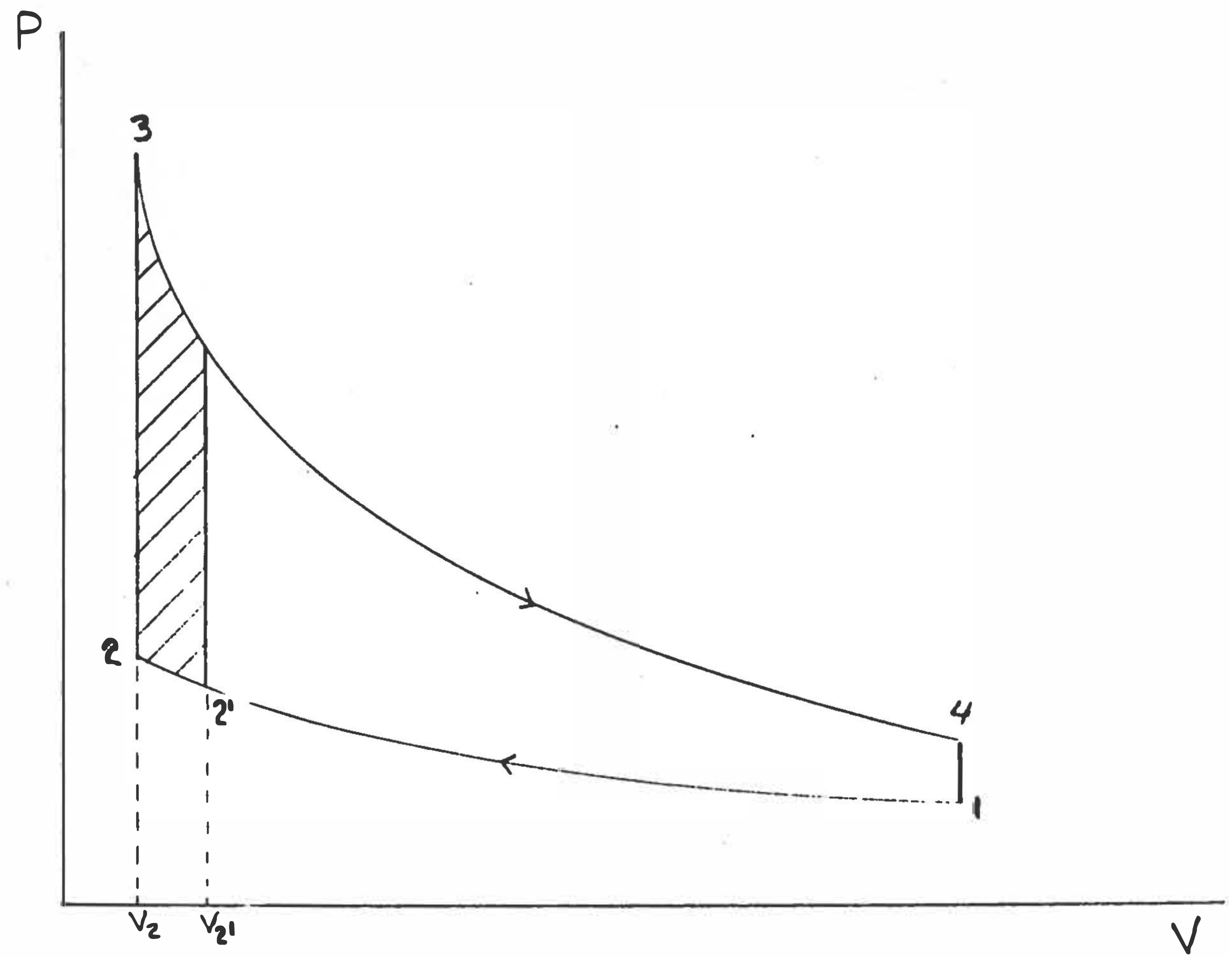


Figure 5.1.1 Otto Cycle

The octane rating of the fuel used limits the compression ratio. If compression is increased, and a fuel with a higher octane rating is not used, then detonation will occur. Detonation is the instantaneous combustion of the fuel air mixture. When detonation occurs, shock waves are transmitted throughout the engine, and after some time, engine damage will surely result.

There is a beneficial side effect of increasing the compression ratio, besides the increase in thermal efficiency.

Consider the end of the exhaust stroke. When the piston reaches TDC it stops briefly, and a small amount of exhaust (equal to the cylinder head clearance volume) will remain trapped in the cylinder. This exhaust gas dilutes the incoming intake charge, and performance will be reduced.

But if the compression ratio is increased, this clearance volume becomes less. So the amount of trapped exhaust gas decreases, intake air dilution decreases, and power output increases.

5.2 Compression for the Formula SAE Engine

The Honda 500 engine has a stock compression ratio of 11.0 to one. If compression is increased, power gains will result.

Calculations in Appendix B show that if a .020 in. head gasket is used, and if .020 in. of material is milled from the cylinder head mounting surface, the compression ratio can be increased from 11.0 to 1, to 14.0 to 1. Since the competition rules allow any type of gasoline, high octane aviation fuel or racing gasoline can be used, and detonation will not occur with r = 14.0.

Further calculations in Appendix B show that changing the compression ratio from 11.0 to 14.0 will result in a 4.5% power increase.

Chapter 6

Charge Cooling

6.1 The Reason For Charge Cooling

The power output of an engine is directly proportional to the mass flow rate of air through it. Any increase in air mass flow will result in power increases.

The rules of the Formula SAE competition specify that all intake air must pass through a 20 mm restriction. The restriction is used to limit power output of the engine by reducing volumetric efficiency. Volumetric efficiency is reduced because the restriction acts as a great resistance to intake air mass flow. If the mass flow of air could be increased through this restriction, large power gains will result. This is where charge cooling becomes important.

The mass flow rate of air is given by:

$$M_a = \rho A v \tag{6.1}$$

where: M_a = mass flow rate of air

ρ = density of air

A = flow area

v = flow velocity

Therefore, if the density, area or velocity can be increased, the mass flow rate of the air will increase.

The area is fixed. Since the rules specify a 20 mm diameter restrictor, any increase in the restrictor area would be a direct violation of the rules.

Velocity is proportional to the root of the pressure difference creating the flow. Once the downstream pressure drops to roughly half of the upstream pressure, the velocity will become sonic. When the flow reaches sonic velocity, further increases in the pressure drop will no longer result in increases of velocity, so the velocity remains fixed.

The density of the air, however, can be increased. Since air behaves as an ideal gas, density will vary according to the ideal gas law:

$$\rho = P/(RT) \tag{6.2}$$

where: P = pressure

R = ideal gas constant

T = temperature

By looking at the ideal gas law, the reason for charge cooling can be seen. Density is inversely proportional to temperature. By decreasing the temperature of the charge, density is increased, and therefore mass flow is increased. Therefore power output is increased. Of course, any increase in the inlet air density must be accompanied by a similar increase in carburetor jetting, in order to maintain the correct air fuel ratio.

6.2 The Charge Cooler for the Formula SAE Race Car

The charge cooling apparatus used for the Formula SAE race car is a simple one pass shell and tube heat exchanger. Figure 6.2.1 shows a sketch of the basic heat exchanger. Air is ducted through twenty .25 in. ID aluminum tubes, while the polyethelene shell is filled with a mixture of dry ice and ethanol. Each tube is approximately 12 inches long, and the shell measures roughly 12" x 9" x 6". The shell is insulated from the outside by 2" styrofoam insulation. The dry ice and ethanol mixture gives roughly a constant -80°C surface temerature to the tubes.

Appendix C shows complete calculations for the heat exchanger. Assuming an outside air temperature of 30°C, an average engine speed of 11500 rpm and 85% v.e., calculations predict the outlet air temperature to be -30.8°C. This results in an air density increase of:

$$1 - \frac{(30 + 273) \times 100}{(-30.8 + 273)}$$

$$= 25\ \%$$

The 25 % increase in air density would translate into a 25 % increase in power output, however, there is a pressure drop associated with the heat exchanger that decreases power output. The pressure drop through the heat exchanger is calculated to be 3 kPa in Appendix C. Because of the pressure drop, actual engine power increases can not be predicted

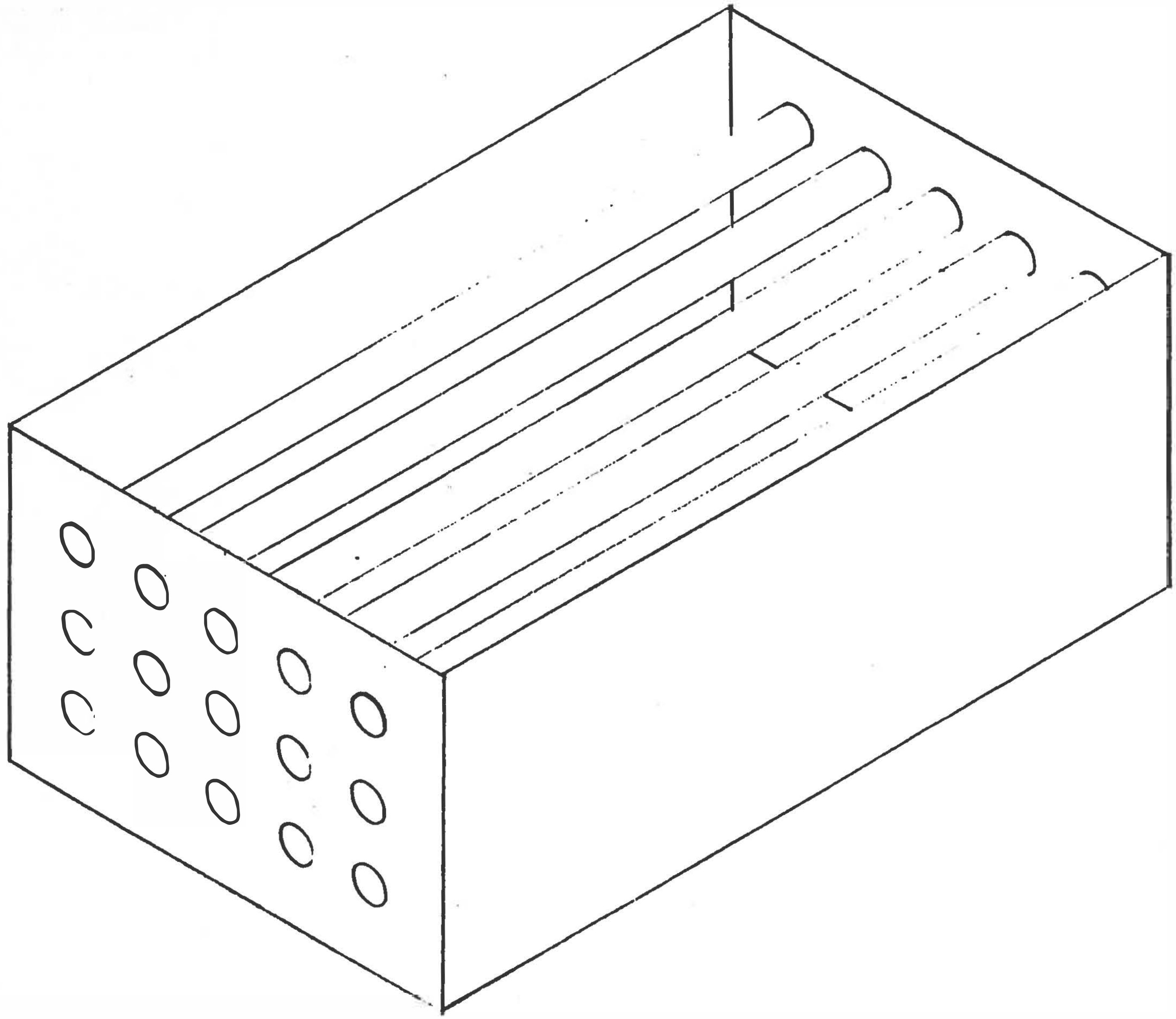


FIGURE 6.2.1 CHARGE COOLER

with sufficient accuracy. However it can be generalized that if the air density increases by 25 %, large gains in power will result.

Chapter 7

Test Results

7.1 Overview

All the engine tests were performed on a Froude dynomometer in the University of Manitoba Mechanical Engineering Engine Laboratory. A test was performed each time a change was made to the engine in order to see how the change would affect engine performance. A SUN exhaust gas analyser was used whenever possible (when it worked!) in order to gain some knowledge on the combustion efficiency of each test.

The dynomometer was coupled to the transmission output shaft of the engine, so all tests were performed with a speed reduction. The dynomometer tachometer was calibrated and found to be accurate to 3500 rpm. Therefore, third gear of the transmission was used so the maximum engine speed of 12500 rpm would correspond to a dynomometer speed of 3285 rpm.

Continuous 12500 rpm engine testing eventually took its toll. Bolts began to rattle loose from the engine, resulting in oil leaks, vacuum leaks and improperly grounded ignition coils. Also, the intake manifold prototype began to errode from the gasoline dissolving the ABS plastic which it was constructed of. The intake manifold also broke on three occasions.

7.2 Test 1: Stock Engine

The first test performed was on the factory stock Honda engine. It was performed in order to see if the engine would deliver rated performance results, as well as to check if all the dynomometer systems were functioning properly.

Table 7.2.1 gives the actual dynomometer measurements. Table 7.2.2 gives the calculated performance results, and Figure 7.2.1 shows the graph of torque and horsepower vs. engine speed.

Torque output and volumetric efficiency stayed relatively constant from 8000 to almost 12000 rpm. Peak torque was 30.4 lb ft at 10000 and 10500 rpm. Horsepower increased at a constant rate up to a maximum of 63.8 HP at 11500 rpm, and then fell sharply.

7.3 Test 2: Engine with Intake Manifold

In the second test the only change made to the engine was the replacement of the 4 carburetors, with a single carburetor and the intake manifold. Table 7.3.1 gives the dynomometer measurements, and Table 7.3.2 gives the calculated engine performance. Table 7.3.3 gives the exhaust gas analysis results and Figure 7.3.1 shows the graph of engine torque and

Table 7.2.1 Stock Engine Dynomometer Results

Engine RPM	Dyno RPM	Dyno Load (lb)
4000	1051	70
5000	1313	82
6000	1577	76
7000	1840	86
8000	2100	96
9000	2365	97
10000	2628	99
10500	2760	99
11000	2890	94
11500	3022	95
12000	3153	89
12500	3285	81

Table 7.2.2 Stock Engine Performance

Engine RPM	Torque (lb ft)	HP	v.e. (%)
4000	21.5	16.3	56
5000	25.1	23.9	66
6000	23.3	26.6	61
7000	27.0	36.0	69
8000	29.4	44.8	77
9000	29.7	51.0	78
10000	30.4	57.8	80
10500	30.4	60.7	80
11000	28.8	60.4	76
11500	29.1	63.8	77
12000	27.3	62.6	72
12500	24.8	59.1	65

FIGURE 7.2.1 STOCK PERFORMANCE

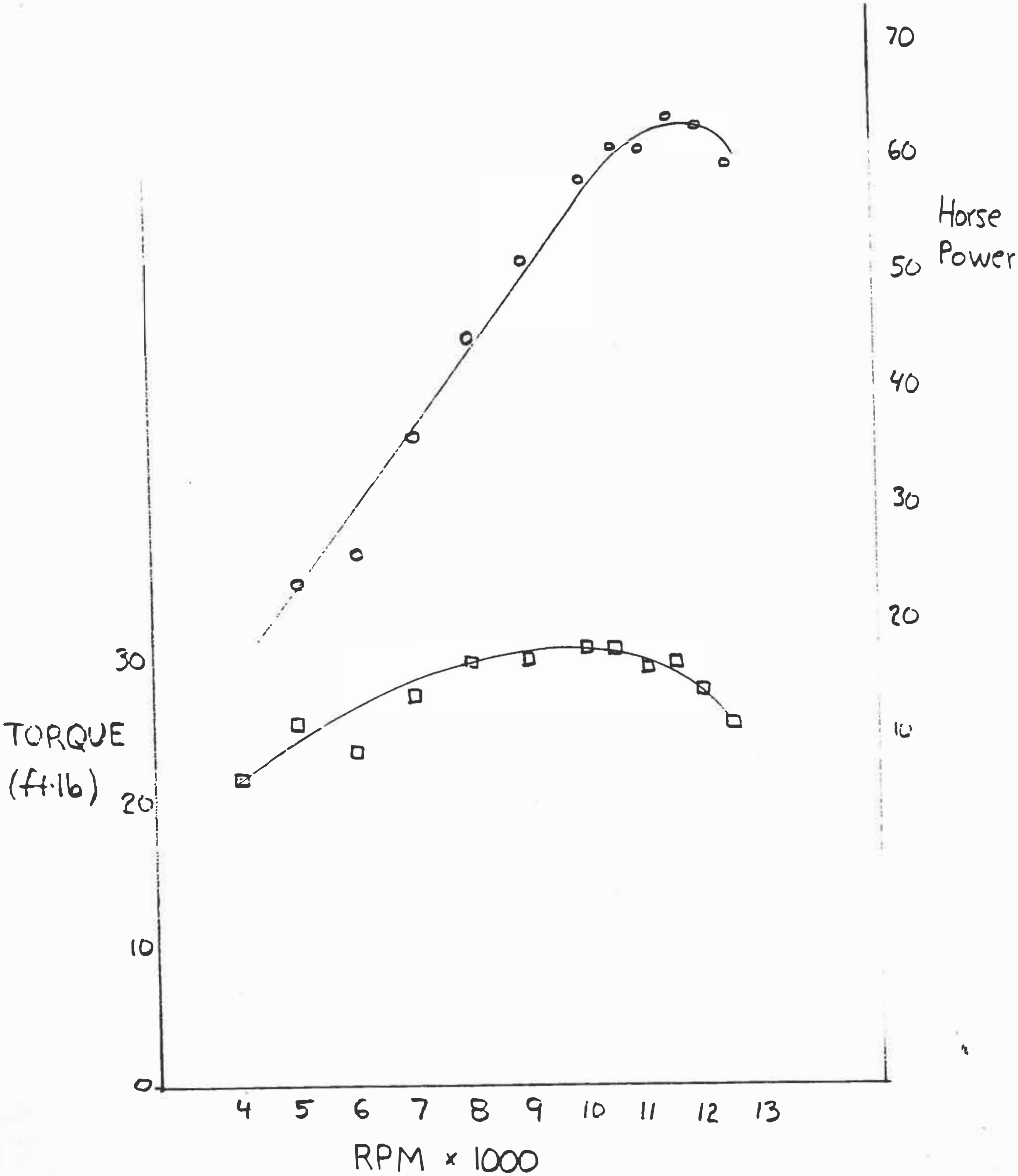


Table 7.3.1 Engine Dyno Results With Intake Manifold

Engine RPM	Dyno RPM	Dyno Load (lb)
5000	1313	45
6000	1577	60
7000	1840	62
8000	2100	74
9000	2365	69
10000	2628	68
10500	2760	56
11000	2890	50
11500	3022	50
12000	3153	46
12500	3285	44

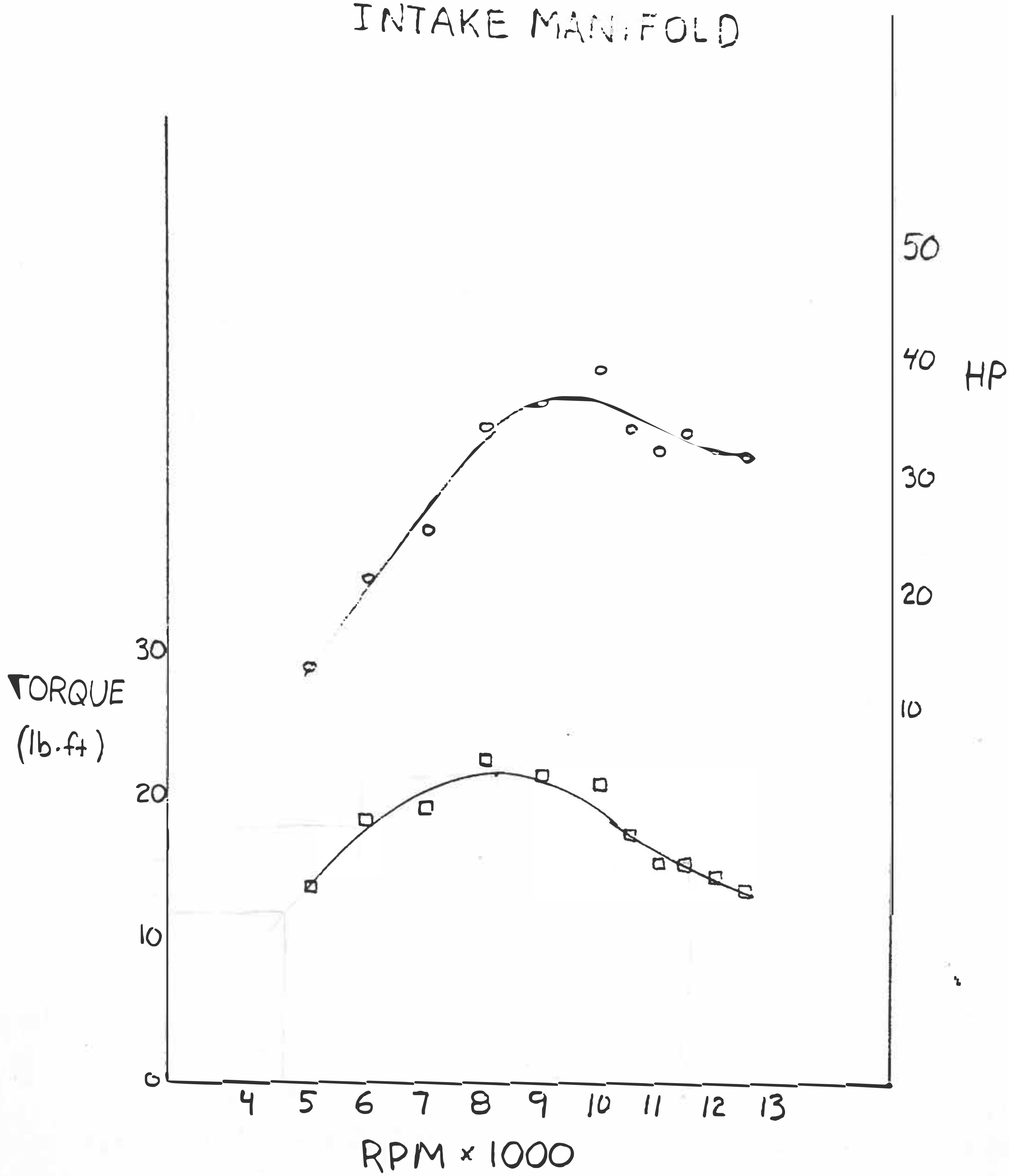
Table 7.3.2 Engine Performance With Intake Manifold

Engine RPM	Torque (lb ft)	HP	v.e. (%)
5000	13.8	13.1	36
6000	18.4	21.0	48
7000	19.0	25.4	50
8000	22.7	34.5	60
9000	21.2	36.3	56
10000	20.9	39.7	55
10500	17.2	34.3	45
11000	15.3	32.1	40
11500	15.3	33.6	40
12000	14.1	32.2	37
12500	13.5	32.1	35

Table 7.3.3 Exhaust Gas Analysis with Intake Manifold

Engine RPM	HC (ppm)	O_2 (%)	CO (%)	CO_2 (%)
5000	424	0.0	1.33	11.93
6000	380	0.0	.18	12.24
7000	377	0.0	1.45	11.93
8000	332	.1	1.00	11.61
9000	254	0.0	1.73	11.20
10000	278	0.0	2.14	12.10
10500	221	0.0	1.53	12.41
11000	394	0.0	2.64	11.24
11500	374	.6	2.38	9.99
12000	284	0.0	2.73	10.60
12500	271	0.0	2.87	10.53

FIGURE 7.3.1 PERFORMANCE WITH INTAKE MANIFOLD



horsepower.

Torque output and volumetric efficiency rose to a maximun at 8000 rpm, and then fell sharply. Maximun torque was 22.7 lb ft. Horsepower rose to a maximum of 39.7 at 10000 rpm and then fell slowly.

Volumetric efficieny dropped substantially at higher rpm, so it was evident that the 20 mm intake restrictor was severely limiting airflow.

The exhaust gas analysis showed very good combustion. Hydro carbon levels, as well as O_2 and CO levels were low.

7.4 Test 3: Small Intake Ports

The third test was performed with 1" sleeves inserted into the runners of the intake manifold in order to see if low speed performance would improve. Table 7.4.1 gives the measured data, Table 7.4.2 gives calculated results and Table 7.4.3 gives the exhaust gas analysis data. Figure 7.4.1 shows a plot of engine torque and power output.

Low speed v.e., power and torque were all improved substantially, however high speed performance was sacrificed. Peak torque of 21.5 lb ft occurred at only 5000 rpm, and then fell off continuously as engine speed increased. Peak power

Table 7.4.1 Engine Dynomometer Results With Small Intake Ports

Engine RPM	Dyno RPM	Dyno Load (lb)
5000	1313	70
6000	1577	61
7000	1840	64
8000	2100	56
9000	2365	56
10000	2628	49
10500	2760	41
11000	2890	36
11500	3022	36
12000	3153	26
12500	3285	22

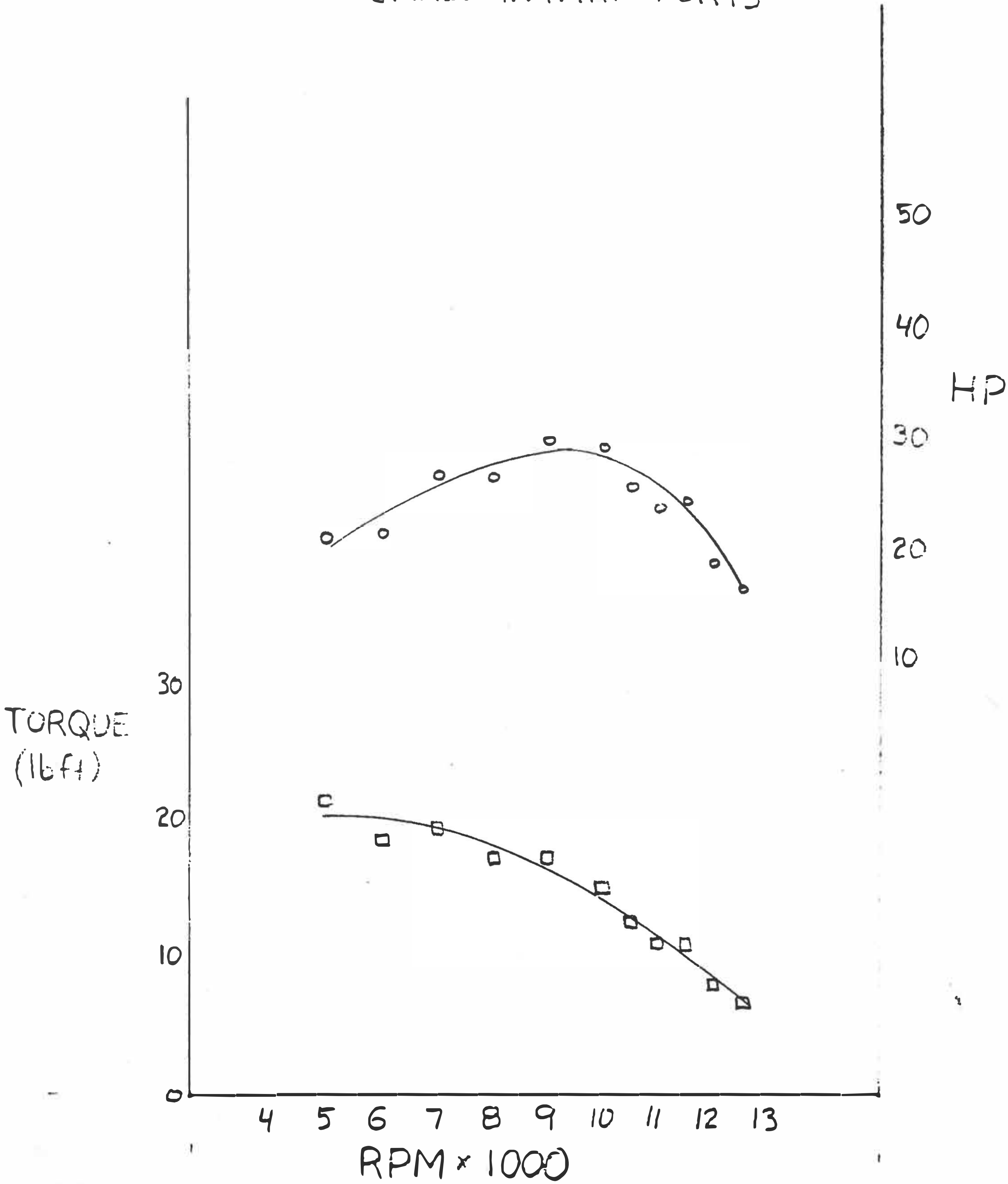
Table 7.4.2 Engine Performance with Small Intake Ports

Engine RPM	Torque (lb ft)	HP	v.e. (%)
5000	21.5	20.4	56
6000	18.7	21.4	49
7000	19.6	26.2	52
8000	17.2	26.1	45
9000	17.2	29.4	45
10000	15.0	28.6	39
10500	12.6	25.1	33
11000	11.0	23.1	29
11500	11.0	24.2	29
12000	8.0	18.2	21
12500	6.7	16.1	18

Table 7.4.3 Exhaust Gas Analysis with Small Intake Ports

Engine RPM	HC (ppm)	O_2 (%)	CO (%)	CO_2 (%)
5000	352	6.6	.09	8.08
6000	244	6.8	.06	7.95
7000	213	6.8	.08	7.95
8000	193	6.1	.12	7.59
9000	165	6.1	.14	8.43
10000	174	4.6	.60	8.91
10500	139	4.7	.49	8.85
11000	118	4.7	.72	8.85
11500	122	4.4	.98	8.78
12000	102	4.1	1.11	9.04
12500	109	3.9	1.59	8.74

FIGURE 7.4.1 PERFORMANCE WITH SMALL INTAKE PORTS



was reduced to only 29.4 HP, however low speed power increased significantly.

The exhaust gas analysis showed very high levels of O_2, but an intake manifold vacuum leak was later discovered that could account for the lean combustion.

7.5 Test 4: Engine with Aftermarket Camshafts

The fourth test was performed with the aftermarket camshafts installed in the engine. Table 7.5.1 gives the dynomometer data, Table 7.5.2 gives the calculated performance and Figure 7.5.1 shows the graph of torque and horsepower output. The exhaust gas analyser did not work for this test.

Mid range (6000 to 8000 rpm) performance had increased, however high speed performance showed no significant gain. This was probably due to the intake restrictor limiting the high speed air flow. At low speeds the engine ran so roughly dynomometer tests could not be run.

Peak torque of 24.7 lb ft occurred at 7000 rpm. Peak horsepower was 40.9 and occurred at 10000 rpm.

7.6 Test 5: Aftermarket Camshafts and Small Intake Ports

In the fifth test, the 1" intake ports were again inserted

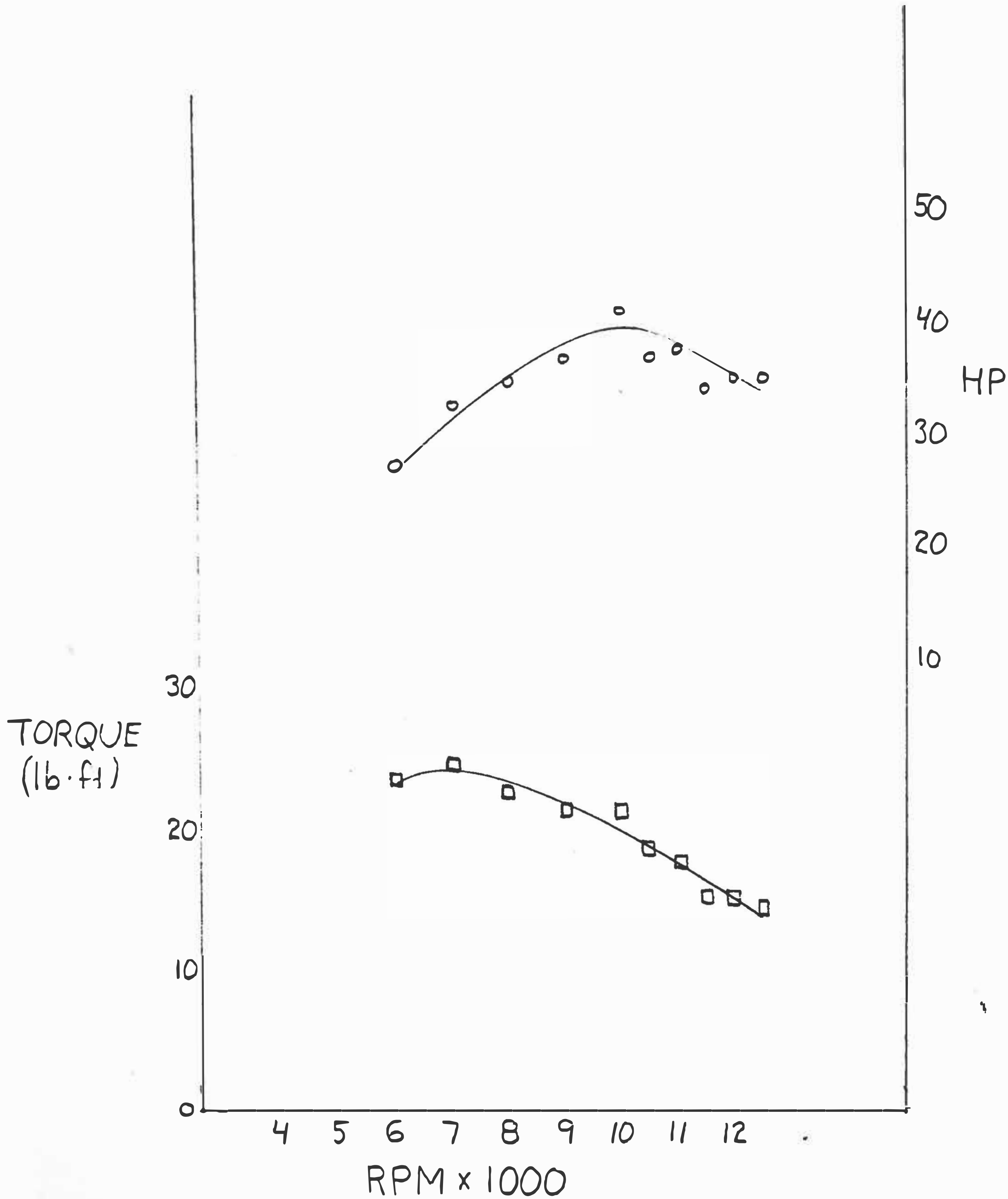
Table 7.5.1 Engine Dynomometer Results With Aftermarket Camshafts

Engine RPM	Dyno RPM	Dyno Load (lb)
6000	1577	78
7000	1840	80
8000	2100	74
9000	2365	70
10000	2628	70
10500	2760	60
11000	2890	58
11500	3022	50
12000	3153	50
12500	3285	48

Table 7.5.2 Engine Performance with Aftermarket Camshafts

Engine RPM	Torque (lb ft)	HP	v.e. (%)
6000	23.9	27.3	63
7000	24.7	32.7	64
8000	22.7	34.5	60
9000	21.5	36.8	56
10000	21.5	40.9	56
10500	18.4	36.8	48
11000	17.8	37.2	47
11500	15.3	33.6	40
12000	15.3	35.0	40
12500	14.7	35.0	39

FIGURE 7.5.1 PERFORMANCE WITH AFTERMARKET CAMS



into the intake manifold to see if the low speed performance could be regained with the aftermarket camshafts. Table 7.6.1 gives the data collected from the dynomometer, and Table 7.6.2 gives the calculated results. Table 7.6.3 gives the exhaust gas analysis data, and Figure 7.6.1 shows the torque and horsepower output of the engine.

Low speed engine performance increased slightly, but high speed performance was reduced significantly. Peak torque fell to 20.2 lb ft at 6000 rpm and peak power fell to 30.0 HP at 9000 rpm.

The exhaust gas analysis once again showed near ideal combustion, except at 5000 rpm where combustion was very inefficient. This was probably due to the long duration of the aftermarket intake camshaft.

7.7 Test 6: Charge Cooling

The last test was performed with the stock camshafts, and with the charge cooler functioning. The test could not be fully completed because of a fuel shortage and a broken intake manifold. However, three engine speeds were tested.

The air temperature leaving the heat exhanger was between 0C and 50C, much higher than calculated. This was due to severe foaming of the ethanol, which prevented the upper tubes in the

Table 7.6.1 Dyno Results With Aftermarket Cams and Small Ports

Engine RPM	Dyno RPM	Dyno Load (lb)
5000	1313	54
6000	1577	66
7000	1840	64
8000	2100	56
9000	2365	57
10000	2628	44
10500	2760	42
11000	2890	47
11500	3022	42
12000	3153	39
12500	3285	34

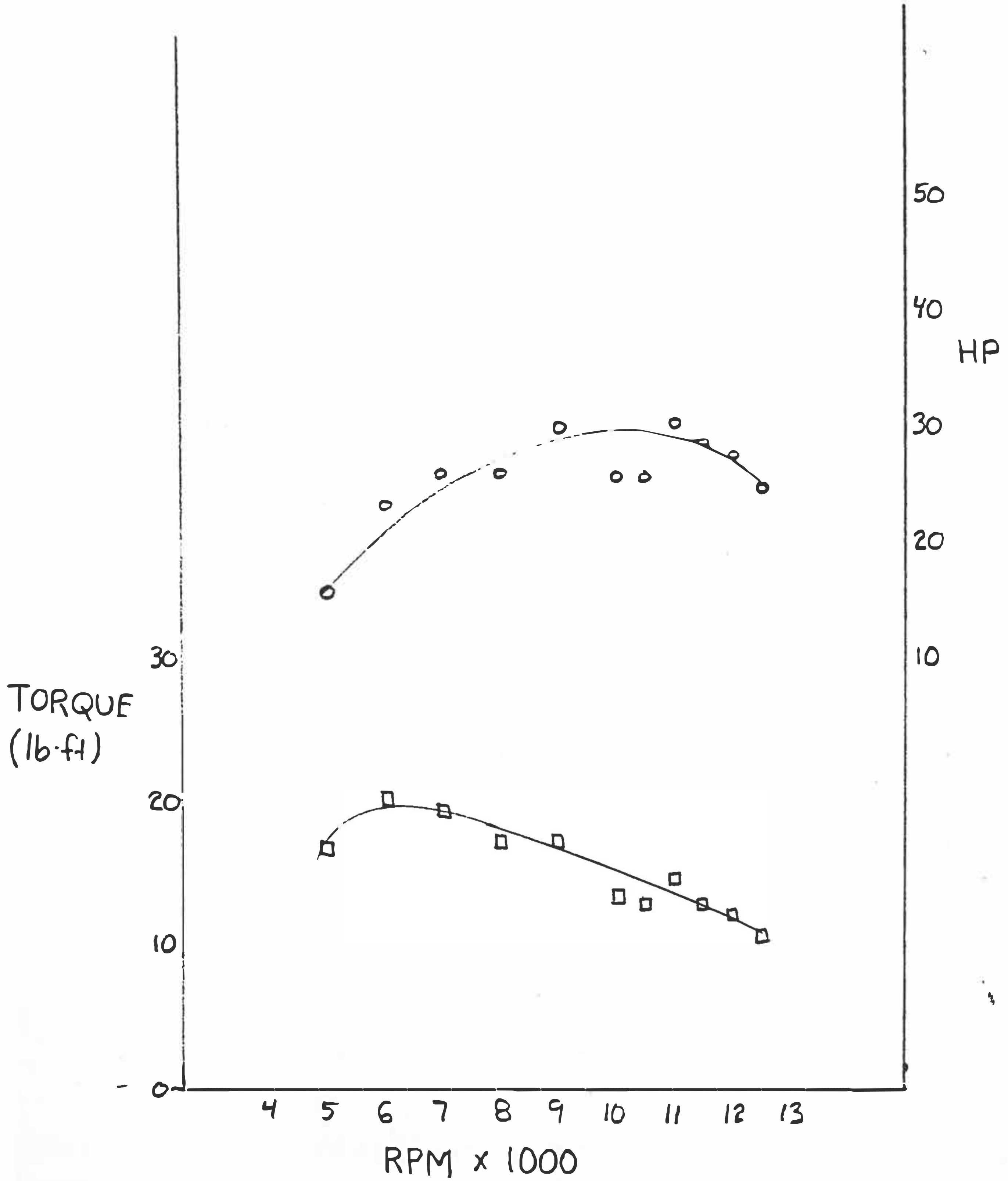
Table 7.6.2 Performance with Aftermarket Cams & Small Ports

Engine RPM	Torque (lb ft)	HP	v.e. (%)
5000	16.6	15.8	43
6000	20.2	23.1	53
7000	19.6	26.2	52
8000	17.2	26.1	45
9000	17.5	30.0	46
10000	13.5	25.7	35
10500	12.9	25.8	34
11000	14.4	30.2	38
11500	12.9	28.2	34
12000	12.0	27.3	31
12500	10.4	24.8	27

Table 7.6.3 Exh. Gas Analysis: Aftermarket Cams & Small Ports

Engine RPM	HC (ppm)	O_2 (%)	CO (%)	CO_2 (%)
5000	2000	1.8	10.0	3.3
6000	459	.2	.14	11.53
7000	312	1.0	.14	11.13
8000	279	.2	.59	11.31
9000	231	0.0	.94	11.61
10000	196	0.0	1.22	12.14
10500	205	0.0	1.36	11.75
11000	271	0.0	2.55	11.10
11500	310	0.0	2.54	11.37
12000	328	0.0	2.96	10.85
12500	339	0.0	3.23	10.77

FIGURE 7.6.1 PERFORMANCE WITH AFTERMARKET CAMS AND SMALL INTAKE PORTS



heat exchanger from being fully submersed in the ethanol.

Table 7.7.1 gives the dynomometer data, Table 7.7.2 gives the calculated performance, Table 7.7.3 gives the exhaust gas analysis and Figure 7.7.1 shows a plot of the engine performance.

It is hard to generalize about the engine performance from the little data that was collected. However, the low speed engine performance was increased significantly, so one could be optimistic about the higher speed performance of the engine with charge cooling.

Table 7.7.1 Engine Dynomometer Results With Charge Cooling

Engine RPM	Dyno RPM	Dyno Load (lb)
4000	1051	75
5000	1313	82
6000	1577	75

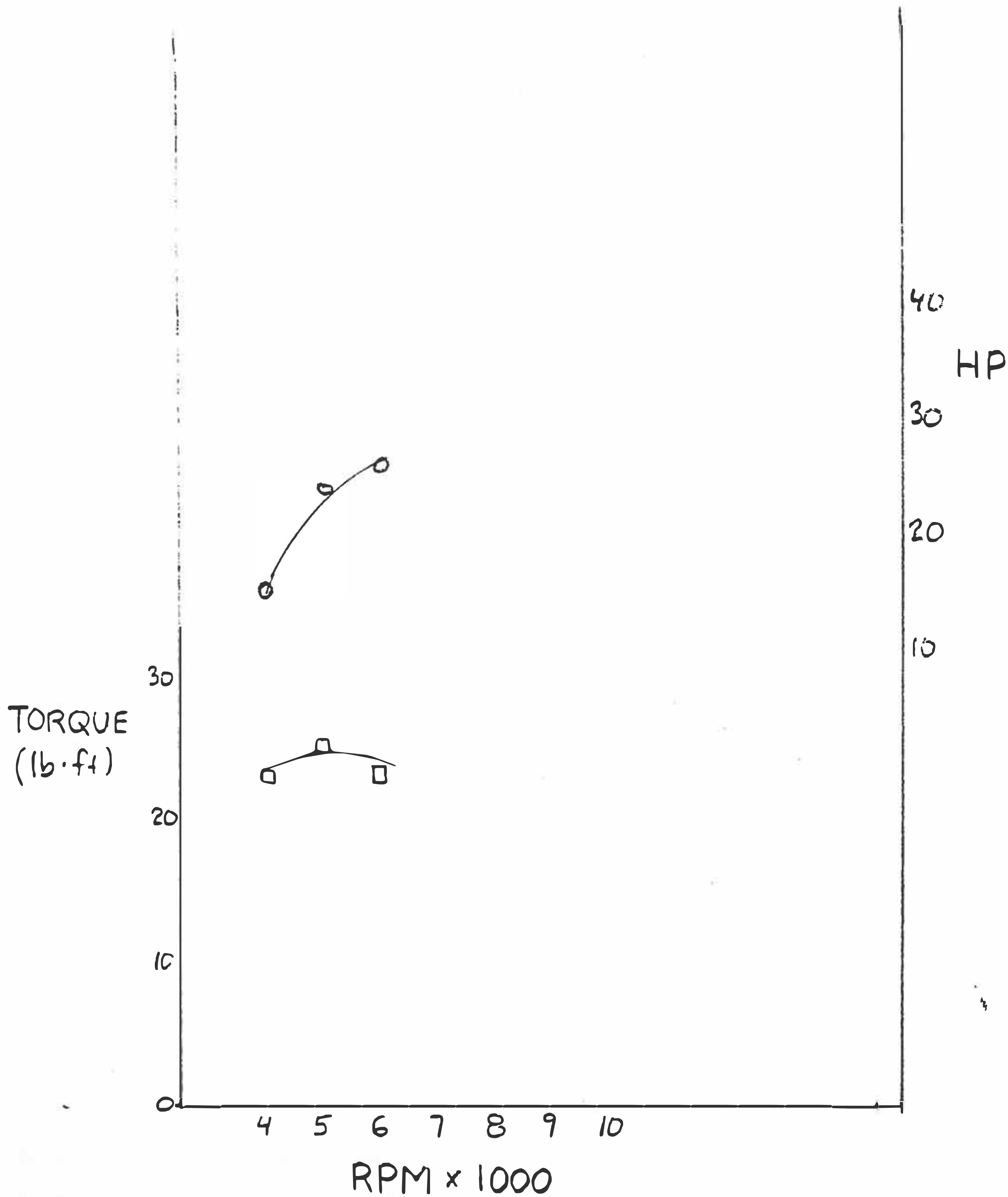
Table 7.7.2 Engine Performance with Charge Cooling

Engine RPM	Torque (lb ft)	HP	v.e. (%)
4000	23.0	17.5	60
5000	25.1	23.9	66
6000	23.0	26.3	60

Table 7.7.3 Exhaust Gas Analysis with Charge Cooling

Engine RPM	HC (ppm)	O_2 (%)	CO (%)	CO_2 (%)
4000	649	3.0	3.50	7.57
5000	900	.8	9.96	4.79
6000	821	.8	9.99	4.49

FIGURE 7.7.1 PERFORMANCE WITH CHARGE COOLING



Chapter 8

Conclusions and Recomendations

The the manditory 20 mm intake restriction severely limits the power of the engine. The resistance to air flow through the restriction at high engine speeds is so great that volumetric efficiency drops substantially.

Because the air flow is decreased at higher engine speeds, the aftermarket camshafts could not be used to their full potential. Consequently the aftermarket camshafts showed no significant power increases at higher engine speeds. The aftermarket camshafts did, however, show substantial decreases in engine performance at low engine speeds. It is therefore recommended that the stock Honda camshafts be used in the engine.

The intake manifold when equipped with the 1" runner sleeves showed large gains in low speed performance. The higher speed engine performance was reduced substantially, and overall power output decreased. It is therefore recommended that the 1" intake sleeves not be used.

The charge cooler could not be fully tested, however the tests that were completed showed favorable results. It is therefore recommended that the charge cooler be used in the

Formula SAE race car.

The compression ratio of the engine could not be increased due to a lack of time. Increasing the compression ratio, however, has no negative side effects if a proper fuel is used. Therefore, since increasing the compression ratio can result only in power gains, it is recommended that the compression ratio of the engine be increased to 14.0 to one.

APPENDIX

Appendix A Volumetric Efficiency Calculations

Bore = 60 mm = 2.362 in.

Stroke = 44 mm = 1.732 in.

r = 11.0 to 1

Factory performance ratings:

HP = 66 BHP @ 11500 rpm

T = 30.4 lb ft @ 10500 rpm

$F_r = 1.0$

F = .0678 lb fuel/lb air

f = .05

m = 29 lb/mole

$(1 + F)/m = .0368$

From chart C1:

T_1 = 580 0R

V_1^o = 430 ft^3/mole

P_1 = 14.7 psia

H_1^o = 1290 Btu/lb mole

E_1^o = 100 Btu/lb mole

$V_2^o = V_1^o/r_c$ = 430/11 = 39.1 ft^3/mole

T_2 = 1289^{o}R

P_2 = 360 psia

H_2^o = 7000 Btu/lb mole

E_2^o = 4400 Btu/lb mole

$V_3^* = V_2^o(1 + F)/m = 39.1(.0368) = 1.439\ ft^3$

$E_3^* = (1 + F)/m(E_2^o) + (1-f)F(H_v)$

$H_v = 19180$ Btu/lb for gasoline

$E_3^* = .0368(4400) + .95(.0678)(19180)$

$= 1414$ Btu

From chart C3:

$P_3 = 1450$ psia

$T_3 = 5200\ ^oR$

$H_3^* = 1800$ Btu

$V_4^* = V_3^* \times r = 1.439(11.0) = 15.83\ ft^3$

$T_4 = 3230\ ^oR$

$P_4 = 81$ psia

$H_4^* = 900$ Btu

$E_4^* = 667$ Btu

$w^*/J = (E_3^* - E_4^*) - (1 + F)/m(E_2^o - E_1^o)$

$= (1414.1 - 667) - (.0368)(4400 - 100)$

$= 588.9$ Btu

$imep = 778/144(w^*/J)/(V_1^* - V_2^*)$

$= 588.9/(15.83 - 1.439)$

$imep = 221.1$ psia

$mmep = 32$ psia from fig 9.8 in [1].

$bmep = imep - mmep$

$= 221.1 - 32 = 189.1$ psia

$BHP = plan/33000nc$

where p = bmep in psia

l = length of stroke in ft

a = piston area in in^2

n = number of power strokes per minute

nc = number of cylinders

BHP = (189.1)(1.732/12)(P/4*2.362²)(11500/2)4/33000

BHP = 83.35 HP

v.e. = rated HP/calc HP

= 66/83.35

v.e. = 79.2 %

Appendix B Compression Ratio Calculations

Increasing Compression Ratio:

Stock compression ratio = 11.0 to 1

$$= (V_1 + V_2)/V_2$$

$V_1 = \pi/4 \times B^2 \times S$

where B = bore = 2.362 in

S = stroke = 1.732 in

$V_1 = \pi/4*(2.362)^2(1.732) = 7.59\ in^2$

compression ratio $r = 11.0 = (7.59 + V_2)/V_2$

$V_2 = .759\ in^2$

Head Gasket:

By using a thinner head gasket V_2 can be reduced and r will increase.

Stock head gasket thickness = .040 in

By using a head gasket of .020 in thickness V_2 can be reduced by $\pi/4(2.362)^2(.020) = .0876\ in^2$

Head Mill:

By milling .020 in. of material off the head mounting surface V_2 can be reduced by $\pi/4(2.362)^2(.020) = .0876\ in^2$

New Compression Ratio:

By performing the above two modifications V_2 wil be reduced to: $.759 - .0876 - .0876 = .584\ in^2$

The new compression ratio is: $(V_1 + V_2)/V_2$

$$= (7.59 + .584)/.584$$

$$r = 14.0 \text{ to } 1$$

Power Increase for r = 14.0

Power increase is directly proportional to brake mean effective pressure (bmep) increase. The indicated mean effective pressure can be estimated by a combustion chart analysis using the combustion charts in [1]. Brake mean effective pressure can be calculated by subtracting frictional mean effective pressure from indicated mean effective pressure.

Combustion Chart Analysis:

Bore = 60 mm = 2.362 in.

Stroke = 44 mm = 1.732 in.

r = 14.0

relative fuel air ratio = F_r = 1.0

fuel air ratio = F = .0678 lb fuel/lb air

residual gas fraction f = .05

molecular weight m = 29

$(1 + F)/m = .0368$

On chart C1:

$T_1 = 580°R$

$V_1^o = 430\ ft^3/lb\ mole$

$P_1 = 14.7\ psia$

$H_1^o = 1290\ Btu/lb\ mole$

$E_1^o = 100\ Btu/lb\ mole$

$V_2^o = V_1/r = 430/14 = 30.7\ ft^3/lb\ mole$

$T_2 = 1372^oR$

$P_2 = 500\ psia$

$H_2^o = 7700\ Btu/lb\ mole$

$E_2^o = 5000\ Btu/lb\ mole$

$V_3^* = V_2^o x\ (1 + F)/m = 30.7(.0368) = 1.13\ ft^3$

$E_3^* = (1 + F)/m(E_2^o) + (1 - f)(F)(H_v)$

$H_v = 19180$ Btu/lb for gasoline

$E_3^* = (.0368)(5000) + .95(.0678)(19100)$

$E_3^* = 1436$ Btu

On chart C3:

$P_3 = 1900\ psia$

$T_3 = 5290^oR$

$H_3^* = 1840$ Btu

$V_4^* = V_3^* \times r$

$= 1.130 \times 14 = 15.82\ ft^3$

$T_4^* = 3120^\circ R$

$H_4^* = 870$ Btu

$P_4 = 80\ psia$ $\qquad$ $E_4^* = 630$ Btu

$$w^*/J = (E_3^* - E_4^*) - (1 + F)/m(E_2^o - E_1^o)$$

$$= (1436 - 630) - (.0368)(5000 - 100)$$

$$= 625.7 \text{ Btu}$$

Indicated mean effective pressure imep:

$$\text{imep} = 778/144(w^*/J/(V_1^* - V_2^*))$$

$$= 625.7/(15.82 - 1.130)$$

$$\text{imep} = 230.1 \text{ psia}$$

Frictional (or mechanical) mep:

mmep = 32 psia from figure 9.8 in [1]

Brake mean effective pressure:

$$\text{bmep} = \text{imep} - \text{mmep}$$

$$= 230.1 - 32$$

$$\text{bmep} = 198.1 \text{ psia}$$

Power Increase:

From Appendix A the bmep for r = 11.0 was found to be 189.1 psia. The pecentage increase in power for r = 14.0 over r = 11.0 will then be:

$$1 - \frac{\text{bmep}_{r=14}}{\text{bmep}_{r=11}} \times 100$$

$$=1 - (198.1/189.1) \times 100$$

$$= 4.5 \%$$

Appendix C Charge Cooling Calculations

The following calculations estimate the effectiveness of the charge cooling heat exchanger.

Total air flow rate:

Q_{tot}= 87 cfm @ 11500 with 85% volumetric efficiency

=.0422 m^3/s

Air density:

inlet ρ_i= 1.1614 kg/m^3 @ 30°C (Texas in summer)

outlet ρ_o= 1.479 kg/m^3 @ -35°C (estimated)

T_s=-80°C

T_{mi}=30°C air ⟶ T_{mo}= ?

L=12" = .3048 m

D=.25" = .00635m

$Q' = Q_{tot}/20$ tubes = .0422/20 = .02211 m^3/s

m= $Q'*\rho_o$ = .00211*1.479 = .00312 kg/s

T_m= 270.5 °K

k = 23.94×10^{-3} W/mK

C_p= 1.0065 kJ/kgK

Pr= .715

μ = 169.85×10^{-7} Ns/m^2

μ_s= 153.10×10^{-7} Ns/m^2

$$Re_d = \frac{4m}{\pi D\mu} = \frac{4(.00312)}{\pi(.00635)(169.85\times10^{-7})} = 36832$$

$$Nu = .027 Re^{.8} Pr^{.33} (\mu/\mu_s)^{.14}$$

$$= .027(36832)^{.8}(.715)^{.33}(169.85/153.10)^{.14}$$

$$Nu = 110.2$$

$$Nu = h_c D/k$$

$$h_c = Nuk/D = (110.2)(23.94\times10^{-3})/.00635 = 415.4 \ W/m^2K$$

$$L/D = .3048/.00635 = 48$$

$$h_{ci}/h_c = 1 + 6D/L$$

$$h_{ci} = h_c(1 + 6D/L) = 415.4(1 + 6(.00635)/.3048)$$

$$h_{ci} = 467.3 \ W/m^2K$$

$$\frac{T_s - T_{mo}}{T_s - T_{mi}} = \exp(-\pi D L h_{ci}/m/C_p)$$

$$= \exp(-\pi(.00635)(.3048)(415.4)/(.003120)/(1006.5))$$

$$= .447$$

$$T_{mo} = T_s - .447(T_s - T_{mi})$$

$$= -80 - .447(-80 - 30)$$

$$T_{mo} = -30.8°C$$

Latent heat of sublimation of dry ice = 590.3 kJ/kg

Melting temperature = -80°C

Density ρ = 1384 kg/m^3

Volume of dry ice in heat exchanger = 12" x 9" x 3"

= 324 in^3

= .00546 m^3

Mass of dry ice $= \rho V$

$= 1384(.00546)$

$= 7.6$ kg

Heat absorbed in sublimation = mass x latent heat

$= 7.6(590.3)$

$= 4462$ kJ

Heat given up by air $= mC_p\Delta T \times 20$ tubes

$= .00312(1006.5)(30-{}^{-}30)20$

$= 3.76$ kW

Time to completely sublimate $= 4462$ kJ/3.76 kW

$= 1184$ sec $= 19.7$ minutes

Pressure Drop:

$$\Delta p = f\rho u_m^2 \Delta x/2D$$

$$u_m = Q/A = .00211/(\pi/4(.00635)^2)$$

$$= 66.6 \text{ m/s}$$

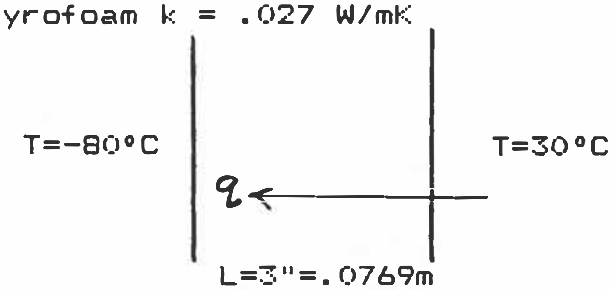
from Fanno diagram $f = .0235$

$$\Delta p = .0235(1.4)(66.6)^2(.3048)/2(.00635)$$

$$\Delta p = 3.505 \text{ kPa}$$

Heat gains through outer insulation:

styrofoam $k = .027$ W/mK



$q_x'' = k\Delta T/L$

$= .027(30-^{-}80)/.0769$

$q_x'' = 38.6\ W/m^2$

$A = 2(6''x9'')+2(12''x9'')+2(12''x6'') = 468in^2 = .3077m^2$

$q_x = q_x''A = 38.6(.3077) = .0119\ kW$

this is negligible compared to the heat gain from the air.

References

1. Taylor, C. F., The Internal Combustion Engine in Theory and Practice., vol. 1,2nd ed., MIT Press, Cambridge Massachusetts, 1985.
2. Taylor, C. F., The Internal Combustion Engine in Theory and Practice ,vol. 2, 2nd ed., MIT Press, Cambridge Massachusetts, 1985.
3. Incropera, F. P., DeWitt, D. P., Fundamentals of Heat and Mass Transfer,2nd ed., John Wiley and Sons, New York, 1985.
4. Alexander, W. D., University of Manitoba, personal communication, 1987.
5. Cahoon, J. R., University of Manitoba, personal communication, 1987.
6. McFarland, Jim, "Fuel Injection", Super power,S A Design Publishing Co., Sante Fe Springs, Calif., 1977.
7. McFarland, Jim, "Understanding Camshafts", Hot Rod,Vol 34 No.12, December 1981, pp. 76-80.
8. McFarland, Jim, "Your Engine the Air Pump", Hot Rod, Vol. 26, No. 7, July 1973, pp. 126-130.
9. McFarland, Jim, "Intake Manifolds", Hot Rod, Vol. 32, No. 4, April 1979, pp. 80-85.
10. McFarland, Jim, "The Intake Manifold", Hot Rod, Vol. 34, No. 9, September 1981, pp. 80-84.

11. McFarland, Jim, "Understanding Intake Manifolds", Popular Hot Rodding, Vol. 23, No. 9, September 1983, p.34.

12. McCaul, Earles 1., "Header Science", Hot Rod, Vol. 26, No. 11, November 1973, pp. 134-139.