DESIGN AND DEVELOPMENT OF A HIGH PRESSURE CNG INTENSIFIER

by

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ABSTRACT

A two-stage, variable-stroke rotary-reciprocating intensifier was built in order to provide high pressure natural gas to a diesel engine. In this application, the intake pressure is variable (20 to 200 bar), as is the mass flow (2 to 50 kg/hr) and the delivery pressure is constant (200 bar). The main feature of this intensifier is that it can provide any mass flow in this range regardless of intake pressure or operating speed.

The mechanism used to provide the variable mass flow uses a variable stroke. This mechanism, as well as the intensifier configuration, was chosen after investigating the alternative design concepts. This investigation showed that the variable stroke would be an energy efficient method of controlling the intensifier mass flow. The specific intensifier dimensions were determined using a graphical design optimization technique.

Prototype testing was limited to speeds below its design operating range because of component failures due to inadequate cooling. In this speed range, the variable stroke proved to be capable of controlling the intensifier mass flow. The rotary-reciprocating geometry was not as energy efficient as originally expected. This shortfall is due to large frictional losses in the rotary gas seals and roller bearings. These frictional losses are essentially constant for all strokes which results in low efficiencies at low mass flows.

It is suggested that a new intensifier design be considered: a configuration which retains the variable stroke capability of the rotary reciprocating design but eliminates the main sources of frictional losses by using non-rotating pistons.
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Chapter 1

**Introduction**

1.1 Alternative Fuels for Diesel Engines

One of the worst contributors to air pollution, fossil fuel burned for transport, has been the target of particularly severe emission regulations. More specifically, diesel engines used in buses and trucks will, in the near future, be subject to stringent new regulations which conventional diesel engine technology appears unlikely to meet.

These regulations mostly restrict the emission of particulate matter (PM) but also place limits on the emission of nitrogen oxides (NOx). The traditional experience in diesel engine design has been that measures which reduce either PM or NOx emissions generally result in an increase in the emission of the other. Even with new engine designs, better electronic control and particulate traps, the solution of the emission problem seems difficult and costly.

An attractive alternative to solve the diesel engine emissions problem is to convert the engine to run on a cleaner burning fuel. Methanol and natural gas seem to be the most promising alternative fuels.

Methanol has been shown to be capable of meeting the emission restrictions on PM and NOx but requires large amounts of energy to produce. Methanol also has undesirable effects such as corrosion of engine parts and in production of aldehydes, which are pollutants though not
regulated at this point. Natural gas seems to be a more attractive alternative since it is relatively inexpensive, readily available and does not have the corrosive effects of methanol.

1.2 Methods of Using Natural Gas

There are a number of methods of converting diesel engines to natural gas fueling:

- conversion of the diesel engine to spark-ignited combustion
- low pressure gas injection with compression ignition
- high pressure gas injection with compression ignition

The advantages and disadvantages of these methods may be summarized as follows [1].

1.2.1 Spark Ignition

In this case, the natural gas is introduced in the combustion chamber via a carburetor or inlet port injector. The fuel-air ratio is controlled over the engine load range by means of a throttle. With close control of the fuel-air ratio of the premixed mixture, catalytic exhaust treatment can reduce emissions of NOx and PM to acceptable levels.

The disadvantages of this method are that in order to prevent detonation, the engine compression ratio must be greatly reduced from the value for diesel operation. Reduced compression ratio and the need for throttling at part load can seriously reduce the efficiency of the engine. Thus conversion of diesel engines to spark ignition can meet emission requirements but at the cost of reduced efficiency.
1.2.2 Low Pressure Injection

Much experience has been gained with low pressure injection of natural gas into diesel engines. Injection is done either directly into the intake manifold or at the intake port just before intake port closure. Operating at full load and using supplementary injection of diesel fuel for ignition of the natural gas, this method of injection can produce high efficiency and reasonably low emissions.

The disadvantages of this method are associated with part load operation. The diesel engine operates without a throttle which results in an air admission rate which is essentially independent of load. Because of this, the mixture fuel-air ratio inside the cylinder becomes too low at part load to support combustion. Emissions become serious because of unburned fuel; at certain loads, damaging auto-ignition can be encountered. Port injection produces better results than manifold injection because it entails less mixing of the fuel and air prior to combustion; this results in higher local fuel-air ratio and more successful combustion. Also, port injection produces better results in two-stroke engines because it prevents waste of fuel in the scavenging period. However, experience with port injection to date indicate that use of low pressure injection of natural gas will not produce satisfactory efficiency and emissions over the whole load range of a vehicle diesel engine.

1.2.3 High Pressure Injection

The advantage of injecting the natural gas directly into the combustion chamber is that it permits the high efficiency of the diesel engine to be maintained over the entire load range. No reduction in compression ratio and no throttling are required for successful operation.
The disadvantage of this method is the need for an on-board pressurization of the natural gas. It happens that the optimum tank pressure for storage of natural gas (approximately 200 bar) is of the same order as the required injection pressure. However, as fuel is consumed from the tank, the associated pressure drop requires recompression before injection.

The task of recompression can be simplified if the natural gas is stored in liquid form (LNG). In this case, the liquid is stored at low pressure in the tank, pumped as a liquid to high pressure and vaporized in a heat exchanger before injection. In certain applications, LNG will be a suitable alternative. However, the disadvantages include the need for natural gas liquefaction, cryogenic containment and gas escape during the venting needed to control the container pressure. These disadvantages make the use of compressed natural gas (CNG) advantageous in vehicle application such as bus fleets in which there is sufficient space on board the vehicle for the larger CNG tanks. Also, most of the infrastructure relating to natural gas is equipped to deal with it in gaseous form.

It is with these considerations in mind that the Department of Mechanical Engineering at the University of British Columbia has undertaken a study of natural gas fueling of diesel engines. The purpose of this study is to evaluate the effectiveness of natural gas at reducing the emissions of the engine. This investigation is using a Detroit Diesel 6V92-TA engine with direct injection of the natural gas which will be stored as CNG. This engine was chosen because is a durable and proven engine widely used in urban buses in North America. If natural gas fueling of the diesel engine can be shown to be an effective method of reducing emissions, then the conversion of urban bus engines to run on natural gas would be an important application.

As discussed earlier, the choice of direct injection of CNG requires on-board recompression of the natural gas. It is the design and development of a method of pressurizing
natural gas for direct injection in a vehicle diesel engine that will be described in this thesis. The term intensifier will be used instead of compressor because the latter is usually used when the intake pressure is constant. It will be shown that the intake pressure in this case is both elevated and variable, hence the term intensifier.

1.3 General Design Requirements

As will be shown on chapter 2, the principal design requirements for the intensifier are:

- high and constant delivery pressure
- inlet pressure varying over a wide range
- mass flow varying over a wide range
- high efficiency
- small size and weight

Additional design requirements include durability, safety and low production and maintenance costs. The preliminary design process revealed that commercially available technology which could meet these requirements does not exist. It also showed the difficulty of maintaining high compressor efficiency over the desired range of intake pressure and mass flow rate. Thus an important goal of this work was to design the intensifier for high efficiency over the operating range.
1.4 Previous Work

The groundwork for the intensifier project was mostly laid out by C. Aichinger and his contributions are contained in his master's thesis [2]. One important contribution of this work is a study of the different intensifier designs to examine which type of design would be suitable for this design application. This study will be examined and expanded in chapter 3.

Aichinger also designed and developed two prototype shaft-driven reciprocating intensifiers. His evaluation of the performance of these intensifiers supplied valuable information on design limitations and component performance. His results strongly emphasized the need for an intensifier design with variable capacity which is the subject of this work and which required a new intensifier concept.

1.5 Goals and Objectives

The general goal of this work is to design and develop a natural gas intensifier that would meet the requirements outlined above with special emphasis placed on achieving high efficiency variable capacity operation. In working towards this goal and after review of specific design requirements and evaluation of different possible design concepts, the reasons for choosing the rotary-reciprocating intensifier configuration were identified. Then followed the evaluation of the pertinent design limitations and the formulation of a computer simulation of the selected design concept. With the simulation and additional design optimization techniques, a final design was developed for production and testing.
The specific objectives of this work are:

1. To design and build a variable-stroke, multi-stage, mechanically actuated rotary-reciprocating intensifier which would meet the design requirements for fuel flow in both design and off-design operation of the diesel engine.

2. To measure the performance of this intensifier (paying close attention to its variable capacity capabilities) in the light of the thermodynamic model used in the design process.

3. To assess the essential design limitations of this new class of variable stroke, high pressure intensifiers.
2.1 Introduction

Chapter 3 discusses intensifier design concepts and determining which intensifier configuration is best suited to the application of natural gas fueling of diesel engines. Before evaluating the different design concepts, the specific design requirements must be known. This chapter will examine the design requirements as determined by the engine demands and the operating conditions in which the intensifier will operate. The design requirements that have the greatest effect on the choice of intensifier configuration are the operating pressures (intake and exhaust pressure), capacity and capacity control. Furthermore, there are practical considerations that will affect the choice of intensifier design concept.

2.2 Exhaust Pressure

Since the intensifier will provide fuel to the injector, the exhaust pressure of the intensifier must be the same as the injection pressure. The present working value for the injection pressure is approximately 200 bar. This value comes from the need for rapid penetration of the gas across the cylinder to avoid excessive combustion duration. Rapid penetration is possible when the velocity of the gas leaving the injector nozzle is sonic or faster. To obtain sonic velocity through
a nozzle, a pressure ratio of approximately 2:1 is required. Since the cylinder pressure at the time of injection is in the region of 80 to 100 bar, an injection pressure of approximately 200 bar is required. Sonic velocities also result in a choked nozzle which has the added benefit that the mass flow through the nozzle is independent of combustion chamber pressure. The 200 bar injection pressure is not a rigidly set value and it is possible that this value may be increased or lowered in the future as testing of the injector progresses.

2.3 Intake Pressure

Since the fuel will be stored as CNG in high pressure tanks, the intake pressure of the intensifier is the same as the tank pressure. The filling pressure for a CNG storage tank is 200 bar which sets the maximum intensifier intake pressure. This 200 bar filling pressure comes from the fact that it is close to the pressure which, at ambient temperature, maximizes the ratio of mass of gas contained to the mass of the tank required. The optimum occurs near 200 bar due to the shape of the compressibility curve of natural gas.

As for the minimum tank pressure, there is no theoretical lower limit since the tanks could be emptied until they contain a partial vacuum. This is not feasible as it would mean a very high pressure ratio for the intensifier. Even if the tanks were only permitted to go down to atmospheric pressure (approximately 1 bar), the pressure ratio required would be 200:1. This would require a very large machine due to the number of compression stages required. For this reason, a lower limit must be arbitrarily set. This lower value must be chosen in such a way that the tanks are nearly empty since this would minimize the amount of tanks required on the bus (an important consideration due to the weight of the tanks) and yet provide a reasonable pressure ratio for the intensifier. A lower limit of 20 bar was chosen. This means that the tanks are
approximately 90% empty (from a simple perfect gas analysis with constant temperature). With a more precise analysis taking into account the compressibility of natural gas, the tanks are actually 92.2% empty when the pressure is allowed to drop to 20 bar. This is a reasonably high value which sets the maximum pressure ratio for the intensifier at 10:1, which is quite manageable.

2.4 Capacity

The capacity of an intensifier is dependent on the speed at which it is operated. It is thus important to determine the operating speed range before any capacity considerations are made.

A convenient power source for the intensifier is the engine crankshaft. The simplest way to drive the intensifier would be to connect directly to the crankshaft and turn it at engine speed. There would be some added complexity if a gearbox was added to reduce or increase the speed, but it is an option. An important point to note is that with this option the intensifier speed is proportional to the engine speed. The addition of a hydraulic or pneumatic system to drive the intensifier would make the intensifier speed independent of engine speed.

The capacity requirement is set by the engine fuel flow demand. The data from Detroit Diesel for the 6V92 lists the maximum required flow rate of natural gas is 53 kg/hr at 2100 RPM at full load. The minimum can be found from experiments on the 6V92 and is less than 2 kg/hr at idle (600 RPM at no load). If the intensifier speed is proportional to engine speed, a more significant value for fuel flow is the amount of fuel required per engine revolution. The maximum flow then occurs at 1200 RPM at full load and is approximately 500 mg per engine revolution and the minimum, at idle, is approximately 50 mg per engine revolution. A detailed calculation of the maximum values can be found in Appendix A while the values at idle were taken from experimental data.
2.5 Capacity Control

Another requirement related to fuel flow is capacity control. The desired situation is one where the intensifier capacity exactly matches the engine fuel demand at any given operating point. The engine fuel demand is a function of engine load and speed. The intensifier mass flow rate is a function of its speed and intake pressure (since delivery pressure is to be kept constant), that is storage tank pressure. Since the intensifier speed is only a function of engine speed and the tank pressure simply decreases with time as fuel is consumed, some type of capacity control system is required to match the intensifier flow rate to the engine requirements.

It would seem simple to design the intensifier for the worst case, that is use the case where the required fuel flow is maximum and the tank pressure is minimum as a design point. Using the worst case as a design point would ensure that the intensifier would never fall short of engine demand. Indeed, the intensifier could supply in the order of 200 times engine demand in a more favorable off-design situation. Appendix B demonstrates this point with a simple single-stage reciprocating machine.

The problem of handling this extra flow is not trivial. If it is fed from the exhaust lines back into the intake lines, the gas expands to intake pressure which results in a considerable waste of energy. If the gas is fed back to the intake lines but is not allowed to expand to intake pressure (through the use of check valves) the result is a very large mass flow rate through the intensifier associated with high gas velocities. These high velocities result in substantial pressure drops across valves which translate into another important loss. Also, high velocities result in accelerated wear of the valves which is undesirable. Thus designing an intensifier for the worst case and trying to control the unneeded flow at other operating points often results in large losses.
Since the engine spends a very small amount of time at maximum fuel flow with low pressure in the tanks, the off-design operating conditions must be taken into account. The need for some type of capacity control to reduce the intensifier capacity in the off-design cases is evident. This requirement of variable capacity will prove to be the most challenging to meet and is the primary area of work in the intensifier project.

2.6 Additional Requirements

Some further requirements that must be met by the intensifier are set by the environment in which the intensifier will be required to function. This environment is determined from the fact that the intensifier would be used to supply compressed natural gas to a vehicle engine.

The power required to drive the intensifier should be as low as possible so losses such as friction or uncontrolled expansion should be avoided. The power requirement should be especially low at low load, when fuel demand is low, since the intensifier should not consume a large percentage of engine output. Compressor efficiency is thus important since the intensifier is drawing power from the vehicle engine.

The intensifier must be placed near the engine in the engine compartment. Since space in the engine compartment is fairly restricted, the intensifier must be as compact as possible. Also, if the need for additional supports for the intensifier is to be avoided, the intensifier weight must be kept low.

The intensifier must be properly sealed as a leak of natural gas would be a potential explosion hazard, especially from a bus left overnight in a garage. If a leak in the system cannot be avoided, it must be properly vented. One option available is to vent any leak to the engine intake. The escaped gas would then be burned in the combustion chamber.
The 6V92 TA is renowned as a very rugged engine. The intensifier, being tucked away in the engine compartment and as accessible as the engine, should be as durable as the engine. Its components should thus be designed with a long life and they should not require frequent replacement.

For the application of natural gas fueling of a bus engine to be attractive, the cost of the entire system must be reasonable. If one considers the tanks, the new gas lines, the injectors, modification to the engine control system, the intensifier and the installation of all these components, it is obvious that the intensifier should not be expensive to build. Simplicity and ease of manufacturing are thus essential. The requirement for low cost also means that the intensifier must take advantage of any existing system (pneumatic, electric, etc.) already on the bus.

2.7 Summary

In summary, the principal design requirements are:

- 200 bar exhaust pressure
- 20 to 200 bar intake pressure
- variable mass flow in the range of 50 to 500 mg/rev
- high compressor efficiency
- low size and weight
- completely leak free
- long life
- low production and maintenance costs

The requirements listed above define the problem of constructing a suitable intensifier. The most important design consideration is to be able to match the intensifier capacity to the
engine fuel requirements and it was shown that this is not trivial. The incorporation of variable
capacity in the intensifier design is what makes this work significant. The next chapter will
examine the different intensifier design concepts and select the intensifier configuration best suited
to meet these requirements.
Chapter 3

Design Concepts

3.1 Introduction

The following chapter is a study of the different intensifier design concepts. The purpose of this review is to determine, in light of the specific design requirements detailed in chapter 2, the intensifier configuration best suited to the design application. This study expands on the work done by Aichinger [2] on this subject.

The principal design alternatives are identified in Fig. 3.1. The design alternatives are divided into two families of compressors: the intermittent flow and continuous flow machines. The intermittent flow family includes the reciprocating and the rotary compressors. The continuous flow include the dynamic compressors or turbomachines. The advantages and disadvantages of the different types will be summarized while discussing the selection of a configuration that will meet the design requirements.
3.2 Turbomachinery

The continuous flow compressors, or turbomachines, will be examined first. The only turbomachine that can operate at a 10:1 pressure ratio with a single stage is the centrifugal compressor. Detailed calculations of the required diameter and angular speed of a typical high pressure ratio centrifugal compressor are described in Appendix C. The results of these calculations are as follows: for the given operating conditions, the required rotor diameter is \(4 \times 10^{-4}\) cm and the angular velocity is \(3 \times 10^9\) RPM. This extremely small rotor would be impossible to machine. Also, because of the small size of the rotor, the clearances would be critical and there would be unacceptable leakage and loss of efficiency. Finally, the required angular velocity is unacceptably high.
Since the angular velocity calculated above is unacceptable, a different approach will be taken. The operating speed will be set at engine speed and the specific speed, a dimensionless parameter commonly used as a criterion for selection of compressor type, will be examined.

The dimensionless parameter specific speed ($N_s$) is defined as:

$$N_s = \frac{\Omega \cdot \sqrt{Q}}{\left(\frac{\Delta p_o}{\rho}\right)^{\frac{3}{4}}}$$

in which $\Omega$ is the angular velocity, $Q$ is the inlet volumetric flow rate, $\Delta p_o$ is the pressure difference and $\rho$ is the inlet density.

As previously mentioned, the angular speed $\Omega$ is chosen to be of the order of the engine speed, that is 2000 RPM or 210 rad/s. The volumetric flow rate $Q$ would be taken to be the maximum flow, 53 kg/hr rate at a density defined by the lowest tank pressure, 20 bar. This density is 13.9 kg/m³ (20 bar, 25°C) and the volumetric flow rate works out to be $1.06 \times 10^{-3}$ m³/s. The pressure difference $\Delta p_o$ is simply the difference between intake and exhaust pressure and is 18 MPa.

The dimensionless specific speed works out to be $1.78 \times 10^{-4}$ and it has been shown that, for best efficiency, the specific speed parameter for different types of dynamic compressors are [3]:

- $N_s > 1.8$ - Axial compressor
- $1.2 > N_s > 2.2$ - Mixed flow pumps
- $0.2 > N_s > 1.2$ - Centrifugal compressor
- $0.1 > N_s$ - Positive displacement
Of course, these values are only guidelines but the value of $N_s$ for the urban bus application is of the order of 0.0002 and is clearly in the range covered by positive displacement or intermittent flow compressors.

### 3.3 Rotary Compressors

The next family of compressors, the intermittent flow machines, can be subdivided into the reciprocating and the rotary compressors. The rotary compressors are usually limited to a 4.5:1 pressure ratio and have an exhaust pressure that falls well short of the required 200 bar. Also, they typically require 25% more power than an equivalent reciprocating machine. This additional power requirement results in a lower efficiency. These shortfalls are due to problem with sealing the sliding surfaces of the rotary compressors [2].

### 3.4 Reciprocating Compressors

The possible design concepts for the application in question has been reduced to reciprocating machinery since both turbomachines and rotary compressors proved to be unable of meeting the design requirements. This category of compressors includes all machines that use a piston sliding in a cylinder to produce the compression. Within this category are options open to the designer in methods of controlling capacity and actuating the piston.

Before capacity control and actuation are examined, the general reciprocating cycle, as well as the concept of volumetric efficiency, will be described.
3.4.1 Compression Cycle

All reciprocating compressors function by the principle of pressure rise due to volume reduction. This volume reduction is supplied by a piston sliding inside a closed cylinder (Fig. 3.2).

![Figure 3.2](image_url)

Simple Reciprocating Compressor

In the isentropic case, that is with no heat transfer or irreversibilities such as friction, the pressure rise is described as: [4]

\[
\left( \frac{P_2}{P_1} \right) = \left( \frac{V_1}{V_2} \right)^k
\]

where \( k \) is the isentropic coefficient (ratio of the specific heats) and the subscript 1 denotes the start of compression and 2 the end of compression. A complete compression cycle will now be described.

The compression cycle begins when the piston is at the lowest point in its reciprocating motion, that is bottom dead center (BDC) (Fig. 3.3). At this point the chamber formed by the cylinder and the piston has its largest volume and it is filled with gas at intake pressure. The
piston starts its upward motion with both valves closed so there is a reduction in volume and thus compression (Fig. 3.4).

When the pressure exceeds the exhaust pressure, that is the pressure that is on the other side of the exhaust valve (which could be the pressure of an accumulator or intercooler for example), the exhaust valve opens and the gas passes out of the cylinder (Fig. 3.5). When the piston reaches top dead center (TDC) or its highest point in its reciprocating motion, there is no longer any reduction in volume so the exhaust valve closes and the gas stops flowing out of the cylinder (Fig. 3.6). The piston then starts its downward travel and any gas that was left in the clearance volume (volume present in the cylinder when the piston is at TDC) begins to expand (Fig. 3.7). The pressure inside the cylinder is described by the same equation as the compression cycle (eq 3.1). The intake valve opens when the pressure of the gas inside the cylinder drops below the intake pressure (Fig. 3.8) and fresh gas enters the cylinder until the piston reaches bottom dead center (BDC) or its lowest point in its reciprocating motion. The intake valve then closes and the cycle is repeated.

Figure 3.3
Beginning of Cycle at BDC

Figure 3.4
Compression Phase
A typical compression cycle can be plotted on a P-V diagram and would look like Fig. 3.9.

Figure 3.9
Typical P-V Diagram for Compression Cycle
The temperature of the gas in the cylinder will also vary through the compression cycle. These variations are described, if the process is isentropic, by the following equation: [4]

\[
\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\frac{k-1}{k}},
\]

in which \( k \) is the isentropic coefficient.

The temperature will thus rise during the compression, drop during the expansion and stay constant during the exhaust and intake phases. Because of these variations and the fact that the temperature of the wall of the cylinder will be fairly constant, there will often be a temperature difference between the gas and the wall and heat transfer either from or to the gas will occur. The assumption of isentropic compression and expansion does not hold and equations 3.1 and 3.2 need to be modified to take into account the heat transfer. This is done by replacing the isentropic coefficient with a polytropic coefficient which is different for compression and expansion. This polytropic coefficient is found in the following way:

The second law of thermodynamics requires: [4]

\[
TdS = dh - vdp
\]

and for an ideal gas, the law for polytropic compression and expansion is

\[
\frac{dp}{p} + n \frac{dv}{v} = 0.
\]

The second law can be rewritten as:

\[
\frac{ds}{R} = \frac{k}{k-1} \left(\frac{dT}{T} - \frac{dp}{p}\right)
\]
\[
\frac{ds}{R} = \frac{k}{k-1} \left( \frac{dp}{p} + \frac{dv}{v} \right) - \frac{dp}{p}
\]

\[
\frac{(k-1) \cdot ds}{R} = k \cdot \frac{dv}{v} + \frac{dp}{p}
\]

\[
\frac{ds}{c_v} = k \cdot \frac{dv}{v} + \frac{dp}{p}
\]

and combining with ideal gas law becomes:

\[
\frac{ds}{c_v} = (k - n) \cdot \frac{dv}{v}.
\]

In this last equation, the only unknowns for a given compression cycle are the polytropic coefficient \( n \) and the entropy variation \( ds \). This equation is thus used to get an idea of the value of \( n \) for different situations. In other words, it is expected that \( n \) be smaller than \( k \) during the compression phase since the gas has a higher temperature than the wall for most of the phase and thus there is a net heat transfer out of the gas and \( ds \) is negative. The opposite holds for the expansion phase, that is \( n \) is expected to be larger than \( k \). The actual values for the \( n \) for compression and expansion are found experimentally. Of course, these values vary depending on the pressure ratio, the intake temperature and the wall temperature.

Mention was made of the clearance volumes which play an important role during the expansion phase of the gas since there would be no expansion phase if there were no clearance volume. In this case, all the gas would be expelled from the cylinder when the piston would be moving towards TDC and as soon as the piston would start its downward motion, the intake
valve would open and admit fresh gas into the cylinder. This seems like a better situation but the presence of clearance volumes at TDC is inevitable. This volume is a result of space in the valves, space left between the top of the piston and the top of the cylinder to provide a safety margin in case of thermal expansion and diametrical clearance between the piston and the cylinder between the top of the piston and the first piston ring.

3.4.2 Volumetric Efficiency

The presence of clearance volumes introduces the concept of volumetric efficiency. This parameter is defined as the ratio of the actual mass flow rate to the theoretical mass flow rate, that is the mass flow if there were no clearance volumes and there were perfect filling of the cylinder just before the piston starts its downward motion. The volumetric efficiency would be: [2]

\[ \eta_v = \frac{\dot{m}}{\rho_{\text{inlet}} \cdot n \cdot V_d} \]

in which \( \dot{m} \) is the actual mass flow, \( \rho_{\text{inlet}} \) is the inlet density, \( n \) is the angular velocity and \( V_d \) is the displaced volume. In terms of clearance volumes, the volumetric efficiency can be expressed as:

\[ \eta_v = 1 - \epsilon_0 \left[ \left( \frac{P_2}{P_1} \right)^{\frac{1}{\gamma}} - 1 \right] \]

where the clearance volume ratio \( \epsilon_0 \) is the ratio of the clearance volume to the displaced volume \( V_d \) and is typically between 4 and 20 \%. 
The pressure ratio and the clearance volumes are the most important factors in determining the volumetric efficiency but there are also other factors that arise when dealing with real machines such as:

- poor filling of the cylinder caused by a pressure drop across the intake valve.
- preheating of the incoming gas by the hot wall causing the gas to expand and reduce the amount of gas that can enter the chamber. This effect has been shown to be able to reduce the volumetric efficiency by as much as 10% for a 5:1 pressure ratio [1].
- gas leakage through the exhaust valve, permitting high pressure gas to enter the cylinder from the exhaust tract during the expansion phase delaying the opening of the intake valve.
- leakage across piston rings and rod seals.

The purpose of keeping the volumetric efficiency high is to keep the machine small. This is attractive since the forces tend to be lower in smaller compressors because there are smaller areas exposed to high pressure and because the masses that need to be accelerated in reciprocating motion are smaller. The use of multistaging provides an important advantage, that of increasing the volumetric efficiency by lowering the pressure ratio of each stage.

The second chapter described the need for the intensifier to have a variable capacity. Different types of capacity control for the reciprocating compressor will now be examined.

### 3.4.3 Capacity Control

One way to control the capacity is to vary the speed at which the intensifier is operating. This can be done continuously or discreetly. Given that the intensifier is going to be powered by a vehicle engine, there are a number of methods of varying the drive speed. These include hydraulic and pneumatic actuation discussed in section 3.4.5. If the intensifier is to be directly driven by the
engine, a continuously variable drive could be possible in principle but successful prototypes have not yet been developed. Alternatively, the intensifier could be run at engine speed when required and disengaged from the engine and thus not running it when the fuel flow does not require it. This is an energy efficient method since no energy is used when the intensifier is not run. The major problem is that this setup would cause variations in the injection pressure though these could be minimized with a large accumulator. Also, this method would require a clutch and control system. This latter option would unfortunately require valuable engine bay space.

An alternative for capacity control is a variable clearance volume arrangement. Since a larger clearance volume would reduce the volumetric efficiency, the mass flow rate would also be reduced. This can be extended to zero mass flow rate when the pressure at the end of the expansion phase is equal to the intake pressure. This option is not as energy efficient as the variable speed option because there is still considerable friction loss when the intensifier is running at zero mass flow. On the other hand, this system is quite easier to implement. Fig. 3.10 and 3.11 show how a simple single stage intensifier could operate with variable clearance volume. One problem associated with this capacity control system is the large pressure loads on the sliding assembly containing the valves.
A third option which seems to combine the advantages of the first two is to vary the stroke of the piston. When high mass flow is required, the piston has a full stroke and acts exactly like a conventional reciprocating compressor. When lower mass flow is needed, the stroke is shortened; this has the same effect as increasing the clearance volume. As opposed to the variable clearance volume option, there are decreasing frictional losses as the stroke is shortened. In fact, if the stroke is reduced to zero, there would no frictional losses due to the piston motion in the cylinder. Fig. 3.12 and 3.13 show the operation of such a variable stroke intensifier.
Varying the stroke of a reciprocating intensifier is difficult if mechanical actuation is considered. The technique used to accomplish this task is the rotary type intensifier which will be described in later sections of this chapter and is the core of the work covered by this thesis.

If the intensifier geometry is not to be variable, that is if the intensifier is a conventional one with constant clearance volume and stroke with an uncontrollable speed, there exist two further techniques to control the capacity. One is to unload the intake valve, that is open the intake valve during the compression phase of the cycle. This causes the gas to flow back into the
intake tract so that it does not get compressed. With this technique the injection pressure would vary unless a large accumulator were used and there is the additional disadvantage of increased complexity and size of the intake valve. This modification to the intake valve usually leads to an increase in clearance volume and thus larger components for a given mass flow rate.

The second method of controlling the capacity of a fixed geometry intensifier is bypassing. In this case, the compressed gas is fed from the exhaust lines back into the intake lines of the intensifier if the mass flow required is less than the mass flow provided by the intensifier. The presence of a check valve prevents the expansion of the gas back down to intake pressure (pressure of the tanks). This system was used in the first prototypes of the intensifier [2] and it was found to consume a lot of power. Theoretically, there should be no power loss from this system (except for frictional losses) but at high intake pressures, the mass flow rate through the bypass system was high enough to cause excessive pressure drop across the check valves.

3.4.5 Actuation

The class of reciprocating compressors can be divided into categories depending on the method of actuation of the piston. Three options were studied in the evaluation: hydraulic, pneumatic and mechanical actuation.

The hydraulic method of actuation uses a double-acting hydraulic cylinder to impart a reciprocating motion to the piston. The system consists of a hydraulic pump driven by the engine, a double-acting hydraulic cylinder, a switch valve that controls the motion of the cylinder and a flow control unit that controls the frequency of the reciprocating motion.
The advantages of this type of system are as follows:

- The capacity of the intensifier can be continuously varied by the hydraulic system and thus the engine fuel requirements can be exactly matched.
- Hydraulic systems have been proven to be reliable and efficient.
- Hydraulic components are interchangeable.

The disadvantages are:

- The system is heavy because of the mass of hydraulic components required.
- The hydraulic system will require a reservoir for the hydraulic fluid and this will require a large amount of space.
- The system is expensive (as shown by Aichinger [2]).

The pneumatic method of actuation is similar to the hydraulic one, the main difference being that the working fluid is air instead of hydraulic fluid. This system seems to have the advantages of the hydraulic driver and seems to eliminate some of the disadvantages since the urban bus already has a pneumatic system on board used mainly for the braking system.

The pneumatic system on the bus can supply air at approximately 110 psi. In order to compress natural gas to 3000 psi, the area exposed to the air must be approximately 30 times larger than the area exposed to the gas. This requirement of a large piston, in addition to the intensifier, results in a large space penalty.

Commercially available gas intensifiers were examined by Aichinger [2]. None could meet the gas mass flow requirement with a typical bus pneumatic system as a source of compressed air. Considering the possibility of upgrading the bus pneumatic system to provide a larger air flow, a gas intensifier manufacturer proposed a system that was composed of two gas intensifiers that would meet the gas mass flow requirements. This intensifier system was found to be very
expensive even before the cost of the necessary modifications to the bus pneumatic system were included. Also, using two gas intensifiers would require a large amount of space due to the requirement of large air pistons.

In short, the hydraulic and the pneumatic actuation systems suffer from the same disadvantages of being bulky, heavy and expensive. On the other hand, they share the advantage of offering a method of continuously controlling the capacity of the intensifier.

The mechanical actuation indicated in Fig. 3.1 encompasses all systems that use a solid link between the prime mover (engine) and the reciprocating motion of the pistons. There are different methods of giving the pistons their reciprocating motion and three were studied in the evaluation: camshaft actuation, crankshaft actuation and rotating piston.

The camshaft actuation would operate in the same manner as the diesel injector, that is that a plunger would be actuated by a lobe on the camshaft (Fig. 3.14).
This option is attractive because it would eliminate the need for a separate intensifier. However, detailed study revealed that the size of the cylinder and piston assembly would be much too large to fit in the space designed for the diesel injector. Also, there is the problem of capacity control. Since the cam lift is constant, the capacity of the camshaft-operated-intensifier is only a function of engine speed and it would be difficult to install a system that would also make it load dependent. The small size of the components would also make machining difficult and expensive. These problems make camshaft actuation unattractive.

Crankshaft actuation would function in the same manner as the pistons in an engine in that the reciprocating motion is provided through a crankshaft and connecting rod assembly. There are two different systems that use crankshaft actuation. The first is direct transmission in which the compressor piston is itself connected to the connecting rod (Fig. 3.15).

This allows for single action compression only. The other type is crosshead transmission (Fig. 3.16). In this configuration, a crosshead is connected to the connecting rod and is guided in a cylinder. The intensifier is then mounted on top of this arrangement and a shaft attached to the crosshead is connected to the compression piston. This arrangement allows for both single or double action compression.

The possibility of converting a small internal combustion engine into an intensifier has been examined but it was found that the forces generated by the compression of the gas would be much larger than the typical forces generated by a combustion engine so extensive modification to the engine would be required.
Figure 3.15
Crankshaft Actuation
Figure 3.16
Crankshaft Actuation with Crosshead
Another possibility of adapting a combustion engine to become an intensifier would be to replace the piston by a crosshead and mount an intensifier above it as shown in Fig. 3.16. The use of a double-acting intensifier greatly reduces the loads on the engine and thus makes this use of the engine safe.

The advantages of using this type of mechanical actuation are:

- The design is simple and inexpensive since it uses existing parts.
- The crosshead design allows for double acting intensifier which results in a small and light package.

The disadvantages of this system are:

- The danger of excessive friction causing wear and heat generation.
- Need for some type of capacity control as the capacity of the intensifier is only dependent on engine speed.

The last type of mechanical actuation is the rotating piston intensifier. This intensifier would function like a hydraulic pump (Fig. 3.17). The reciprocating motion of the pistons is supplied by the fact that the swashblock is not perpendicular to the axis of the shaft to which it is attached. As the angle of the swashblock changes, so does the stroke of the pistons. This allows for the attractive advantage of a variable capacity. The problem with the rotating piston configuration would be with sealing. The fact that the pistons are rotating means that any inlet or outlet for the pistons are also rotating and there has to be a transfer from these rotating components to the stationary lines or vice versa. In the hydraulic pump this problem is handled by having the inlet and outlet ports as slots in a stationary plate at the top of the cylinders. As the pistons and cylinders rotate, the top of the latter line up with these ports at the proper time in the compression cycle. Sealing between the inlet and outlet port is mostly handled by small tolerances
combined with the viscosity of the hydraulic fluid but there is some leakage. The fact that the natural gas would be in gaseous form renders this configuration unattractive as the leakage would be considerable and would lead to very poor intensifier performance.

![Figure 3.17](image)

Hydura PVWH open loop pump. 1-control system, 2-cylinder mounted journal bearing, 3-swashblock lubrication, 4-shaft, 5-shaft bearing, 6-swashblock with bearing, 7-plunger bearings, 8-valve plate, 9-valve plate port, 10-through shaft, 11-quiet valve plate design, 12-frame

Although this design does not seem capable of being a viable solution to the intensifier problem, it does demonstrate the use of the variable stroke method of capacity control.

Of the three mechanical actuation configurations, only the crankshaft actuation seems to be able to meet the requirements for the intensifier. Its advantages are:

- simple design
- small and lightweight package
- inexpensive system
- high efficiency operation.
Its disadvantages are:

- external flow control system required in order to have a variable capacity
- possible problems with excessive wear and heat generation due to friction.

In summary, Table 3.1 provides the relative advantages and disadvantages of the three types of actuation:

<table>
<thead>
<tr>
<th>Design Parameter</th>
<th>Hydraulic Drive</th>
<th>Pneumatic Drive</th>
<th>Mechanical Drive</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency</td>
<td>80-90%</td>
<td>80-90%</td>
<td>90-97%</td>
</tr>
<tr>
<td>Flow Control</td>
<td>variable</td>
<td>variable</td>
<td>control system</td>
</tr>
<tr>
<td></td>
<td>displacement</td>
<td>displacement</td>
<td>required</td>
</tr>
<tr>
<td>System Complexity</td>
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<td>complex</td>
<td>simple</td>
</tr>
<tr>
<td>Cost</td>
<td>high</td>
<td>high</td>
<td>low</td>
</tr>
<tr>
<td>Durability</td>
<td>high</td>
<td>high</td>
<td>low</td>
</tr>
<tr>
<td>Weight and Space</td>
<td>large</td>
<td>large</td>
<td>small</td>
</tr>
<tr>
<td>Maintenance Work</td>
<td>low</td>
<td>low</td>
<td>high</td>
</tr>
<tr>
<td>Noise</td>
<td>low</td>
<td>high</td>
<td>low</td>
</tr>
</tbody>
</table>

Table 3.1
Comparison of alternative drive systems (C. Aichinger [2])

Hydraulic and pneumatic actuation configurations both have the attractive advantage of supplying the intensifier with variable flow but both suffer from the fact that the systems would be large, heavy and complex. The mechanical actuation would operate at a higher efficiency and would be lighter, smaller and simpler but would require the implementation of one of the capacity control techniques discussed earlier. The following section will describe the intensifier design of Aichinger, that is the single-stage and the two-stage reciprocating designs.
3.5 Previous Intensifier Design

The first prototype built by Aichinger was a single-stage, double-acting intensifier which could produce the maximum required mass flow and pressure ratio. This intensifier was mechanically actuated using the crosshead arrangement shown in Fig. 3.16. Tests of this configuration showed that the temperature resulting from a 10:1 pressure ratio led to problems with the sealing at the valves as the materials chosen could not sustain these temperatures. Other problems were that given the high pressure loading of the piston, maintaining alignment was extremely critical and side loads on the crosshead were severe. Because of this, this prototype was not tested at speeds higher than 200 RPM. Also, the valve losses were high resulting in a low volumetric efficiency.

The second prototype, the double acting, two stage intensifier, was found to solve many of the problems encountered with the first design. The two stages meant that intercooling was possible and this drastically reduced the operating temperature and improved the performance of temperature-sensitive devices such as valves and seals. The valve design was greatly improved and this resulted in a much higher volumetric efficiency. The problems with alignment were still present though and this prevented the testing at high speeds (more than 200 RPM).

Both prototypes had problems when it came to capacity control. Bypassing, where the exhaust gas is fed into the intake track, was found to produce high gas velocities through the valve and this resulted in a back pressure and a loss of efficiency. Valve unloading, in which the intake valve is forced opened during the compression part of the cycle, was found to be difficult to implement because of the added complexity and the increase in clearance volume (accompanied by the decrease in volumetric efficiency).
The previous designs thus did not meet the requirement for a variable capacity but did establish a good foundation of test experience on which to base a new design.

3.6 Choice of Intensifier Configuration

From the design requirements and the study of alternative design concepts, the first decision concerning intensifier configuration was to choose a reciprocating machine. From the previous sections on volumetric efficiency, capacity control and actuation, the following can be concluded:

1. Multi-staging has beneficial effects on volumetric efficiency and thus tends to reduce intensifier size and the pressures forces felt by intensifier components.
2. Varying the stroke of the intensifier is an energy efficient technique to control the capacity of the intensifier.
3. Mechanical actuation is a space- and cost-efficient method of transferring energy from the engine to the intensifier.

With these facts in mind, and with knowledge gained from previous intensifier designs, the following intensifier configuration was chosen: a mechanically actuated, multi-stage, variable stroke reciprocating intensifier.

The number of stages will be determined in the design optimization described in chapter 4. In the previous sections, the advantages of variable stroke operation were explained but the mechanism used to vary the stroke was not described. In the following section, a new design concept for achieving variable stroke operation in a compact mechanism is described.
3.7 Rotary-Reciprocating Intensifier Terminology

Fig. 3.18 and 3.19 are simplified schematic of the rotary intensifier.

Figure 3.18
End View of Rotary-Reciprocating Intensifier
The basic working element of the rotary intensifier is a piston (1) sliding inside a cylinder (2). As seen in the compression cycle, each cylinder has an intake (3) and an exhaust valve (4). In the configuration shown, there are four cylinders, two of which act as a first stage (5), the others as second stage (6). Between these two stage, there are intercooling cavities (7). The connecting rod (8) has at one end the piston head (1) and at the other the rollers (9). The
connecting rod is guided into a strictly linear motion by a linear bearing (10). All the components listed above are referred to as the rotor as it this assembly that spun by the engine.

Attached to the rotor and rotating with it are two shafts. The first is the drive shaft (11) through which power is transmitted from the power sources to the rotor. Also on this shaft is the oil seal (12) through which oil enters the rotor. The second shaft is the gas shaft (13). On this shaft there are gas seals (14) through which the intake gas enters the rotor and the exhaust exits. The gas and oil seals are held in place by the frame (15) which also houses two bearings (16) which hold the rotor in place. Finally, the frame guides the motion of the outer ring (17) whose position is controlled by the eccentricity control mechanism (18).

Those are the main components of the rotary intensifier. Their functioning will be explained in more detail in the following section.

3.8 Rotary-Reciprocating Intensifier Operation

The easiest way to describe the operation of the intensifier is to follow a particle of gas as it goes through the intensifier and gets compressed. The particle first enters the intensifier at the intake gas seal. These seals are simple O-ring type mechanical seals which seal between the stationary intake gas lines and the rotating gas shaft. The gas then enters the shaft through two holes and follows a path through the shaft into the rotor, and more specifically, the first stage intake valves. These valves admit gas to the first stage cylinders during the intake phase of the compression cycle. It is worthy of note that there are two paths that the gas could have taken since there are two first stage cylinders, one opposite the other in the rotor.

Once compressed by the first-stage piston, the gas leaves the first stage through the exhaust valves into the intercooling cavities. After being cooled, the gas enters the second stage
cylinders and the pressure of the gas is further increased. Finally, the gas exits the second stage and follows a path through the gas shaft to the exhaust gas seals, which function in the same manner as the intake gas seals.

So far, apart from the fact that the piston and cylinders are rotating, nothing has been described that makes the rotary intensifier different from a conventional reciprocating compressor. What sets this design apart is its variable stroke. The outer ring can be moved in such a way that its center does not coincide with the axis of rotation of the rotor. The distance between the axis of rotation and the outer ring center is referred to as the eccentricity.

When the eccentricity is set at zero, in other words the axis of rotation of the rotor and the center of the outer ring are superimposed, the pistons do not have any reciprocating motion, they simply stay at the mid point of the cylinders and do not compress the gas at all (fig 3.20).

Figure 3.20
Intensifier with Zero Eccentricity
On the other hand, when the eccentricity is set at half of the maximum stroke, the pistons have the largest reciprocating motion and offer maximum compression to the gas (fig 3.21). Of course, the eccentricity can be set at any point between these two extremes and thus the amount of compression done by the intensifier is controllable.

Figure 3.21
Intensifier with Maximum Eccentricity

It should be noted that although the rotary intensifier was always shown with two stages and four cylinders, it is not the only possible configuration. In fact, there are many variables that must be considered in order to obtain the final intensifier design. The next chapter will examine the design process and show how the final design was chosen.
4.1 Summary of Design Requirements

The principal design requirements for the intensifier identified in the second chapter are:

- 200 bar exhaust pressure
- 20 to 200 bar intake pressure
- variable mass flow in the range of 50 to 500 mg/rev
- low size and weight
- completely leak free
- long life
- low production and maintenance costs

The prototype has additional requirements that come into consideration because only one unit will be produced. The first factor that affected the design of the prototype was speed of construction. Since only one unit will be produced, it is not time efficient nor economical to have special tooling developed for the machining of the prototype. Also, techniques such as molding, stamping or casting were avoided since it would not be economical to use these techniques for the prototype even though they may prove to be a wise choice for the production model. This often resulted in components that were heavier or bulkier than required. This fact contradicts the
requirements of low size and weight but for the production model, more elegantly machined components would meet this requirement.

The prototype was designed with easy assembly and disassembly in mind. Most of the pieces are bolted together and there is minimal welding of components. Also, the foreseen problem areas were made easy to access without having to disassemble the rest of the intensifier.

The safety factor also had to be considered in the design of the prototype. For this reason, some components, such as the outer ring, the frame plates and the main bearing housings, were deliberately over designed.

It was deemed economical to use the same test rig, that is the same electric motor and mounting frame, as the intensifier versions 2.1 and 1.2 and this affected the design. The overall size, the mounting brackets and the position of the drive shaft were strongly affected by this consideration.

In summary, the additional design requirements for the prototype version are:

- speed of machining
- easy assembly and disassembly
- high safety factor
- use of existing test rig

These are the requirements that the intensifier prototype had to meet. The techniques used to incorporate these requirements into the design process will be examined in later sections. The design process itself will now be described and the terminology that will be used throughout this chapter will be defined.
4.2 Description of Design Procedure

The number, size and position of the intensifier components could not be chosen arbitrarily since this would yield an intensifier which either does not meet the design requirements or does so with waste of resources. The ultimate goal of the design process is to produce an intensifier that meets all the requirements with minimum use of resources. The optimization technique will be described by examining the various elements of the process.

The system parameters are the variables that are dictated by the operating conditions. These parameters cannot be changed in the optimization process. For example, the intake and exhaust pressure of the intensifier are system parameters since they do not vary with changes in the design.

The design variables are the variables that describe the number, size and relative position of the various components of the intensifier. The design variables are the ones that are varied in an attempt to reach an optimum design. For example, the bore and stroke of a given compression stage are design variables.

The design constraints limit the range of the design variables. These constraints originate from the design requirements, from specific component limitations or from practical considerations. The required mass flow rate is an example of a design constraint originating from the requirements. A maximum rolling velocity of a bearing is a constraint that arises from a component limitation and a minimum size for a given piston would be a practical constraint.

In order to evaluate the optimality of given values of the design variables, an objective function must be defined. The objective function is a statement of the general goal of the optimization process. In the simplest case, trying to find the lightest intensifier design could be an objective function. This function is usually in the form of a mathematical equation that links the
design variables to the quantity described in the objective function (such as weight for this example).

In most design problem, there is not a single quantity to be maximized or minimized while satisfying the constraints. Trying to find the lightest intensifier with maximum efficiency would be an example of an objective function that contains two quantities. Obviously, it is highly unlikely that both these conditions are met simultaneously at a design point. For this reason, compromises must be made when such a function is used. The terms minimum and maximum will still be used but they will not have the absolute definition they usually have. Instead, the term minimum will be defined as “the lowest possible value for a quantity consistent with reasonable values for the other quantities” and similarly for the term maximum.

The quantities that form the objective function are often taken from non specific design requirements such as low size and weight or long life. The four elements of the optimization process defined above will now be examined in greater depth.

4.3 System Parameters

Some of the requirements can be transformed into system parameters. These are quantities that do not vary and are set by the operating conditions. The system parameters for the case of the intensifier design are listed below:

- exhaust pressure (200 bar)
- minimum intake pressure (20 bar)
- maximum required flow rate (500 mg/rev)
- operating speed (engine speed)
- natural gas properties
aluminum and steel properties

It is to be noted that system parameters such as intake pressure and mass flow rate actually vary during the operation of the intensifier but it is the extreme cases listed above that will be used during the design, the off-design cases being handled by the variable stroke. The reason that the extreme case of minimum intake pressure and maximum flow rate was chosen as a design point is that it is at this point where the intensifier requires the most power and thus the highest torque. Furthermore, the intensifier is a full eccentricity at this point which results in maximum contact angle between the roller bearings and the outer ring surface. This maximum contact angle results in maximum side loads on the piston connecting rod which makes the linear bearing constraints important.

From the point of roller bearing loads, the worst case occurs at the highest intake pressure, 200 bar. This point was not used as the design point since the eccentricity would be zero at this point which results in zero side loads on the piston connecting rod. The high roller bearing loads at this point were taken into account by verifying that a given bearing size could sustain these maximum loads.

4.4 Objective Function

The objective function for the intensifier has four main quantities that are taken from the non-specific design requirements. Since there are more than one quantity in the objective function, the terms minimum and maximum will be used with the definition given in section 4.2 in mind.

The first quantity in the objective function is size which will be minimized. This requirement stems from the fact that the smallest intensifier that meets all the other requirements
will be the most attractive since the eventual production model will conceivably have to function in the restrictive space of an engine bay.

The design also seeks a minimum weight intensifier. This part of the objective function greatly affects the choice of materials used and usually works in concert with the low size requirement.

The third quantity, that of cost, will also be minimized. The cost of the intensifier includes cost of the material, machining cost, which is related to machining time, and use of existing resources such as the test rig.

The last quantity in the objective function is one that will be maximized and is the life of the intensifier. This often translates into having a high safety factor for components such as bearings since these components have a life that decreases as they are loaded nearer to their maximum load.

As previously mentioned, it is impossible to minimize or maximize four quantities simultaneously. In principle, the four quantities would be assigned a weighting factor and combined into one equation. In practice this was not done because it was difficult to determine the relative importance of the four quantities before starting to design. Also, the relative importance of the quantities varies depending on the situation. In fact the four elements of the objective function do not always work in harmony. For example, trying to achieve long life in a bearing results in opting for a larger bearing which contradicts the minimization of size, weight and cost. In these cases, a decision as to the relative importance of the various elements of the objective function must be made. This decision is greatly affected by the situation. For example, in the case of the design of a bearing bracket which is outside of the working components of the intensifier, the minimum machining time (minimum cost) consideration would be the dominant
deciding factor and low size and weight would be temporarily ignored. On the other hand, the
design of a piston will definitely emphasize low weight since the piston affects many other
components in the intensifier.

The objective function then permits the designer to define the optimality of a particular
configuration. This configuration is determined by the value of the design variables which will
now be described.

4.5 Design Variables

The design variables are the variables which have to be determined. They range from
parameters such as number of compression stages to the sizes of individual parts. The design
variables will now be listed and the considerations that come into effect when determining their
values will be discussed in the following sections.

The design variables that have the greatest effect on the final configuration are:

- number of stages
- number of pistons per stage
- stroke of the pistons
- bore of the pistons
- rod diameter
- linear bearing length
- roller bearing length and diameter
- outer ring diameter

These design variables are interdependent, that is that a change in one of these variables
requires a change in several others. For this reason, they have to be determined simultaneously.
Decisions as to the sizing of components such as frame supports or bolts are made after these variables are determined.

4.5.1 Number of Stages

The compression of a gas through a 10:1 pressure ratio could conceivably be handled by a single cylinder though earlier experiments by Aichinger [2] showed that this intensifier configuration led to excessively high operating temperatures. The advantage of multistaging is that it adds the possibility of cooling the gas between the stages, which reduces the maximum operating temperatures and compression work. This has the effect of increasing the life of most components such as seals and bushings. The disadvantage of multistaging is the added complexity and space required by the intensifier.

4.5.2 Number of Pistons

Each compression stage can consist of a single or multiple pistons. Increasing the number of pistons results in a higher complexity and production cost but can have many advantages. One advantage is that if one of the components of a given piston were to fail, the intensifier could still function, albeit at a reduced capacity. In the vehicle application, it could mean that the vehicle would be able to reach a service station where the intensifier could be repaired. Another advantage which is particular to the rotary intensifier is ease of balancing the rotor. If pistons of a same stage are placed in an opposing manner, then there are greatly reduced forces acting on the shaft of the rotor. This can result in smaller components and reduced weight. Finally, a larger number of pistons reduces the pressure forces felt by each individual piston so components such as the roller bearing and the linear bearing are not as severely loaded.
4.5.3 Stroke

From preliminary design experience gained with the simulation program, the stroke of the pistons was found to be a key design variable. Because of the rotary configuration, the stroke of all the pistons is the same. In the design phase, the term stroke is used to refer to the maximum stroke of the intensifier; the fact that it varies to fulfill the variable capacity requirement is of no concern since most of the design centers around the worst case, that is maximum pressure ratio and maximum capacity.

The stroke is key because it affects many other design variables. For example a reduction in stroke requires an increase in piston bore in order to maintain the same maximum flow rate. The larger bores result in larger pressure forces acting on the pistons. On the other hand, an increase in stroke has the effect of moving many components away from the center of rotation and thus increasing centrifugal forces. It is desirable to find the point where the combination of pressure and centrifugal forces is minimum since the sizing of the rod and the roller bearings depend on these forces. For this reason, a large part of the optimization process revolves around the determination of the stroke.

4.5.4 Bore

The bore of the pistons is closely related to the stroke and the number of pistons through the requirement of capacity. The most important effect of the bore is that pressure forces increase with an increase in bore. Also, large bores require a large rotor and leave less space for the intercoolers. Large bores are also unattractive because they increase the clearance volume that is due to the space left between the top of the piston and the head of the cylinder. There is a
minimum size for the bores which comes from machining considerations and availability of piston
seals.

4.5.5 Rod Diameter

The rod diameter has two factors determining its value. The load carrying capacity of the
linear bearing that guides the rod depends on its length and the diameter of the rod. A larger rod
permits larger side loads, but has the negative effect of generating larger centrifugal forces.

The second factor that affects rod diameter is the possibility of pressurizing the back side
of the piston with interstage pressure. This has the beneficial effect of counteracting the pressure
and centrifugal forces. Since the back side pressure acts on an area equal to the area of the piston
minus the area of the rod, the beneficial effect is reduced with a larger rod. Also, this back side
pressurization needs a rod seal which reduces the length of the rod available to the linear bearing
or increases the size of the rotor.

4.5.6 Bearing Diameter and Length

The size of the bearings is mostly determined by the pressure and centrifugal forces. The
problem is that as the bearings become larger to accommodate larger forces, they generate larger
centrifugal forces themselves. The mass of the bearings is an important factor since they are the
components that are situated the farthest from the center of rotation. Also, larger bearings have a
lower allowable rotating speed. For these reasons, a balance between the load carrying ability and
the mass and the allowable speed of the bearing must be found by studying the effect of varying
both diameter and length.
4.5.7 Outer Ring Diameter

The negative effect of large centrifugal forces has been noted in the preceding discussion of the design variables. Furthermore, a large outer ring results in a higher cost for the materials and the machining. For these reasons, the outer ring diameter should be kept at a minimum. Factors that tend to increase the outer ring diameter are: longer stroke, longer required length of the linear bearing, larger diameter bearings and more required space in the rotor to accommodate large amounts of pistons.

It was shown that the determination of the design variables is not trivial. The design variables described above are all interdependent and it is not always clear if the changing of one will have a positive or negative overall effect. The last element of the optimization process, the design constraints, will now be examined to see what limits are placed on the design variables.

4.6 Design Constraints

The values for the design variables cannot be chosen arbitrarily. Not only do the values of the design variable affect the optimality of the design, they are also bound by the design constraints. These constraints are linked to the requirements or originate from limitations of particular components. There are also many practical constraints such as the minimum and maximum sizes for the components but these do not need to be elaborated in detail. The five constraints that most affected the design will now be discussed.

4.6.1 Roller Bearing Load and Speed

The first two constraints have to do with the roller bearings that roll on the inside of the outer ring and are connected to the piston through the rod. These bearing are loaded by the
pressure forces acting on the piston head which are transmitted down the rod. Also, these bearings must support the centrifugal forces generated by the rotating piston, rod and bearings themselves.

The maximum loads that these bearings can support depend on the bearing type, diameter and length. Since preliminary calculations had shown that these bearings would be heavily loaded, the type of bearing chosen was the one that would be best suited to high loads. After consultation with INA (a bearing manufacturer) engineers, the type chosen was the unit cage needle bearing. These bearings consist of a cage holding the rolling elements in place. The races for the bearing are supplied by the shaft and casing that support them, both of which must be ground and hardened. Since other types of bearing have races made of drawn steel (which is the weakest link of the bearing), it is this lack of built-in races that make this bearing type suitable for high loads. Also, the unit cages are small compared to full bearings which is an advantage because a small bearing results in low centrifugal forces.

The maximum load still depends on diameter and length of the bearing and it would slow the design process considerably if commercial bearing specifications had to be consulted at every design iteration to verify if the bearings are overloaded. After examining commercially available bearing specifications [5,6], it was found that the load bearing capacity of the unit cages is approximately proportional to both diameter and length. This can be seen in figure 4.1 and 4.2. For a first approximation, the load bearing capacity was reduced to two linear equations which can be seen on the graph which were used to accelerate the design process. The square points on the graphs are data points taken from the bearing specifications and the straight lines are the linear approximations used in the design process.
The second design constraint that is generated by the roller bearings has to do with their maximum allowable speed. The rollers will be kept as small as possible to reduce the centrifugal
forces and the outer ring has a diameter that is an order of magnitude larger than that of the rollers. These two factors result in considerable angular speed for the bearing (of the order of 10-15 times engine speed); the maximum allowable speed depends on the type of bearing and its diameter. Since the unit cages were also well suited to high speeds because of their relatively simple construction, there were no reasons to search for a different type of bearing. After examination of the commercially available bearing specifications [5,6], it was found that the allowable speed decreased approximately linearly with the diameter of the bearing (figure 4.3). As with the case of allowable load, the allowable speed was reduced to a linear equation.

\[
L_a = \left( \frac{16666}{n} \right) \left( \frac{C^{10}}{F} \right) \]

The life of the bearing is affected by both load and speed as can be seen from this formula [5]:

Figure 4.3
Linearization of Commercial Bearing Data [5,6]
Where $L_b$ is the life of the bearing in hours, $n$ is the speed in RPM, $F$ is the load in N and $C$ is the basic load rating of a given bearing type and size (listed in INA catalogues [5,6]). A C/F ratio of 6 is the minimum recommended. It can be seen that if the bearing is close to the maximum load (C/F=6) and the speed is of the order of 20000 RPM, the bearing would have a life of approximately 300 hours. This is very short compared to the life of a typical diesel engine. For this reason, it is important to try to remain as far as possible from the maximum load and speed cases. In Fig. 4.10 and 4.11, a C/F ratio of 6 is used to determine the maximum allowable load.

These two constraints are satisfied when the following guidelines are observed: the forces acting along the shaft of the rod must be as small as possible, the outer ring diameter must be as small as possible to reduce centrifugal forces and reduce the angular velocity of the roller bearings.

4.6.2 Linear Bearing Load and Sliding Velocity

The third and fourth design constraints are also supplied by a component limitation. This component is the linear bearing that guides the rod. This linear bearing is loaded by the side loads generated by the fact that the roller bearings are not rolling on a surface that is perpendicular to the rod axis. These side loads must be supported by a linear bearing because the piston rings would not be able to support them.

The first design constraint provided by the linear bearing is a sliding velocity constraint. Most linear bearings have an upper limit on the speed that the rod they are guiding can slide past them. After some preliminary calculations, it was found that the Teflon bushing by Permaglide [7] would be suitable. This bushing has been used in the previous intensifier design with satisfactory
results. It also offered a load carrying capacity comparable to more complex linear bearing in a very small package.

This type of bushing has three types of limitations. The first is a maximum value of the sliding velocity \(v\). The second is a maximum value of the ratio of the force and the projected area of the bushing (length times diameter) which is referred to as \(p\). Finally, the last limitation is a maximum value of the product of the first two quantities \((pv)\). After preliminary examination of the bushings, the force-area ratio \((p)\) was found to be always well within the allowable range. For this reason, only the other two limitations will be considered in the analysis [7].

These bushings have a maximum sliding velocity of 3 m/s. Since the rod is traveling twice the stroke every revolution, it can be found that this constraint limits the stroke at approximately 43 mm. This was found with an intensifier speed of 2100 RPM and averaging the piston speed over the stroke. In fact, this stroke would generate a peak velocity higher that the allowable velocity of the bushing so the stroke is limited at 40 mm.

The second design constraint provided by the bushing is a combination of load per unit area and sliding velocity. The Permaglide bushing has an allowable \(pv\) value of 3 \(\text{N/mm}^2\)(m/s). This constraint affects the size of the bushing. It is advantageous to have a small bushing because this results in a shorter rod, a smaller outer ring diameter and thus lower centrifugal loads.

These two constraints are best satisfied in the following cases: when the stroke is as short as possible and when the forces acting along the rod (which translate into side loads on the bushing) are as low as possible.
4.6.3 Required Mass Flow

The required capacity constraint comes directly from the design requirements. It is an important constraint as it sets the required displaced volume of the various compression stages. The method used to determine the required displaced volume was to use the computer program to run different configurations of intensifier.

4.7 Design Procedure

All the elements of the optimization process having been identified, the following sections will describe the determination the design variables which will satisfy all the design constraints and yield the best objective function. First the tools used in the analysis will be examined.

4.7.1 Design Tools

The task of determining the design variables was done with two main tools. One was the spreadsheet program Lotus 123. Using this program permitted convenient graphical examination of the effects of varying the design variables. Some choices were simple to make simply by pinpointing the minimum on a graph. The use of this software sometimes required some approximations which will be discussed as they are used in the analysis. The insight gained by the use of the second design tool, the simulation program, was invaluable in helping to make these approximations as realistic as possible.

The simulation program is a Quick Basic program that simulates the functioning of a rotary intensifier and can be found in Appendix I. The program calculates the pressures and temperatures in all the stages and the intercoolers at each degree of crank shaft rotation (\( \theta \)). With
this information and a knowledge of how the check valves function, the mass transfers through the stages can be calculated.

The simulation program used the following approximations. To find the pressures in the cylinders, polytropic compression and expansion are used:

\[
\frac{P_{i+1}}{P_i} = \left( \frac{V_i}{V_{i+1}} \right)^n
\]

where the subscript \( i \) denotes a given degree of rotor rotation and the volume \( V \) is determined from geometry for each degree of rotor rotation. The polytropic coefficients used were typical coefficients found in the previous intensifier design. The values used are \( n = 1.25 \) for compression and \( n = 1.4 \) for expansion. This is consistent with the discussion of polytropic coefficients in section 3.4.1. Of course, these coefficients vary with pressure ratio and temperatures, but it was believed that this method would yield more realistic results that the assumption of isentropic compression and expansion. To find the temperatures from the pressures, the ideal gas relationship was used.

The simulation program functions as follows. The first step is to set all the intensifier parameters such as bores, stroke and intercooler volume. The pressures and temperatures throughout the intensifier are then set at intake conditions. This simulates the actual start-up of the real intensifier. The volumes of both stages for each angle \( \theta \) are then calculated and placed in an array. The program then starts a loop which goes through a complete revolution one degree at a time.

From the volume at angles \( \theta \) and \( \theta + 1 \), and the initial pressures, the pressures in both stages are calculated from polytropic compression and expansion for every angle \( \theta \). The way the
check valves are handled is that the pressures on both sides of the valve are compared and when there is a pressure difference in the correct direction, the valve opens and permits gas to be transferred. The pressure in the interstage is calculated from its volume, temperature and the inflow from the first stage and outflow into the second stage. The temperatures are calculated from the ideal gas law in the stages.

The program also has a simple heat transfer model. This model accounts for the heating and cooling of the gas by the rotor. A bulk temperature for the rotor is set and the temperatures in the stages and the intercooler are modified using the following equation:

\[ T = C(T_{blk} - T) + T \]

in which \( T \) is the temperature which is modified (a stage or interstage temperature), \( C \) is a constant and \( T_{blk} \) is the chosen bulk temperature. \( T_{blk} \) was set at 80°C which was a typical bulk temperature of the previous intensifier versions and \( C \) was chosen so that the operating temperature of the intensifier were similar to the results obtained by Aichinger [2].

At the end of a revolution, the mass flow through the first and second stages are compared. Steady state is said to occur when these mass flows are the same. If they are different, the program goes through another revolution. Since mass flow through the stages is mostly dependent on interstage pressure, this definition of steady state means that interstage pressure has also reached a steady state value. This technique proved to yield very good results as this predicted value of interstage pressure matched the measured value almost exactly.

The simulation program also calculates the forces felt by the roller bearings at every angle \( \theta \). This is done from knowledge of the pressure in the cylinder and the centrifugal forces generated by the components such as the roller bearings and connecting rod. With a knowledge
of the contact angle between the roller bearing and the outer ring (which can be found from geometry), the torque felt by the rotor can be calculated. This torque was used to calculate the power required to run the intensifier.

The principal use of the program was to make possible the simulation of many different intensifier configurations. The program also gave the design an understanding of the interdependence of the design variables without having to build any prototypes. This was useful in the preliminary design stage when a design strategy was formulated. The process of arriving at an optimal design will now be described.

4.7.2 Number of Stages and Cylinders

In this section, two design variables will be determined: the number of stages and the total number of cylinders. The main design constraints that come into effect when determining these variables are practical ones: there are no particular components which limit the choices.

It is advantageous to have more than one compression stage for two reasons. First, multistaging decreases the pressure ratio across a particular stage which has a beneficial effect on the volumetric efficiency of that stage. Second, multistaging permits the use of intercoolers which cool the gas between the compression stages. This results in lower operating temperatures and lower work of compression. The lower operating temperatures increase the life of components such as seals and valves. The disadvantage of having more than one compression stage is the added complexity.

The number of cylinders depends on the number of stages and the number of cylinders per stage. There are two advantages of having more than one cylinder for a given stage. First, the required displaced volume of a given cylinder is smaller which results in smaller areas exposed to
the pressure. The smaller pressure forces result in smaller components, lower centrifugal forces and often longer predicted life of the components. Second, for the special case of the rotary intensifier, the cylinders of a given stage can be arranged in an opposing manner which tend to balance the forces acting on the shafts of the rotor. Also, these opposing forces tend to smooth out the torque curve of the intensifier. Again, the main disadvantage of having more than one cylinder per stage is the added complexity. Also, if there are too many cylinders in the rotor, there is little room left for the intercooling cavities which reduces their heat dissipation capability and one advantage of multistaging is lost.

The compromise which was found between the advantages of multistaging and cylinders and the disadvantage of the added complexity was the following: two stages of two cylinders each (figure 4.4).

![Figure 4.4](image)

Figure 4.4
Stage and Piston Configuration

The four cylinders were arranged in a cross configuration with the cylinder of a given stage opposing each other. The use of two compression stages has been proven effective in the
intensifier version 2.1 which had operating temperatures which were quite manageable. Also, as will be shown in the next section, this choice for the number of stages and cylinders resulted in cylinder which were of a size that did not present any machining difficulties. This configuration also left plenty of room for the intercooling cavities.

4.7.3 Design of the First and Second Stage

This section describes the core of the design work. Sizing the various parts of the two stages proved to be an iterative process. The first step in this process was to simply find a design point, that is a consistent set of values for the design variables that did not violate any of the design constraints. Working with the computer simulation and spreadsheets, investigation of the effects of varying the design variables around this point proved to be very valuable to obtain an understanding of the interaction between the design variables, the quantities of the objective function and the constraints.

In this section the determination of seven design variables will be discussed:

- stroke
- bore of the first stage
- bore of the second stage
- rod diameter
- linear bearing length
- roller bearing diameter
- roller bearing length
- outer ring diameter
The design constraints that will come into play are the linear bearing speed and load limits, the roller bearing speed and load limits, the required mass flow and practical constraints such as the minimum machinable size of a piston. The list of known variables, unknown variables, equations and information extracted from the computer simulation will now be developed.

The geometric variables are shown in figure 4.5 below, except for the length of the roller bearing $L_b$ since it is in the perpendicular direction.

The variables used in the design process are classified in Table 4.1 as known, unknown and limited by the design constraints.
## Variables

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<thead>
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<th>Known</th>
<th>Unknown</th>
<th>Limited by Constraints</th>
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<tbody>
<tr>
<td>Sym</td>
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<td>Sym</td>
</tr>
<tr>
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<td>intake pressure</td>
<td>$S$</td>
</tr>
<tr>
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<td>intake temperature</td>
<td>$B_1$</td>
</tr>
<tr>
<td>$\rho_1$</td>
<td>intake density</td>
<td>$B_2$</td>
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<td>$P_3$</td>
<td>exhaust pressure</td>
<td>$D_t$</td>
</tr>
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<td>$\omega_e$</td>
<td>angular velocity of the rotor</td>
<td>$L_r$</td>
</tr>
<tr>
<td>$\epsilon_1$</td>
<td>1st stage clearance ratio</td>
<td>$D_b$</td>
</tr>
<tr>
<td>$\epsilon_2$</td>
<td>2nd stage clearance ratio</td>
<td>$L_b$</td>
</tr>
<tr>
<td>$C_1$</td>
<td>distance from top of cylinder to center of rotation</td>
<td>$D_o$</td>
</tr>
<tr>
<td>$C_2$</td>
<td>length of shaft seal</td>
<td>$L_{4b}$</td>
</tr>
<tr>
<td>$n$</td>
<td>isentropic coefficient</td>
<td>$\eta_{V2}$</td>
</tr>
<tr>
<td>$R$</td>
<td>gas constant for NG</td>
<td>$\rho_a$</td>
</tr>
<tr>
<td>$\rho_a$</td>
<td>density of aluminum</td>
<td>$\rho_2$</td>
</tr>
<tr>
<td>$T_{blk}$</td>
<td>bulk temperature of rotor</td>
<td>$T_2$</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>contact angle</td>
<td></td>
</tr>
</tbody>
</table>

### Table 4.1
Design Variables
The governing equations for the design process are:

Mass flow through the first stage

\[ \dot{m}_1 = \eta_{\nu 1} \cdot \rho_1 \cdot \frac{\pi}{4} \cdot B_1^2 \cdot S \]  \hspace{1cm} 4.1

Volumetric efficiency of the first stage (from section 3.4.2)

\[ \eta_{\nu 1} = 1 - \varepsilon \left[ \left( \frac{P_2}{P_1} \right)^{\gamma} - 1 \right] \]  \hspace{1cm} 4.2

Mass flow through the second stage

\[ \dot{m}_2 = \eta_{\nu 2} \cdot \rho_2 \cdot \frac{\pi}{4} \cdot B_2^2 \cdot S \]  \hspace{1cm} 4.3

Interstage density

\[ \rho_2 = \frac{P_2}{R \cdot T_2} \]  \hspace{1cm} 4.4

Volumetric efficiency of the second stage (from section 3.4.2)

\[ \eta_{\nu 2} = 1 - \varepsilon \left[ \left( \frac{P_3}{P_2} \right)^{\gamma} - 1 \right] \]  \hspace{1cm} 4.5
Forces on the first stage roller bearing.

\[ F_{b1} = P_2 \cdot \frac{\pi}{4} \cdot B_1^2 + \omega^2 \cdot \rho_a \cdot L_r \cdot \frac{\pi}{4} \cdot D_r^2 \cdot \left( S + C_1 + \frac{L_r}{2} \right) \]

\[ + \omega^2 \cdot \rho_r \cdot L_b \cdot \frac{\pi}{4} \cdot D_b^2 \cdot \left( S + C_1 + L_r \right) \]  

The first of the three terms on the right hand side is the pressure force acting on the face of the piston. The second term is the centrifugal force generated by the rod and the third term is the centrifugal force generated by the roller bearings. The last two terms are of the form \( \omega^2 \rho \text{(volume)}(\text{distance from center of rotation}) \).

Forces on the second stage roller bearings.

\[ F_{b2} = P_3 \cdot \frac{\pi}{4} \cdot B_2^2 + \omega^2 \cdot \rho_a \cdot L_r \cdot \frac{\pi}{4} \cdot D_r^2 \cdot \left( S + C_1 + \frac{L_r}{2} \right) \]

\[ + \omega^2 \cdot \rho_r \cdot L_b \cdot \frac{\pi}{4} \cdot D_b^2 \cdot \left( S + C_1 + L_r \right) \]  

This equation has the same form as eq 4.6.

Speed of the roller bearings

\[ \omega_b = \frac{\omega \cdot D_o}{D_b} \]  

Force-area ratio for the linear bushing of the first stage

\[ p_{L1} = \frac{F_{b1} \cdot \sin(\alpha)}{D_r \cdot L_b} \]  

4.9
Force-area ratio for the linear bushing of the second stage

\[ p_{L2} = \frac{F_{b2} \cdot \sin(\alpha)}{D_r \cdot L_{ib}} \]  

Average sliding velocity on linear bushing

\[ V_r = \frac{\omega}{2 \cdot \pi} \cdot 2 \cdot S \]  

Length of connecting rod

\[ L_r = L_{ib} + S + C_2 \]  

The information extracted from the simulation program can be expressed in the form of functional relationships for

interstage pressure

\[ P_2 = f(S, B_1, B_2, \varepsilon_1, \varepsilon_2, P_1, P_3, T_1, T_{ba}) \]  

interstage temperature

\[ T_2 = f(S, B_1, B_2, \varepsilon_1, \varepsilon_2, P_1, P_3, T_1, T_{ba}) \]  

contact angle

\[ \alpha = f(D_o, L_r, D_b, S, C_1) \]
This set of equations and unknowns can easily be reduced. By combining Eq 4.2 with Eq 4.1 and combining Eq 4.4 and Eq 4.5 with Eq 4.3, three equations and three unknown can be eliminated. Similarly, by combining Eq 4.12 with Eq 4.6 and Eq 4.7 another equation and unknown are eliminated. Finally, the three quantities found from the computer simulation can essentially be treated as equations since the values found only depend on geometric variables (such as stroke, bores, outer ring diameter etc.). By combining these three quantities with the equations where they are used, the set is further reduced by three unknowns.

The reduced set is thus 8 unknowns:

\[ S, B_1, B_2, D_1, L_4, D_b, L_6, D_o, \]

and 8 equations which are:

mass flow for first stage

\[
\dot{m}_1 = \left[ 1 - \varepsilon \left( \frac{P_2}{P_1} \right)^{\gamma_n} \right] \cdot \rho_1 \cdot \frac{\pi}{4} \cdot B_1^2 \cdot S \quad \text{4.16}
\]

mass flow for second stage

\[
\dot{m}_2 = \left[ 1 - \varepsilon \left( \frac{P_3}{P_2} \right)^{\gamma_n} \right] \cdot \left( \frac{P_2}{R \cdot T_2} \right) \cdot \frac{\pi}{4} \cdot B_2^2 \cdot S \quad \text{4.17}
\]
force felt by first stage roller bearings

\[ F_{b1} = P_2 \cdot \frac{\pi}{4} \cdot B_1^2 + \omega^2 \cdot \rho_a \cdot L_{ib} + S + C_2 \cdot \frac{\pi}{4} \cdot D_r^2 \cdot \left( S + C_1 + \frac{L_{ib} + S + C_2}{2} \right) \]

\[ + \omega^2 \cdot \rho_z \cdot L_b \cdot \frac{\pi}{4} \cdot D_b^2 \cdot \left( S + C_1 + L_{ib} + S + C_2 \right) \]

force felt by second stage roller bearings

\[ F_{b2} = P_3 \cdot \frac{\pi}{4} \cdot B_2^2 + \omega^2 \cdot \rho_a \cdot L_{ib} + S + C_2 \cdot \frac{\pi}{4} \cdot D_r^2 \cdot \left( S + C_1 + \frac{L_{ib} + S + C_2}{2} \right) \]

\[ + \omega^2 \cdot \rho_z \cdot L_b \cdot \frac{\pi}{4} \cdot D_b^2 \cdot \left( S + C_1 + L_{ib} + S + C_2 \right) \]

roller bearing angular velocity

\[ \omega_b = \omega \cdot \frac{D_o}{D_b} \]

force over area felt by first stage linear bearings

\[ p_{L1} = \frac{F_{b1} \cdot \sin(\alpha)}{D_r \cdot L_{ib}} \]

force over area felt by second stage linear bearings

\[ p_{L2} = \frac{F_{b2} \cdot \sin(\alpha)}{D_r \cdot L_{ib}} \]

average sliding velocity on linear bearing.

\[ V_s = \frac{\omega}{2 \cdot \pi} \cdot 2 \cdot S \]
These quantities given by these equations are quantities which were described in the design constraints. The mass flow rate design constraint makes the two mass flow rate equations true equalities: the mass flow rate must be a single set value for the constraint to be satisfied. The rest of the constraints make the 6 other equations inequalities. In other words, any force felt by a given bearing that is smaller or equal to the maximum allowable force for that bearing is acceptable and the constraint is satisfied. The same is true of the sliding velocity and angular velocity equations.

Since there are 8 unknowns, 2 equality constraints and 6 inequality constraints, the design problem has six degrees of freedom. Since six degrees of freedom makes this problem difficult to solve graphically, one of the unknowns will be chosen before starting the optimization. The value for this unknown, Lb, was chosen from experience with previous intensifier design and after running many simulations to get an understanding of the effect of these variables on the design. Furthermore, an additional consideration was used to reduce the problem to a four degrees of freedom problem. As will be shown, the torque curves (torque versus θ found with the simulation program) were used to determine the best ratio of the first and second stage bores. The determination of the 8 unknowns will now be described in detail.

The procedure used to find the optimal point is shown by the flowchart in Fig. 4.6 and described by the subsequent discussion of the procedure. It should be noted that the rectangles with rounded corners in the diagram represent the points at which a value for a particular design variable was chosen. The reason that there are only seven decision boxes while there are eight unknowns is because one of the decision boxes contains two unknowns (the two bores). All other boxes represent steps in the calculations that were needed to make these choices. The fact that an unknown was set by practical considerations can also be seen.
Choose area ratio with torque curves

Set up spreadsheet with stroke as a variable (column)

Calculate forces acting on first stage roller bearings

Pressure Forces
Rod Centrifugal Forces
Bearing Centrifugal Forces

Calculate bore from area ratio, stroke and required mass flow

Set up rod diameter as second spreadsheet variable (row)

Choose roller bearing length from practical considerations

Find intercooler pressure with computer simulation

Choose linear bearing length from practical considerations

Calculate pressure forces from bore and intercooler pressure

Calculate rod length with stroke and linear bushing Pv limitation

Set up bearing diameter as third spreadsheet variable (depth)

Choose bearing diameter from optimal speed considerations

Calculate rod mass

Calculate bearing mass

Calculate rod distance from center of rotation from stroke and rod length

Calculate bearing distance from center from stroke and rod length

Calculate centrifugal forces generated by rod from angular velocity

Calculate centrifugal forces generated by bearings

Choose rod diameter from optimal bearing load and speed considerations

Choose stroke from optimal bearing load and speed considerations

Bore of first and second stage are set by stroke, bore ratio and required mass flow

Choose minimum outer ring diameter from stroke, rod length and bearing diameter

Figure 4.6
Design Procedure
The first step in determining the eight design variables was to determine the ratio of the displacement volumes of the two stages. Since the stroke is the same for the two stages, this is the same as the ratio of piston areas. The effect of varying the area ratio will now be examined.

A large first stage area and small second stage area results in a large portion of the work being done by the first stage which results in a high intercooler pressure. The high intercooler pressure results in a higher average pressure in the first stage and this, combined with the large first stage area, results in large forces generated in the first stage. The average pressure in the second stage is also higher but the area of this stage is small. If the area of the first stage is lowered and the second stage increased, the intercooler pressure drops which means that the forces in the first stage are lowered. At the same time, the second stage area has increased while the average pressure in this stage has decreased.

It is advantageous to have similar forces in both stages, as this results in a smooth torque curve, (i.e., torque vs crank angle). A smooth torque curve results in an intensifier which is easier to run. While this is not a major consideration if the intensifier is run by a diesel engine, it would be important if the intensifier was driven by a small electric motor. Also, a smooth torque reduces the load variations through the drive shaft and rotor which in turn increases the life on the intensifier before fatigue failure. It is thus worthwhile to study the effect of area ratio on the forces generated in the stages and thus the torque curve.

It is very difficult to reduce the torque curve to an equation as the pressure in all four cylinders varies with time. For this reason, the computer simulation was used to obtain the optimum bore ratio. This was done by choosing three strokes that are in the expected range of the final stroke. These strokes are 19.05 mm, 25.40 mm and 31.75 mm. The first and second
stage bores were chosen in such a way as to provide the required mass flow. The bores were then varied in order to yield a different area ratio yet still provide the required mass flow.

The first attempts made showed that the second stage forces were considerably larger than those of the first stage due to the larger pressures. This made it difficult to obtain a smooth torque curve without having a very large first stage and a small second stage. For this reason, the chamber formed at the back of the second stage piston was linked to the intercooler and thus pressurized to intercooler pressure. This had the effect of reducing the second stage forces since an area equal to the second stage area minus the area covered by the connecting rod was exposed to the intercooler pressure. This resulted in a force acting in the opposite direction to the pressure forces generated in the compression chamber.

The graphs generated by the simulation can be seen in Fig. 4.7 to 4.9. It can be seen that the smoothest torque curve always occurs at approximately the same bore ratio.
Figure 4.8
Torque vs Crank Angle - Stroke of 25.4 mm

Figure 4.9
Torque vs Crank Angle - Stroke 31.75 mm
For all three strokes, the bore ratio that gave the smoothest torque curve was in the region of 1.62-1.68. A bore ratio of 1.625 was chosen. This translates into an area ratio of 2.64 which, combined with the required mass flow rate, set the required displaced volume of both stages. These volumes were found to be 33 985 mm$^3$ for the first stage and 12 870 mm$^3$ for the second stage.

The next step on Fig. 4.6 consists of finding the optimum stroke. This step was done using a Lotus 123 spreadsheet. The spreadsheet format is sketched out in Fig. 4.10.

<table>
<thead>
<tr>
<th>Stroke</th>
<th>Varies from 6mm to 40mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore 1st stage</td>
<td>Variance calculated from stroke and req'd mass flow</td>
</tr>
<tr>
<td>Axial Load on Bearing</td>
<td></td>
</tr>
<tr>
<td>Sum of:</td>
<td></td>
</tr>
<tr>
<td>- pressure force calculated from bore and inter-cooler pressure</td>
<td></td>
</tr>
<tr>
<td>- centrifugal force due to rod calculated from rod dia, rod length and distance from center of rotation</td>
<td></td>
</tr>
<tr>
<td>- centrifugal force due to rollers calculated from bearing length, bearing diameter and distance from center of rotation</td>
<td></td>
</tr>
</tbody>
</table>

Figure 4.10
Sketch of Spreadsheet Format
There were in fact two spreadsheets with the same format. The only differences are that the calculated value was bearing speed instead of axial load on the bearing and the last column was maximum allowable bearing speed.

A column was set up which had the stroke varying from 6 mm to 40 mm in 2 mm steps. The reason the maximum allowable stroke is 40 mm comes from the sliding velocity limitation on the linear bushing. A 40 mm stroke results in an average sliding velocity of approximately 3 m/s, which is the maximum allowable velocity, at 2100 RPM. The minimum stroke of 6 mm was chosen because a shorter stroke would probably present many machining difficulties as the tolerances would need to be very low to keep the volumetric efficiency high.

Beside each stroke is the required bore. This bore is calculated from the required displaced volume for a given stage which was found earlier. It should be noted that the spreadsheet was set up for one stage at a time. The remainder of the discussion will refer to the first stage spreadsheet only as the two were found to yield very similar results.

Running the computer simulation gave an idea of the intercooler pressure. This pressure varied little with stroke for a given area ratio.

Knowing the intercooler pressure and the bore, the maximum pressure force acting on the first stage can be calculated for each stroke. To find the total force acting on the first stage, the centrifugal forces need to be calculated.

The centrifugal forces were calculated for two components of the first stage: the connecting rod and the roller bearings. The forces for the piston head were not included as they were found to be considerably smaller since the piston head center of gravity is close to the center of rotation. In order to calculate the force, three quantities need to be found: mass of the
component, its distance from the center of rotation and the speed of rotation. The speed was set at 2100 RPM, or 220 s⁻¹ as this is the worst case.

In order to find the mass of the connecting rod, its length and diameter need to be determined assuming that it is made of aluminum and its density is known. The diameter of the rod is a design variable so it was made to vary from 10 mm to 40 mm in the row direction of the spreadsheet. In order to determine the rod length, the length of the linear bearing must be known. This length was calculated from the load limitation on the linear bushing. Since sliding velocity is known (from the given stroke), all that is needed to find the required bushing area is the load. This load is the side load on the rod which results from the fact that the rollers are not rolling on a surface perpendicular to the rod axis. The angle between the rod axis and the inside surface of the outer ring was found (with the computer simulation) to vary little with stroke and approximately 5°. To find the side load on the rod, the axial force is needed. This quantity is unknown. Indeed, it is the quantity that is presently being found. As a first approximation, the axial force was set at twice the pressure force. This approximation can be improved later as the centrifugal forces are found which made this process an iterative one. With this approximation, a linear bushing length can be calculated. The centrifugal force generated by the rod can now be calculated.

At this point the spreadsheet looks like a matrix with the stroke varying from top to bottom and the rod diameter varying from left to right. At each position in the matrix, the sum of the pressure forces and the centrifugal forces generated by the rod are added together. The only force left to add in is the centrifugal force generated by the roller bearings.

It was assumed that the roller bearings were made of steel which yields the upper limit for their density. Both their length and their diameter were chosen as design variables but it would be
difficult to include them in the matrix as a third and fourth dimension. For this reason, the bearing length was chosen and the bearing diameter was found graphically.

The bearing length was set at 100 mm. The length was chosen after making it vary in the spreadsheet and the behavior of the design was examined. Since the load bearing capability of the rollers increases linearly with length, increasing the bearing length increases the safety factor associated with the bearing load. The effect is somewhat diminished by the fact that the bearing mass (and the centrifugal force due to this mass) also increases with length. From the standpoint of bearing life, the longest possible bearing is the most attractive. From the considerations of minimum size and weight, a longer bearing incurs a large penalty because the outer ring (which is very massive) and the entire frame must be made larger to encase the roller bearings. For this reason, it was judged that a maximum bearing length of 100 mm would be set.

The bearing diameter was initially chosen at 20 mm. It will be shown how this value was varied in the spreadsheet to find the optimum bearing diameter.

Now knowing the bearing length and diameter, as well as their distance from the center of rotation (from the stroke and the rod length), the centrifugal force generated by the rollers can now be included in the matrix. The matrix now yields the total axial force for each stroke and rod diameter combination for a given roller length and diameter. This result is shown graphically in Fig. 4.11. Fig. 4.12 is an enlargement of Fig. 4.11 in the region of interest. The maximum allowable bearing force was found from linear equation found in section 4.6.1.
Figure 4.11
Bearing Load vs Stroke and Rod Diameter

Figure 4.12
Bearing Load vs Stroke and Rod Diameter
The following points can be extracted from the graphs. First, the forces felt by the rollers increase greatly with a decreasing stroke. Second, the rod diameter has a relatively small effect on the bearing forces. There is a noticeable improvement when the rod diameter is increased from 10 mm to 20 mm but any larger rod offers little improvement. Third, with the bearing length and diameter chosen, it is possible to have forces smaller than the maximum allowable.

By considering the bearing load limitation only, it seems that a long stroke is preferable. The only design constraint that has not yet been used is the roller bearing speed limitation. The roller bearing speed depends on three quantities: the speed of the rotor (2100 RPM), the bearing diameter and the outer ring diameter. The outer ring diameter is chosen by knowing the stroke, rod length and bearing diameter. The bearing speed can thus be calculated for every stroke and rod diameter combination. This result is shown graphically in Fig 4.13 and 4.14.

![Figure 4.13](image-url)  
**Figure 4.13**  
Bearing Speed vs Stroke and Rod Diameter
The following points are shown by these graphs. First, the bearing speed increases with increasing stroke. Second, the rod diameter has an effect on speed similar to its effect on bearing forces: there is a significant improvement for bearing speed if the rod diameter is increased from 10 mm to 20 mm but any further increase in diameter does not yield much improvement.

The graphs showing bearing load and speed versus stroke and rod diameter were generated for a given roller bearing diameter. The optimum value for this design variable will now be determined.

When the roller bearing diameter was decreased from the starting point of 20 mm, the load situation became worse, that is to say that the axial load increased faster than the maximum allowable load. The opposite is true when the roller bearing diameter was increased: the load situation was better.

The same is not true for the speed situation. It was found that the situation changed little when the roller bearing diameter was increased or decreased slightly. When the variation was
larger, the situation became worse for both the increase and decrease of the roller bearing diameter. This behavior points to an optimum bearing diameter for the speed considerations. Fig. 4.15 shows the actual speed and the maximum allowable speed for a given stroke and rod diameter. It should be noted that the graph would have the same shape for different strokes or rod diameters but the position of the allowable line would be different.

From Fig. 4.15, it is clear that a bearing diameter between 18 mm and 31 mm is optimum for the speed considerations.

In summary, the findings of the analysis are as follows:

- the maximum stroke is 40 mm from linear bearing sliding velocity considerations
- the roller bearing load limitation can be met and the safety factor increases with increasing stroke
• the roller bearing speed limitation can be met and the safety factor increases with decreasing stroke
• the roller bearing load safety factor also increases with increasing roller bearing diameter
• the roller bearing speed safety factor is maximum for a roller bearing diameter between 18 mm and 31 mm
• the roller bearing load and speed safety factors increase with increasing rod diameter but with little improvement past a rod diameter of 20 mm.

The above procedure yielded all the values for the design variables related to the first stage. Because of the rotary design, the stroke of the two stages must be the same. The fact that the area ratio was chosen from torque considerations raises questions about the optimality of the chosen stroke for the second stage. For this reason, the above procedure was repeated for the second stage in order to ensure that this stroke was optimal. In fact, that was relatively easy to do since the analysis was in spreadsheet form.

In view of these findings, the following choices were made. First a bearing diameter of 22 mm was chosen since it is in the optimal range. The rod diameter was also chosen to be 22 mm since it would yield the advantages of reducing bearing loads and of keeping the bearing rolling speeds down without going to a very large rod. Also, it would be easier to mate the rod with the bearings if the sizes were the same. With the help of the load and speed graphs, a stroke of 25,4 mm was chosen which yielded a first stage bore of 41,3 mm and a second stage bore of 25,4 mm. The roller bearing length was kept at 100 mm. From the stroke, rod length and bearing diameter, the outer ring diameter was set at 346,1 mm.
4.8 Auxiliary Components

In addition to the design variables optimized in the previous sections, there are other components that need to be designed. These components do not affect directly the design variables above so their design can be treated separately. The two main components which need to examined are the valves used in the intensifier and the rotary seals, a component that is specific to the rotary intensifier configuration.

4.8.1 Valves

The valves used in the intensifier are modified NUPRO check valves [8]. This is the same type of valves used in the previous intensifier design. A drawing of the valves can be seen in Fig. 4.16.

Figure 4.16
NUPRO Check Valve [8]
There were, in the previous intensifier versions, problems associated with these valves such as short life. This was mainly due to the area of the valve being too small for the required flow. This caused the flow velocities to be very high through the valves which cause undue wear on the bonded O-ring. Once this O-ring was worn, the valves permitted considerable leakage. This problem was solved in the previous design by going to a larger size of valves (CH8). It is these larger valves that were used in the rotary intensifier design. The only place that the smaller valves (CH4) were used is for the second stage intake. This was done because the larger valves yielded an unacceptably high clearance ratio for the small second stages.

The only parts from NUPRO that were actually used in the intensifier are the poppet, the poppet stop and the spring. The valve body was not used but rather holes of the same size were machined directly into the rotor. After the poppet and spring were inserted, an insert of the same size as the second half of the valve body was fitted into the hole and held all the pieces in position. The inserts were held in place by the shafts attached to the rotor. This arrangement permitted the valves to be placed much closer to the cylinders which reduced the clearance volume penalty of the valves.

4.8.2 Rotary Seals

In order to transfer the high pressure gas from the stationary gas lines to the rotating hub requires some type of rotary seal. Three methods of accomplishing this task were considered for the intensifier application: the face seal, non-contacting seals and the mechanical seal.

A simplified face seal can be seen in Fig. 4.17. The stationary seal to which the gas lines are connected house a graphite ring. This ring is pressed against the rotating face of the rotor by a spring or compressed O-ring. The disadvantage of this type of seal is that they are not well
suited to high pressure differences. Also, the graphite provides better sealing when it is hot so when the rotor is not rotating and there is no friction to heat the graphite ring, there is poor sealing. For the installation of the intensifier in an urban bus, this poor sealing translates into important leakage when the bus is not in use.

In order to improve the sealing, it is possible to add more graphite ring, thus reducing the pressure difference across a given ring. The problem with this arrangement is the limited space available on the rotor face, especially when the fact that there need to be two separate stationary seals for the intensifier, one for intake and the other for exhaust, which need to be on the same face (since the other face is used to drive the rotor).
The second type of rotary seal is the non-contacting seal. In this arrangement, the sealing is done by the fluid that is being transferred. The two parts of the seal, the rotating face and the stationary face, are machined in such a way that the clearances between them are very small. The viscosity of the fluid and the sliding motion of the two faces stop the fluid from flowing out between the two faces. This arrangement is satisfactory when the fluid is a viscous liquid such as hydraulic fluid or oil. The problem with trying to seal a high pressure gas is that the clearances need to be extremely small and precise.

Finally, the last type of rotary seal considered is the mechanical seal, sometimes called a shaft seal. This type of seal can be seen if Fig. 4.18.

![Figure 4.18 Mechanical Seal](image)

In this type of seal, the seal body and the exterior casing are stationary and the shaft is rotating. The exterior casing is usually attached to a larger framed and thus linked to the bearings.
that guide the shaft. The seal body outside diameter is slightly smaller than the inside diameter of the casing which permits it to take up any eccentricity between itself and the shaft. The actual sliding is done by O-rings which are held stationary in the seal body. As can be seen in Fig. 4.18, two O-rings are usually used to reduce the leakage across the seal.

This type of seal is used in many applications where a rotating shaft needs to be sealed. Marine propeller shafts and centrifugal compressor shafts are two important applications. For the case of the propeller shaft, the purpose of the seal is to stop water from entering the hull. For a ship, the pressure is considerable lower than the pressures encountered in the intensifier. In the case of submarines, the pressures are comparable, but leakage is tolerated since it is possible to pump out any inflowing water.

The centrifugal compressor shaft seal is used to separate the high pressure gas inside the compression chamber and the atmosphere. The pressures encountered in a centrifugal compressor are usually considerably lower than the pressure in the intensifier and centrifugal compressor are often used to compress air where leakage is tolerable. The state of the art centrifugal compressor shaft seals use non-contacting surfaces that are shaped in such a way as to provide some dynamic sealing [9]. This dynamic sealing requires high angular velocities which are not present in the intensifier. A shaft seal manufacturer who was consulted did not recommend the use of this type of seal for the high pressures and relatively low angular velocities of the intensifier.

The only type of seal that seemed to be adaptable to the particular case of the intensifier was the mechanical seal that did not use any dynamic sealing but used O-rings for the sealing (Fig. 4.18). A recent publication was found which described the design of such seals and all the equations that follow were taken from this reference [10].
One of the problems of using O-rings is what is referred to as the Gow-Joule effect. According to this effect, an O-ring under tensile stress which is heated tends to contract. If the O-ring is used in a conventional manner, that is slightly stretched over the shaft, it will contract when it is heated from friction. When the O-ring contracts, the friction increases, as does the tendency to contract. This cycle continues until there is catastrophic failure of the O-ring.

In order to avoid the failure of the O-ring due to the Gow-Joule effect, the O-ring must be placed under compressive stress. This is accomplished by designing the O-ring groove in such a way that the outside diameter of the groove is smaller than the outside diameter of the O-ring. The procedure used to find the appropriate size of the groove and the correct O-rings given the diameter of the shaft will now be discussed.

The following values need to be determined:

- H  housing diameter
- G  O-ring groove diameter
- W  width of O-ring groove
- OD O-ring outside diameter
- ID O-ring inside diameter

The following parameters are known

- S  Shaft diameter
- N  angular velocity
- P  pressure difference across the seal

The shaft diameter was chosen to be 25.4 mm. It is advantageous to have a small shaft since this reduces the sliding velocity between the shaft and the seal and thus reduces frictional
heating. The reason the shaft was not made even smaller is because both intake and exhaust gases must travel in this shaft and it is desirable to have large passages in order to avoid a pressure drop in the shaft.

The angular velocity for the worst operating condition is 2100 RPM. This translates into a sliding velocity of approximately 550 fpm. The worst pressure difference across a seal is 200 bar.

For high pressure differences and high sliding velocities, the publication recommends using a small cross section O-ring (0.070"). For the pressure involved, the clearance between the shaft and the housing should be smaller than 0.006" with a 90 shore A durometer hardness O-ring material in order to avoid extrusion. Since this is the maximum allowable clearance, a radial clearance of 0.003" was chosen. The housing diameter (H) must then be 1.006".

The reference then gives a table for the O-ring groove depth and width which is applicable for this case. The groove width (W) is given as 0.079" and the depth (measured from the shaft surface) is given as 0.066". The O-ring groove diameter (G) must then be 1.132".

The last step is to find an O-ring that will fit into the groove with the proper amount of compressive stress. The way this is done is to find an O-ring with approximately 8% larger outside diameter than the groove. For this groove, the O-ring should have an 1.223" outside diameter. An 2-023 O-ring would have a 5.2% smaller outside diameter and an 2-024 O-ring would have a 10.8% smaller outside diameter. Because of the demanding operating conditions of the intensifier it would be preferable to use a 2-024 but there were some problem in making a hard (90 shore A durometer hardness) O-ring fit into a groove that was much smaller than the O-ring itself. For that reason, a 2-023 O-ring was used.
The life of the O-ring used in the seal can be predicted from the product of the sliding velocity and the pressure difference (Pv). For the intensifier, the sliding velocity of 550 fpm and the pressure difference of 3000 psi result in a Pv value of $16.5 \times 10^5$ fpm x psi. For indefinite life, a value of $12 \times 10^5$ is recommended by an O-ring manufacturer and a value of $2 \times 10^5$ is recommended by the reference. The elevated Pv value for the intensifier reduces the expected life of the O-rings to approximately 45 min. It should be noted that this very short life is for the worst operating condition: maximum speed and pressure difference. The intensifier spends very little time at this extreme operating point.

In order to further extend the O-ring life, some measures were taken to remove heat from the vicinity of the O-ring. As can be seen in Fig. 4.18, there is an oil chamber between the two O-rings of a given seal. This oil serves two purposes. First, it provides a fluid film between the shaft and the O-ring. This has the same effect as having a smoother shaft which causes less wear on the O-ring. Second, it transports heat away from the O-ring into the body of the seal. Another measure taken to remove heat was to build the seal body out of bronze which was recommended by the reference for good heat dissipation.

This concludes the discussion of the design of the intensifier. Assembly drawings for the final design can be found in Appendix D. Also in this Appendix are all the drawings for the individual pieces.
Chapter 5

Experimental Apparatus

The purpose of this chapter is to describe the equipment used to measure the performance of the intensifier. This equipment includes the test rig, the gas piping and the instrumentation. A key part of the measurement of the performance of the intensifier is the data acquisition system used and this will also be described. Finally, the testing procedure used will be examined.

5.1 Test Rig

The term test rig is used to describe the equipment used to run the intensifier. This includes a power source and frame on which to attach the intensifier and all related components. Fig. 5.1 shows the major components of the test rig.

The power source is an electric motor equipped with a DC motor controller. Its speed is set by the operator on the control panel. The tachometer attached to the shaft of the motor measures the speed and sends a signal to the control panel (which displays the speed) and to the data acquisition system. The tachometer also sends a signal to the motor controller which uses this signal as feedback to keep the motor speed constant.

A flywheel is attached to the shaft of the motor. The purpose of this arrangement is to dampen out cyclic variations in power requirements of the intensifier. Also on this shaft is the belt
pulley through which the power of the motor is transmitted to the intensifier. Finally, there is an oil pump attached to the end of the shaft which takes oil from a reservoir and supplies oil to the intensifier.

The motor casing is mounted on two trunnion bearing which permit it to rotate freely. Because of this arrangement, any torque developed by the motor and transmitted to the intensifier generates an equal and opposite torque in the motor casing. Since the casing is free to rotate, a torque arm and load cell attached to it permit the direct measurement of the torque.

All these components, the motor, pulley, flywheel and oil pump are held in place with bearings mounted on the frame. This frame rests on rubber pads which dampen out vibration of
the rig. Also attached to this frame is the table which lies above the axis of the motor shaft. On this table is mounted the intensifier with its pulley. The motor pulley and intensifier pulley are linked by a gear belt.

Technical specifications for the electric motor, motor controller, belt drive and flywheel are available in Appendix E. The piping used to supply the intensifier with gas and oil, as well as the instrumentation of the intensifier, will now be described.

5.2 Piping Plan and Instrumentation

The piping plan can be divided into two parts: the oil supply system and the gas system. The oil system is shown in Fig. 5.2.
From an oil reservoir (approximately 2 l capacity), the oil is pressurized by a DDC 1-71 fuel pump to approximately 70 psi. The oil is then directed through a flexible hose to the oil valve used to control the flow of oil into the intensifier. A pressure gage measures the oil pressure after the valve. The oil pressure is usually kept at approximately 5 psi above atmospheric. This pressure was found to be sufficient to ensure flow of oil through the intensifier.
The oil then enters a manifold from which it is directed through three hoses to the main bearings and the oil seal. The seal transfers the oil into the shaft of the intensifier from which it is routed to all four pistons. At the pistons, the oil lubricates the Permaglide bushing and also enters the connecting rod to flow to the roller bearings.

The oil is then collected at the bottom of the outer ring as it flows from the two main bearings and the roller bearings. It then leaves the intensifier through a hole at the bottom of the outer ring and returns to the reservoir via flexible hose.

It was found that the oil coming out of the roller bearings did not immediately drain through the hole in the outer ring. Some of this oil leaked through the sides of the outer ring and onto the table. In order to remedy this problem, O-rings were fitted on the sides of the ring but some leakage persisted. A pan was then fitted under the intensifier to collect any leakage of oil. This pan is emptied at the end of each test.

Fig. 5.3 shows the natural gas piping. In this circuit, the following data is measured:

- Inlet Pressure (P1)
- Intercooler Pressure (P2)
- Exhaust Pressure (P3)
- Inlet Temperature (T1)
- Intercooler Temperature (T2)
- Exhaust Temperature (T3)
- Mass Flow Rate
The gas is provided by high pressure CNG tanks. Coming out of the tanks, the gas is filtered and goes through a regulator which sets intake pressure to the intensifier. The gas then goes through a flow meter which measures the mass flow rate. The temperature of the intake gas is measured by a thermocouple (T1) and the gas is then directed to an accumulator whose function is to dampen out cyclic variations in intake pressure due to the reciprocating nature of the intensifier. There is a similar accumulator in the exhaust tract that has the same function. From the accumulator the gas goes to the intake gas seal after the inlet pressure has been measured by a strain gage pressure transducer (P1). The purpose of the gas seal is to transfer the gas to the rotating shaft of the intensifier rotor.
After being compressed by the intensifier, it exits the latter through the exhaust gas seal where the exhaust temperature is measured by a thermocouple (T3). The gas pressure is measured by a strain gage pressure transducer (P3) before the gas goes through a pressure relief valve which is used to set the exhaust pressure of the intensifier. The gas is then returned to another high pressure CNG tank to be reused in another test.

It is worthy to note that there is a third gas seal. This seal is used to take some gas at intercooler conditions out of the intensifier in order to measure the temperature (T2) and the pressure (P2) of the gas after the first stage compression. This is done with a thermocouple and a strain gage pressure transducer. There is no flow through this piping except to fill it at the start of a test.

In addition to the gas properties measured at different points measured above, the following variables were also measured in order to evaluate the performance of the intensifier:

- Gas Seal Wall Temperature
- Torque
- Motor Speed

The gas seal wall temperature thermocouple is used to ensure that the gas seals temperature does not exceed the safe operating temperatures of the O-rings used in the seals.

The torque is measured using a load cell attached to a torque arm which is bolted to the casing of the electric motor. Fig. 5.4 shows a partial end view of the rig in which this assembly can be seen. Finally, the motor speed is measured by a tachometer attached to the end of the motor shaft. All the technical specifications of the transducers listed above, as well as details concerning the calibration of the transducers, can be found in Appendix F.
5.3 Data Acquisition

The data acquisition system is shared between the intensifier and the 6V92 test rig. It is composed of two main components: the hardware and the software. The hardware will be examined first and is shown in Fig. 5.5.
The signals from the transducers are fed to signal conditioning modules. These modules filter and amplify the signals as well as providing the cold junction compensation for the thermocouples. After conditioning, the signals are sent through an interface board to a switch box. This switch box controls which signals are sent to the PC: either those from the intensifier or from the 6V92. After the switch box, the signals are sent to an A/D board (PCL 818) which is inside the PC. The software then transforms the signals into useful quantities. Technical information concerning the above components can be found in Appendix G.
The software first uses a calibration file to transform the voltage signals into values in engineering units. This calibration file is generated by the user and it is discussed in Appendix F. The software then accesses a configuration file which contains data such as the bore of the stages and the speed ratio between the motor and the intensifier. This configuration file also contains information on which values to show on the monitor display. These values can be the data taken from the transducers or calculated values such as power consumed, volumetric efficiency and pressure ratio.

Finally, the software permits saving data to a disk so that it can be analyzed later using a spreadsheet. The software generates two output files: one that contains the values read from the transducers and the other that contains the calculated values. The subroutine used to calculate these values is listed in Appendix H.

5.4 Testing Procedure

The following variables are controlled by the tester:

- Intake Pressure
- Exhaust Pressure
- Intensifier Speed
- Eccentricity

Since the present working value for the injection pressure is 200 bar, the exhaust pressure will be set at this value for the testing and will not be varied. The other three variables will be varied in such a way as to cover the range of operating conditions for which the intensifier was designed.
The first series of tests was as follows:

- constant speed (200 RPM)
- maximum stroke (1 inch)
- pressure ratio varying from 4:1 to 12:1
- constant outlet pressure (200 bar)

This series of tests was intended to provide data for a significant comparison to the previous intensifier designs. In fact, there is no use of the variable stroke capability so the intensifier is in fact acting exactly like a conventional reciprocating intensifier.

The second series of tests were done under the same conditions (200 RPM, 200 bar exhaust pressure) but with varying strokes. Each individual test was done with a constant stroke but there were 5 different tests. The strokes for the tests varied from 0.8 to 0.0 inches in 0.2 inches increments. This series of tests were intended to examine the variable capacity aspect of the intensifier.

The third series of tests were done at a speed of 400 RPM. These tests were intended to repeat the second series of tests (five tests at different eccentricities) at a higher speed. Unfortunately, there were problems with failures caused by overheating which limited the series at a single test done at full eccentricity.

The final series of tests was intended to examine the effect of speed on the intensifier performance. Since testing at higher speeds is limited by operating temperatures, these test were done at a given pressure ratio and a given eccentricity. These parameters were chosen in such a way to give a chosen mass flow rate (maximum engine fuel requirement).
Chapter 6

Results

6.1 Introduction

As discussed in chapter 5, there are three variables which are controlled during the testing: speed, stroke and intake pressure. The testing procedure was also described in this chapter and is summarized in Table 6.1.

<table>
<thead>
<tr>
<th>Stroke (in)</th>
<th>200 RPM</th>
<th>400 RPM</th>
<th>Varying from 200 to 350 RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>Intake Pressure of 127 bar</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.4</td>
<td>Intake Pressure varying from 53 to 100 bar</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.6</td>
<td>Intake Pressure varying from 22 to 55 bar</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.8</td>
<td>Intake Pressure varying from 16 to 66 bar</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.0</td>
<td>Intake Pressure varying from 16 to 63 bar</td>
<td>Intake Pressure varying from 18 to 33 bar</td>
<td>Intake Pressure of 20 bar</td>
</tr>
</tbody>
</table>

Table 6.1
Scope of Testing
As can be seen in Table 6.1, the speed was held constant for all but one test and the stroke was always held constant during a given test. As for the intake pressure, it was either held constant or varied with one of two methods. The first method consisted of starting at an elevated pressure and permitting the storage tank from which the gas was taken to empty, thus decreasing the intake pressure. The second method was to change the intake pressure with a regulator and take several data points at each chosen pressure ratio. The first technique was used for most tests while the second was used for the low mass flow test, that is the 0.2 inch stroke test.

Data during a given test was taken as soon as the exhaust pressure was set at 200 bar (with a pressure relief valve) and the intake pressure was at the desired starting value. Steady state was never fully achieved because the temperature of the intensifier was continuously rising due to inadequate cooling. This cooling problem will be discussed in detail in the next chapter. The effect of overheating will be shown in some of the results of the testing. This overheating problem results in the fact that these tests are not completely repeatable since operating temperature affects intercooling and volumetric efficiency.

6.2 Low Speed Testing

Testing at 200 RPM consisted of running the intensifier at a given eccentricity and varying the intake pressure. The test was then repeated for other eccentricities. The performance data that will be examined are mass flow rate, power requirements, power per unit mass flow, volumetric efficiency and isentropic efficiency. Most of the performance data will be examined versus overall pressure ratio.
6.2.1 Mass Flow Rate

The mass flow rate was measured in the intake line between the intensifier and the storage tank. For this reason, the measured mass flow rate also includes any leakage out of the intensifier. This error would tend to make the measured intensifier performance better than reality. For this reason, the leakage was always examined at the beginning of a test before the intensifier was started. Any leakage above 1% of a typical intensifier flow rate was corrected before any testing was done. Fig. 6.1 shows a typical measured mass flow rate for a full stroke test.

![Mass Flow Rate for Full Stroke Testing at 200 RPM](image)

In Fig. 6.1, the solid line is the mass flow rate predicted by the computer simulation. Good agreement was achieved for the following reasons. First, the clearance volume used in the computer simulation is the same as the measured clearance volume in the intensifier. Second, the empirical formulation for heat transfer in the computer simulation was adjusted in such a way that
the interstage pressure in the simulation matched the measured interstage pressure. This matching can be seen in the stage pressure ratios in Fig. 6.2

Finally, the intake temperature for the computer simulation was set at approximately the average intake temperature of the test. The latter would vary as expansion through regulators and check valves and mass flow rate would vary during the test.

The stroke was varied by changing the eccentricity. The following strokes were examined: from 0.2” to 1.0” in 0.2” increments. This series of 5 tests gave the following mass flow rate graph (Fig. 6.3)
The effect of the different methods of varying intake pressure can be seen in Fig. 6.3. The high stroke tests show a continuous variation from high to low pressure ratio while the low stroke test only have a few discreet point.

The mass flow rate graph for various strokes still shows good agreement between the measured mass flow rate and the predicted value from the computer simulation. This graph also shows that the intensifier was overdesigned since the required mass flow rate of 500 mg per revolution translates into 6 kg/hr for a speed of 200 RPM. At the worst pressure ratio (10:1) the intensifier provides a mass flow rate of 7.8 kg/hr at full stroke. This overdesign was intentional and done to compensate for any losses in the valves or leakage across seals. This safety margin should diminish at higher speeds as these losses increase.
6.2.2 Power and Specific Power

The power required to run the intensifier was calculated with the measured torque and speed. A value for power can also be extracted from the computer simulation. This power was calculated with the following equation:

\[ Power = \sum_{0}^{360} F_\theta \cdot \sin(\alpha_\theta) \cdot r_\theta \cdot \omega \cdot d\theta \]

where \( F_\theta \) is the force acting on the roller bearing due to pressure forces on the piston, \( \alpha_\theta \) is the contact angle between the roller bearings and the outer ring calculated form geometry, \( r_\theta \) is the distance from the center of rotation to the contact point between the outer ring and roller bearings and \( \omega \) is the rotational speed. Since all components in the computer simulation are frictionless and the pressures are calculated using polytropic coefficients (1.25 for compression and 1.4 for expansion), the value of power given by the computer simulation is the same as the polytropic power. The difference between the measured power and the power from the computer simulation is the power loss due to friction. The power required for the intensifier can be seen in Fig. 6.4.
This graph shows what was expected: the required power decreases with stroke. This is due to the fact that the power required to compress the gas is lower at lower strokes because there is less gas to compress.

To get an idea of the friction losses, the required power can be compared to the power predicted by the computer simulation and the isentropic power. The isentropic power is calculated in the data acquisition program from the pressure ratio and the inlet temperature to both stages. This comparison can be seen in Fig. 6.5 for the full stroke case.
From this graph, it can be seen that the isentropic power is much less than the predicted power from the computer simulation. There are two reasons for this difference. First, the computer simulation uses polytropic coefficients which are different from the isentropic one and there is a simple heat transfer model in the program. Second, there is an important error in the calculation of isentropic power as it is difficult to obtain a good measurement of the temperature of the gas entering the second stage. Interstage gas is piped out of the intensifier where its pressure and temperature are measured but the gas has time to cool substantially before reaching the thermocouple so that the temperature read is only a few degrees above room temperature. The error is worse at high pressure ratios because this is when the interstage temperature is highest.

Because of the error in the isentropic power calculation, the predicted power will be used to calculate efficiency. This value of efficiency will be referred to as compressor efficiency. The
predicted power does take into account some heat transfer but this efficiency gives a valuable indication of power losses due to friction. As can be seen from Fig. 6.5, approximately half the power is lost to friction. This result will be discussed in greater detail in a latter section.

Another important performance characteristic is the required power per unit mass flow (specific power). The measured results can be seen if Fig. 6.6

![Specific Power for Various Strokes at 200 RPM](image)

The fact that higher strokes require less power per unit mass flow was expected since the mass flow is much higher for higher strokes and power lost to friction is expected to be similar for all stroke because of the three main sources of frictional losses (piston rings, roller bearings and gas seals), only the friction loss due to piston rings should diminish at lower strokes.
6.2.3 Compressor Efficiency

As mentioned earlier, the compressor efficiency calculation will use the power predicted by the computer simulation as opposed to the isentropic power. This is not isentropic efficiency but the efficiency calculated in the manner is still important since it gives an indication of the friction losses. This efficiency can be seen in Fig. 6.7 for various strokes.

The fact that the friction losses are approximately the same for all strokes explains the fact that the efficiency is highest at high strokes since the power required to compress the gas is highest in this case.
6.2.4 Volumetric Efficiency

The volumetric efficiency is calculated in the acquisition program with knowledge of the first stage displaced volume and the intake conditions. The results for volumetric efficiency can be seen in Fig. 6.8.

![Graph of Volumetric Efficiency](Figure 6.8 Volumetric Efficiency for Various Strokes at 200 RPM)

The fact that the volumetric decreases with stroke was expected since the clearance volume increases greatly when the stroke is decreased.

6.3 High Speed Testing

This section will examine the effects of raising the speed of the intensifier. Unfortunately, the testing at higher speeds was limited to a full stroke test because of excessive overheating of the intensifier due to inadequate cooling. This overheating caused premature failures in components such as seals and piston rings.
The importance of the 400 RPM test is to give a basis for the extrapolation of the performance data into the speed range of the engine (600 to 2100 RPM). With the present configuration, testing at such speeds is impossible for two reasons. First, the test rig belt drive and electric motor limit the maximum intensifier speed at approximately 700 RPM. Second, and more importantly, the present intensifier configuration does not permit adequate cooling of the rotor and this causes failures in the rotor which makes the testing impossible.

The overheating of the rotor also affects the data taken at 400 RPM and these effects will be discussed when appropriate.

6.3.1 Mass Flow Rate

The mass flow rate for the intensifier at 400 RPM is compared with the predicted mass flow rate from the computer simulation and also the mass flow rate from the 200 RPM test in Fig. 6.9

![Figure 6.9](image)

Mass Flow for Full Stroke at 200 and 400 RPM
The testing technique used is the same as for the low stroke testing at 200 RPM. The reason for this is that at high speeds, the storage tanks pressure drops too rapidly so the intake pressure is set by a regulator in order to get some steady state data.

The effect of the high temperatures can be seen in the mass flow graph. The point at approximately 11:1 pressure ratio which is above the curve is in fact the first point to be taken. Since the intensifier was not hot at the beginning of the test, there is more heat transfer out of the compressed gas into the intensifier body which results in lower gas density and thus a higher mass flow rate. The next point to be taken is the 16.5:1 point and then the pressure ratio was lowered so that the last point to be taken before the intensifier failed was the 6:1 point.

### 6.3.2 Power and Specific Power

The effect of overheating is also noticeable in the power graph (Fig. 6.10). A similar behavior to the mass flow graph is noticeable:

![Figure 6.10](image)

*Figure 6.10*

Measured and Predicted Power for Full Stroke at 200 and 400 RPM
The 11:1 point required more power than the rest of the point of the test since it was the first point to be taken and had a significantly higher mass flow.

The specific power requirement of the intensifier can be seen in Fig. 6.11.

![Figure 6.11](image)

**Figure 6.11**

Specific Power for Full Stroke at 200 and 400 RPM

It is interesting to note that the power per unit mass flow is very similar for both the 200 and the 400 RPM cases. Since mass flow is proportional to intensifier speed, this graph suggests that the power required for the intensifier is also proportional to speed. A more detailed investigation of the effect of speed on the power requirement will be made in a further section.
6.3.3 Compressor Efficiency

The compressor efficiency calculations for the 400 RPM cases are the same as the 200 RPM cases. The predicted power from the computer program is used instead of the isentropic power. The results for both the 200 and 400 RPM full stroke test can be seen in Fig. 6.12.

![Figure 6.12: Efficiency for Full Stroke at 200 and 400 RPM](image)

It is difficult to compare the two curves because of the differences in operating temperatures. Also, wearing in of the linear bushings and gas seals may have reduced the frictional load on the high speed test as it was done after the set of 200 RPM tests.

6.3.4 Volumetric Efficiency

Since the geometry of the intensifier is the same for both 200 and 400 RPM tests, the volumetric efficiency should not vary from one test to the other. The measured values of volumetric efficiency can be seen in Fig. 6.13.
Figure 6.13
Volumetric Efficiency for Full Stroke at 200 and 400 RPM

The effect of operating temperature is very apparent here. The first data point to be taken in the 400 RPM test shows a higher volumetric efficiency than the equivalent 200 RPM point. This is due to the fact that the intensifier is cooler for this point. As the intensifier becomes hotter, the volumetric efficiency drops below the level of the 200 RPM tests. The difference between the two tests is most noticeable for the low pressure ratio region of the test when the intensifier is at its hottest in the 400 RPM tests but its coolest for the 200 RPM tests. This graph shows the important effect of cooling on the volumetric efficiency.
6.4 Variable Speed Testing Results

All previous tests were done with a set eccentricity and speed and a variable pressure ratio. In order to study the effect of variable stroke, a given test was repeated with different eccentricities. In order to study the effect of speed, it would be preferable to repeat the same tests at higher speeds but this proved to be impossible because of inadequate cooling of the rotor. For this reason a different testing procedure was used.

The following data was taken with a constant eccentricity and a constant pressure ratio. The speed was varied from 200 RPM to 350 RPM in 20 RPM increments. The mass flow, power, specific power, efficiency and volumetric efficiency are the performance characteristic that are examined.

It proved to be difficult to maintain a constant pressure ratio for the entire test and even small variation in intake pressure caused important variations in mass flow and power. Fig. 6.14 shows the power required by the intensifier at various speeds.

![Figure 6.14](image-url)
The reason that the test ended at 350 RPM was because the gas seals failed due to overheating. The points at a given speed show a wide spread because of the varying intake pressure. The fact that power is proportional to speed can be seen in this limited test range and when comparing the 200 and 400 RPM tests. This results in a constant compressor efficiency for full stroke in the tested speed range.
Conclusions and Recommendations

7.1 Conclusions

7.1.1 Discussion of Objectives

The first objective was to design and build a variable stroke, multi-stage, mechanically actuated reciprocating intensifier which would meet the design requirements for fuel flow in both design and off design operation of the diesel engine. This objective was met as the first rotary intensifier prototype was built. The design optimization procedure used to arrive at the final intensifier configuration was a graphical one which is described in chapter 4.

The eccentricity of an outer ring and the axis of rotation of the rotor generates the reciprocating motion of the pistons. Two compression stages were chosen and the variable stroke is supplied by the fact that the outer ring can be moved and thus the eccentricity can be varied.

The requirements for operating pressures were met as the intensifier could achieve an exhaust pressure of 200 bar with a varying intake pressure. The flow requirements at 200 RPM were satisfied. Testing was limited to 200 RPM tests and very limited 400 RPM tests because higher speeds caused very rapid overheating and subsequent failure of rotor components.
The second requirement, that of measuring the performance of the intensifier in the light of the thermodynamic model was also met. The test rig used in previous intensifier designs was modified and the instrumentation adapted to the rotary prototype. The thermodynamic model, which consisted of a computer program which simulated the intensifier, proved to be an invaluable design tool. Also, the simulation results matched the experimental results closely.

The variable capacity capability of the rotary design was studied and showed that the variable stroke method of capacity control can meet the requirements of variable fuel flow for the diesel engine. This method did not prove to be as energy efficient as had been hoped because of the presence of high friction losses. These friction losses proved to be fairly constant over the range of mass flow studied. This constant friction loss resulted in low compressor efficiencies at low mass flow rates. The power requirements of the intensifier were somewhat lower at reduced mass flow because of the lower compression work required.

The final objective, that of assessing the essential design limitations of this new class of variable stroke, high pressure intensifiers, was also met. The rotary intensifier design concept is limited by intrinsic sources of problems which are discussed in detail in the following section.

7.1.2 Problems of Rotary Configuration

The testing of the first prototype of the rotary intensifier revealed four major sources of problems of the rotary configuration. Two of these problems are unavoidable because they are caused by the rotary geometry. These problems are excessive forces due to the presence of large centrifugal forces and the leakage due to the sealing of rotating parts. The two last problems could be avoided in the next intensifier prototype. These problems are related to alignment and overheating.
Centrifugal Forces

The first major source of problems of the rotary configuration is the fact that the rotating machinery generates centrifugal forces. This is an important problem as at operating speeds of the order of the engine speed, the centrifugal forces are of the same order of magnitude as the pressure forces. The components of the rotary intensifier thus have to withstand twice the forces encountered in a non-rotating intensifier of the same capacity.

The presence of the large centrifugal forces adds complexity to the design calculations. These forces also reduce the size of the feasible domain, that is the options when determining the design variables. Finally, and most importantly for the eventual production model, the large forces reduce the safety factor for components such as bearing and linear bushings which in turn reduces their life.

Rotary Gas Seals

The second major design limitation lies in the rotary seals. The challenge of transferring high pressure gas from stationary lines to the rotor or vice versa is a considerable one. Because of the high pressures and high sliding velocities, the seal life is very short: of the order of one hour. This is the main reason that high speed testing was limited. Also, any misalignment between the gas seals and the shaft greatly reduce the seal performance. Finally, in order to seal the high pressures, the tolerances must be very small which makes the machining difficult. The tight fit also results in high frictional losses and high seal temperatures. The friction losses due to the gas seals are the main cause of the low compressor efficiency at low mass flows.
Alignment

The alignment problems originate from the fact that there are too many components that must be properly aligned in order to ensure proper functioning. First, the two shafts attached to the hub must be on the same axis so that when the rotor is held by the main bearings, neither shaft oscillates. This problem is amplified by the fact that there are plates between the hub and the shafts and little room to fit in bolts. The pistons must then be aligned with the rotor in order to ensure that their reciprocating motion is perpendicular to the axis of rotation. The rollers must be perpendicular to the piston rods and thus parallel to the axis of rotation of the rotor.

The two main bearings must be on the same axis so that the rotor can turn freely. The alignment of the main bearings is made difficult by the fact that it requires the alignment of the two frame plates and two bearing holders. All these components are attached to the base plate which must be flat, even after being bolted to the test rig table. The frame plates also have to hold the outer ring in such a way that its inner surface in parallel to the axis of rotation but is still allowed to move freely in order to have a variable eccentricity.

Finally, the gas cap must hold the gas seals in such a way that they are concentric to the shaft of the rotor. Any eccentricity at the gas seals caused undue wear and poor sealing.

Overheating

Overheating was the main source of failures in the intensifier. High temperatures were encountered in two areas of the intensifier: the gas seals and the rotor. The gas seals temperature was high because of the friction between the O-ring and the shaft and because of the presence of high temperature exhaust gases. The high gas seals temperatures would cause expansion of the O-ring which result in higher contact forces between the O-ring and the shaft, causing even higher
heat production due to friction. This cycle would continue until the O-rings would seize unto the shaft and the seal would fail.

High temperatures were also encountered in the rotor. The main reason for this is that the rotor, being encased in the frame and outer ring, was not properly cooled. There was little flow of fresh air into the outer ring and no supplementary cooling system had been planned. The high temperature caused failure of the Teflon piston rings. The situation is similar to the gas seal: the friction of the piston ring against the cylinder wall would generate heat which would cause expansion of the piston ring. The larger contact force between the piston ring and the wall caused rapid wear of the piston ring which would result in poor sealing and contamination of the oil with Teflon particles.

These are the major design limitations of the rotary configuration. The following section will provide modifications that can be applied to the existing intensifier or incorporated in the next version.

### 7.2 Recommendations

The recommendations that follow first deal with modifications that could be used in order to diminish the effects of the design problems described in the previous section. The last section describes general recommendations that do not touch upon the limitations but could still improve the intensifier performance.

#### 7.2.1 Centrifugal Forces

The centrifugal forces cannot be eliminated. They can be minimized by attempting to reduce the mass of the components, their distance from the center of rotation or the angular
velocity. Reducing the angular velocity would require a gearbox between the intensifier and the engine. This would result in a weight and cost penalty and since the intensifier is turning at a lower speed, it would require larger pistons in order to provide the same mass flow rate. This would result in larger pressure forces so the problem is not entirely solved. Reducing the distance between the components and the center of rotation requires a more compact hub. There cannot be much improvement over the present design as there is little wasted space inside the hub. The final option of reducing the component mass could be done with a more exotic materials but this would result in a sharp increase in price. The centrifugal force limitation, which places limits on the expected life of the intensifier, cannot easily be solved.

7.2.2 Rotary Gas Seals

Perhaps the most important design limitation is the gas seals. The need for transferring high pressure gas between stationary lines and the rotor cannot be avoided (unless the storage tanks and engine are also made to rotate). The three factors that affect seal life are pressure, sliding velocity and operating temperature. The pressure cannot be reduced as they are set by factors outside the intensifier design. The sliding velocity (of the order of 2.5 m/s in the present design) could be reduced by reducing the angular velocity of the intensifier or by reducing the diameter of the shaft. Reducing the angular velocity is an option that has many repercussions in the design because of the required mass flow rate. There is a minimum shaft diameter because of the presence of the passages that carry the gas to the rotor.

Since the pressures and the sliding velocity cannot be changed enough to make an important difference in seal life, the last factor, that of operating temperature, must be examined closely. The temperature immediately around the O-ring that is providing the sealing is
considerably higher than the surroundings because of the friction between the O-ring and the shaft. This friction can be minimized by ensuring that the seal and the shaft are concentric. This results in an even distribution of contact pressure around the O-ring and avoids any excessive friction in one localized area. Also, adding a lubricant close to the O-ring and the shaft can reduce friction by creating a film between the O-ring and the shaft. This method was already used in the present version but the possibility of pressurizing the lubricant could be examined. The lubricant can also be used to carry heat away from the O-ring which reduces the operating temperature. The seal body design could also be improve to increase the heat transfer away from the O-ring. Use of materials such as copper which have a high conductivity or the addition of water cooling are examples of how the heat transfer from the O-ring could be improved.

If the rotary intensifier configuration is to be pursued any further, efforts must be concentrated on the rotary seals. Without a good rotary seal with an acceptable life, the rotary configuration is worthless. It would be beneficial to set up experiments where only the seal is studied, without any attention paid to the rest of the intensifier. A fully floating design with better lubrication and water cooling could prove to be long lasting.

7.2.3 Alignment

The problem of misalignment can mostly be solved with a design which takes into account alignment in the first phases of the design process. Components that need to be aligned should be remachined as one component. For example, the two shafts attached to the rotor could be machined as one shaft going through the hub. Also, the frame plates and the bearing holders could be machined as one U-shaped bearing holder which would house both main bearings and in which would sit the outer ring. Finally, components such as the gas seals should be designed so
that they are self aligning. The shaft or the seals could be made totally free to float so that they automatically take up any eccentricity. The alignment problems of the present prototype could be corrected with some modifications and could be completely avoided in the next design.

7.2.4 Overheating

The obvious solution to the overheating problem is better cooling. This could be done in several ways. First, the intensifier could be built in a more open configuration. In this configuration, the frame would not completely enclose the rotor, thus allowing fresh air to flow around the rotor. This would make the intensifier more dangerous because moving parts would be exposed but it would make it much more easy to diagnose any problems with the rotor. This would be invaluable in the refinement stage of the testing. This could be accomplished with an arrangement as seen in figure 7.1 and 7.2 where there are no frame plates but rather the use of a U-shaped bearing holder discussed earlier.
Figure 7.1
Side View of Open Configuration
This configuration would also have the advantage of simplifying the alignment of the frame. The problem of holding and locating the outer ring could be handled by having a tract in the base plate in which the outer ring could slide back and forth. The eccentricity control could be done by a screw in this tract.

Even with the open design, cooling may prove to be inadequate. Water or oil cooling are also cooling option. The cooling fluid could be made to flow through the rotor by using a seal similar to the oil seal on each shaft of the rotor. These seals, as well as the oil seal, are similar to
the gas seal but do not suffer from their short life since the pressures are much lower and there is effective cooling due to the oil or water flow.

The main problem with the open design is that the oil would tend to flow out of the side of the outer ring. The oil system in the present version already needs some refinement as there is much waste of oil because the oil collection system is inadequate and oil tends to flow out of the outer ring even with the presence of the frame walls. The oil system could be improved so that it could function in an open intensifier design.

7.2.5 General Recommendations

These recommendations are not directly related to the rotary design limitations but are possible modifications that would improve the intensifier performance.

The next version should avoid the use of aluminum. The advantages of using steel are worth the weight penalty. Steel is usually harder than aluminum so that surfaces that need to be smooth (for O-ring seals for example) do not become marked as easily. Steel is also stronger than aluminum so that components that need to be flat in order to provide proper alignment do not warp when they are bolted to other components. Finally, steel does not produce as many wear particles that contaminate the lubrication oil.

The main problems of the rotary-reciprocating configuration, the gas seals and centrifugal forces, come from the fact that the pistons are rotating. A new type on intensifier which incorporates the design concept of variable stroke but does not use rotating pistons is presently being considered. Because of patent laws, this new intensifier cannot be discussed in this thesis.
REFERENCES


8. Swagelok catalogue


APPENDIX A

**Engine Specifications**

The engine for which the conversion kit is being designed is for one configuration of the Detroit Diesel Corporation (DDC) 6V92-TA family. This engine is a heavy duty two stroke diesel engine having 6 cylinders in a V configuration. Each cylinder displaces 92 cubic inches and the engine is turbocharged and after cooled.

The purpose of this appendix is to derive the engine fuel requirement from the data given by DDC for the 285 hp 6V92-TA. This data is reproduced here as are the calculations and graphs for the fuel requirement. First, some general data for the engine will be listed.

**General Data**

<table>
<thead>
<tr>
<th>Model</th>
<th>6V92-TA Coach</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Cylinders</td>
<td>6</td>
</tr>
<tr>
<td>Bore and Stroke (in)</td>
<td>4.84 x 5.00</td>
</tr>
<tr>
<td>Displacement (in³)</td>
<td>552</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>17:1</td>
</tr>
<tr>
<td>Exhaust Valves per Cylinder</td>
<td>4</td>
</tr>
<tr>
<td>Combustion System</td>
<td>direct injection</td>
</tr>
<tr>
<td>Engine Type</td>
<td>63.5° VEE two stroke</td>
</tr>
<tr>
<td>Aspiration</td>
<td>Turbocharged</td>
</tr>
</tbody>
</table>
Physical Data

Length (in) 37.6
Width (in) 37.6
Height (in) 48.0
Dry Weight (lb) 2020

Fuel System

Fuel Injector Part No. 5324850
Fuel Injector Timing 1.460
Cart Code 0002
Fuel Consumption (lb/hr) 103.5
Fuel Consumption (gal/hr) 15.5
Fuel Spill Rate (lb/hr) 489
Fuel Spill Rate (gal/hr) 73.2
Total Fuel Flow (lb/hr) 593
Total Fuel Flow (gal/hr) 88.7

Cooling System

Coolant Flow (gal/min) 160
Maximum Top Tank Temperature (°F) 210
Minimum Top Tank Temperature (°F) 160
Minimum Coolant Fill Rate (gal/min) 3.0

Performance Data

Power Output (bhp) 285
Full Load Speed (RPM) 2100
Peak Torque (ft lb) 870
Peak Torque Speed (RPM) 1200
BMEP (lbf/in²) 92
Table A1.1 shows the performance data for the 6V92-TA Coach and this data is also shown in graphical form below. Notice that the data had to be extrapolated for speeds below 1000 RPM. This was done with a cubic fit to the data with Lotus 123.

<table>
<thead>
<tr>
<th>Engine Speed (RPM)</th>
<th>Power (bhp)</th>
<th>Torque (ft lb)</th>
<th>BSFC lb/bhp hr</th>
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<tr>
<td>2100</td>
<td>285</td>
<td>713</td>
<td>.358</td>
</tr>
<tr>
<td>1950</td>
<td>276</td>
<td>743</td>
<td>.355</td>
</tr>
<tr>
<td>1800</td>
<td>265</td>
<td>773</td>
<td>.350</td>
</tr>
<tr>
<td>1600</td>
<td>248</td>
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<td>.341</td>
</tr>
<tr>
<td>1400</td>
<td>228</td>
<td>855</td>
<td>.335</td>
</tr>
<tr>
<td>1200</td>
<td>199</td>
<td>870</td>
<td>.334</td>
</tr>
<tr>
<td>1000</td>
<td>161</td>
<td>846</td>
<td>.343</td>
</tr>
</tbody>
</table>

Table A.1
Data from DDC for the 6V92-TA Coach
Figure A.1
Power Output of the 6V92-TA Coach

Figure A.2
Brake Specific Fuel Consumption of the 6V92-TA Coach
Figure A.3
Fuel Consumption of the Engine in kg/hr

Figure A.4
Fuel Consumption in mg/revolution
The above data and extrapolations were used to understand the engine fuel requirements and this information was needed to size the intensifier. Notice that even though natural gas has a slightly higher LHV (lower heating value) than diesel fuel, the fuel requirement for natural gas was taken to be the same as the fuel requirement for diesel. This was done to give a slight overdesign to the intensifier that would compensate for other expected problems.
APPENDIX B

Capacity Control Example

The following assumptions will be made: a single stage reciprocating machine compressing natural gas \((n=1.3)\) with a clearance ratio of 10\%. This would give it a volumetric efficiency of 51\% at a pressure ratio of 10:1 and 100\% for a pressure ratio of 1:1 from the fact that:

\[
\eta_v = 1 - \varepsilon_0 \left[ \left( \frac{P_2}{P_1} \right)^{\frac{1}{\gamma}} - 1 \right]
\]

The mass flow rate per revolution of a single stage reciprocating machine is given by:

\[
\frac{\dot{m}}{n} = \frac{P_1 \cdot V_d \cdot \eta_v}{R \cdot T_1}
\]

The displaced volume \(V_d\) is chosen in such a way that the machine provides a mass flow of 500 mg/rev an intake pressure of 20 bar (which means a 10:1 pressure ratio assuming a 200 bar exhaust pressure) and intake temperature of 20\(^\circ\)C, that is the worst case. The displaced volume found is 7.2x10\(^{-5}\) m\(^3\).

Using this displaced volume for the case where the intake pressure is 200 bar (1:1 pressure ratio) intake temperature is the same at 20\(^\circ\)C, then the mass flow per revolution is an immense 9800 mg/rev. This is 195 times engine requirements.
In this appendix, the design of a centrifugal compressor that will meet the requirements for mass flow rate and pressure ratio will be attempted. All equations and assumptions are taken from reference [11]. The design will start with the following known data:

- 20 bar intake pressure
- 300 K intake pressure
- $\gamma = 1.3$ for natural gas
- $R = 500 \text{ J/kg} \cdot \text{K}$ for natural gas

For high pressure ratio centrifugal compressors, the maximum attainable compressor efficiency ($\eta$) is approximately 80%. Also, stresses in the rotor limit the rotor tip velocity ($U_t$) at 650 m/s. Finally, high pressure ratio compressors are typically design with blades leaning back from the radial direction as much as 30° ($\beta_2$).

Since the flow in the tanks and in the injector have relatively low velocities, the pressure rise can be expressed in terms of the stagnation pressures. The stagnation pressure ratio can be expressed as:

$$\frac{p_{03}}{p_{01}} = \left[ 1 + \eta \left( \gamma - 1 \right) \left( \frac{U_t}{a_{01}} \right) \left( 1 - \frac{wr_2}{U_t} \tan \beta \right) \right]^{\gamma/(\gamma - 1)}$$
and assuming that $C_p$ is approximately 2.8 [11], and is approximately 420 K. The outlet density is found with ideal gas and is 95 kg/m$^3$. Finally, the rotor radius is found to be 2x10^{-6} m. This is an extremely small radius for the rotor and it would be impossible to machine. Even is a smaller blade height (b) would be acceptable, the rotor would still be very small which would lead to large clearance losses. Also, with such a small radius, the angular velocity would be extremely high in order to generate the 650 m/s tip velocity (of the order of 3.25x10^8 rad/s or approximately 3x10^9 RPM).
APPENDIX D

Intensifier Drawings

This Appendix contains all the drawings of the rotary intensifier. The first two pages list the parts and part drawings numbers. This list is followed by two assembly drawings and finally the part drawings.

List of Parts

<table>
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<th>Drwg No</th>
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<td>Rotor</td>
<td>I-R1A</td>
</tr>
<tr>
<td></td>
<td></td>
<td>I-R2A</td>
</tr>
<tr>
<td></td>
<td></td>
<td>I-R3A</td>
</tr>
<tr>
<td>2</td>
<td>1st &amp; 2nd Stage Inserts</td>
<td>I-IN1A</td>
</tr>
<tr>
<td>3</td>
<td>1st Stage End Plate</td>
<td>I-IP1A</td>
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<td>4</td>
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<td>5</td>
<td>1st Stage Piston</td>
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<td>1st Stage Piston (mod)</td>
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<td>6</td>
<td>2nd Stage Piston</td>
<td>I-P2A</td>
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<tr>
<td>7</td>
<td>1st Stage Valve Inserts</td>
<td>I-VIF1A</td>
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<td>8</td>
<td>2nd Stage Valve Inserts</td>
<td>I-VIS1A</td>
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<tr>
<td>9</td>
<td>Bearing Rod</td>
<td>I-BR1A</td>
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<tr>
<td></td>
<td>Bearing Wheel</td>
<td>I-BE1A</td>
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<tr>
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<td>Drive Shaft</td>
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<td>Drive Shaft (mod)</td>
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<td>Drive Shaft Circular Plate</td>
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<td>Gas Shaft (mod)</td>
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<td>20</td>
<td>Frame</td>
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<tr>
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<tr>
<td>22</td>
<td>Outer Ring</td>
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<tr>
<td>23</td>
<td>Outer Ring Frame</td>
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</tbody>
</table>
DO NOT BREAK EDGE

NOTE: BREAK ALL EDGES UNLESS OTHERWISE NOTED
ALL OUTSIDE SURFACES MUST BE MACHINED
DO NOT BREAK EDGE

NOTE: BREAK ALL EDGES UNLESS OTHERWISE NOTED
ALL OUTSIDE SURFACES MUST BE MACHINED

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Intensifier Rotor

<table>
<thead>
<tr>
<th>Drawn by</th>
<th>Drawing no</th>
<th>Material</th>
<th>Required</th>
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</thead>
<tbody>
<tr>
<td>Alain Touchette</td>
<td>I-R3A</td>
<td>Aluminium 6061-T4</td>
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</table>
DRILL Ø0.999

MILL Ø1 3/8"
DEPTH 1.000"

GROOVE IS .081" DEEP AND .140" WIDE

1/4"

MILL (BALLNOSE) Ø1/4"
DEPTH .088

GROOVE IS .081" DEEP AND .140" WIDE

1.498
1.500
7/16

NOTE: TOLERANCE ON Ø25.000 HOLE: +.001
NOTE: .075"×45° CHAMFER ON BOTH ENDS OF Ø25.000 HOLES.
NOTE: TOLERANCE ON ALL OUTSIDE DIAMETERS: +.001"
NOTE: TOLERANCE ON #25.000 HOLES +.001
NOTE: .075°x45° CHAMFER ON BOTH ENDS OF #1/16 HOLES.

Drill 05/16
40 HOLES

Department of
Mechanical Engineering
UBC

1st Stage End Plate
drawn by
Alain Touchette
material
Aluminium 6061-T4

NOTE: TOLERANCE ON #25.000 HOLES +.001
NOTE: .075°x45° CHAMFER ON BOTH ENDS OF #1/16 HOLES.

Drill #5/16
40 HOLES

Drill #1/2
4 HOLES

#1/16 NPT

#1/8" holes:
I— 2.0 —

NOTE TOLERANCE ON Ø25.000 HOLE.

NOTE: Ø0.75' x 45° CHAMFER ON BOTH ENDS OF Ø25.000 HOLES.

DRILL Ø0.999

DRILL Ø0.125

4 HOLES

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Mechanical Engineering
UBC

2nd Stage End Plate

drawn by
Alain Touchette

material
Aluminium 6061-T4

drawing no
I-IP2A

no required

NOTE: TOLERANCE ON Ø25.000 HOLE ± 0.001

NOTE: Ø0.75' x 45° CHAMFER ON BOTH ENDS OF Ø25.000 HOLES.
NOTE: BOTH GROOVES ARE .122 WIDE x .189 DEEP
SHAFT SURFACE MUST BE GROUND
NOTE: BOTH GROOVES ARE .122 WIDE X .189 DEEP
SHAFT SURFACE MUST BE GROUND
NOTE: BOTH GROOVES ARE .083 WIDE x .125 DEEP
SHAF T SURFACE MUST BE GROUND
MILL Ø1/2" DEPTH .620

DRILL Ø3/8" -

MILL Ø11/16" DEPTH .360

DRILL Ø5/16" -

NOTE: ALL .09" CHAMFERS ARE 45°

NOTE: TOLERANCES ON ALL OUTSIDE DIAMETERS ARE: +.001, -.001
MILL Ø1/2"
DEPTH .620

DRILL Ø3/8"

MILL Ø7/16"
DEPTH .224

DRILL Ø3/16"

NOTE: ALL .090° CHAMFERS ARE 45°

NOTE: TOLERANCES ON ALL OUTSIDE DIAMETERS: ± .001

Department of Mechanical Engineering UBC

2nd Stage Valve Inserts

drawn by
Alain Touchette
drawing no
I-VISA
material
Aluminium 6061-T4
no required
2

.090
.686
.872
1.040
NOTE: SURFACE OF Ø.875 SHAFT MUST BE HARDENED AND GROUND
NOTE: INSIDE AND OUTSIDE DIAMETERS NEED TO BE HARDENED AND GROUND.

<table>
<thead>
<tr>
<th>Department of Mechanical Engineering</th>
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<tbody>
<tr>
<td>UBC</td>
</tr>
<tr>
<td>Bearing Wheel</td>
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<tr>
<td>drawing no I-BE1A</td>
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<tr>
<td>no required</td>
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<tr>
<td>material</td>
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<tr>
<td>Steel</td>
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</table>

The inside diameter is $\phi 11253 \pm 0.0005$ and the outside diameter is $\phi 1500$.
12500 RING GROOVE I.D. 5.500
.085 WIDE x .052 DEEP

MACHINED WITH .01/4" BALLNOSE
DEPTH 3/8"
2 GROOVES

DRILL .01/4"
1 HOLE

.085 WIDE x .052 DEEP

MACHINED WITH .03/16" BALLNOSE
DEPTH 3/8"
2 GROOVES

DRILL .03/16"
4 HOLES

.700
.392
.506
.763
.750
.125
1.875

.700
.750
1.875

1.250

6.250
3.250

.500

.050.00mm
.050.00mm
.01440

.375 WIDE x .185 DEEP

SQUARE KEYHOLE

ALL RING GROOVES ARE .085 WIDE x .052 DEEP WHERE POSSIBLE. LEAVE A 1/16" WALL BETWEEN HOLE AND RING GROOVE.
ALL D-RING GROOVES ARE .085 WIDE x .050 DEEP
WHERE POSSIBLE, LEAVE A 1/16" WALL BETWEEN
HOLE AND D-RING GROOVE.

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UBC

Drive Shaft

Drawn by
Alain Touchette
I-IDG3

Material
Steel

No required
1
WITH ALL O-RING GROOVE ARE .085 WIDE x .052 DEEP WHERE POSSIBLE. LEAVE A 1/16' WALL BETWEEN HOLE AND O-RING GROOVE.

PLATE IS 3/8' THICK WITH O.D. 6'
ALL D-RING GROOVES ARE .085 WIDE x .052 DEEP WHERE POSSIBLE, LEAVE A 1/16" WALL BETWEEN HOLE AND D-RING GROOVE.

PLATE IS 3/8" THICK WITH O.D. 6" AND HAS SAME PROFILE AS I-CP2B

CENTER HOLE: DRILL THROUGH HOLE Ø1/8" GROOVE Ø.D. Ø.196 DEPTH .052

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UBC

Description:

Oil Plate

drawn by:
Alain Touchette

drawing no:
I-CP1B

material:
Aluminium 6061-T4

no required
1
Note: #1.000 section of shaft needs to be ground to high finish.
Chamfer end of #1.000 shaft 15° for a length of 1/2".
Break all edges.

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UBC

Seal Shaft

Material: Steel

Drawn by: Alan Touchette
Drawing no: 1-SHIA

No required: 1
0—RING GROOVE I.D. 2.5 
.005 WIDE x .52 DEEP

0—RING GROOVE I.D. .405
DEPTH .032

MACHINE GROOVE WITH
.01/4" BALLNOSE 
DEPTH 5/16" 

DRILL Ø1/4
2 HOLES

DRILL Ø1/2'
4 HOLES

DRILL Ø1/4
DEPTH 5.5'
2 HOLES

DRILL Ø1/8" HOLE
DEPTH 3.9
GROOVE I.D. .196
DEPTH 3/32

NOTE: 0.000 SECTION OF SHAFT NEEDS TO BE GROUND TO HIGH FINISH
CHAMFER END OF Ø1.000 SHAFT 15° FOR A LENGTH OF 1/2'
BREAK ALL EDGES

Department of
Mechanical Engineering
UBC

Seal Shaft

drawn by
Alain Touchette

material
Steel

no required

1
NOTE: 

- Ø1.000 SECTION OF SHAFT NEEDS TO BE GROUND TO HIGH FINISH
- CHAMFER END OF Ø1.000 SHAFT 15° FOR A LENGTH OF 1/2"
- BREAK ALL EDGES

---

**Department of Mechanical Engineering**
**UBC**

**Seal Shaft**

<table>
<thead>
<tr>
<th>Material</th>
<th>Steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. req'd</td>
<td>1</td>
</tr>
</tbody>
</table>

**Drawn by**
Alain Touchette

**Drawing no**
1-SHIC
D-RING GROOVE I.D. 3/8" 
.085 WIDE x .052 DEEP

D-RING GROOVE I.D. 2-5/8" 
.085 WIDE x .052 DEEP

MACHINE GROOVE WITH 
3/8" BALLNOSE 
DEPTH 1/2" 

DRILL Ø5/16" 
2 HOLES

DRILL Ø1/2" 
4 HOLES

.3125 
.700 
1.125 
1.250 
.7625

4.500

.625

Department of 
Mechanical Engineering 
UBC

Gas Plate

drawn by 
Alain Touchette 
I-CP2A

material 
Aluminium 6061-T4 
no required 
1
DRILL THROUGH HOLE Ø1/8" GROOVE O.D. .196 DEPTH .040

MACHINE GROOVE WITH Ø1/4" BALLNOSE DEPTH 5/16"

DRILL Ø1/4" 2 HOLES

D-RING GROOVE O.D. 1/2" DEPTH 0.52

D-RING GROOVE I.D. 2.5" .085 WIDE x .052 DEEP

DRILL Ø3/8" 2 HOLES

.328 .700 1.125

.7825

5.500 5.000 4.500

.625

.250

.875

.125

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UBC

Gas Plate
drawn by
Alain TOUCHETTE
drawing no
I-CP2
material
Aluminium 6061-T4
no required

1
NOTE: THE SIX INNER O-RING GROOVES ARE .085 WIDE x .055 DEEP AND ARE 1/16" FROM BOLT HOLES ON EACH SIDE.
OUTSIDE DIAMETER 2.747

GROOVE IS .079 WIDE x .065 DEEP

MACHINE THESE 4 HOLES WITH A 1/2" ENDMILL DEPTH 1/2"

GROOVE IS .094 WIDE x .250 DEEP

GROOVE IS .530 WIDE x .250 DEEP

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UBC

Gas Seals

drawn by
Alain Touchette

material
Bronze

Drawing no
I-GS1A

no required
OUTSIDE DIAMETER 2.747

MACHINE THESE 4 HOLES WITH A 1/2" END MILL
DEPTH 1/2"

MACHINE THIS GROOVE WITH A 3/8" BALLNOSE
DEPTH .750

GROOVE IS .250 DEEP
GROOVES ARE .065 DEEP
45° CHAMFER
15° CHAMFER
30° CHAMFER

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Gas Seals

<table>
<thead>
<tr>
<th>Material</th>
<th>Drawn by</th>
<th>Drawing no</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bronze</td>
<td>Touchette</td>
<td>1-GS1B</td>
</tr>
<tr>
<td>No required</td>
<td></td>
<td>3</td>
</tr>
</tbody>
</table>
1/8"-27 NPT thread depth 3/8".

Drill 1/8".

DRILL 1/8"
4 HOLES

1/2" UNC thread.
4 HOLES

Groove is 1/8" deep.

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UBC

Top Bearing Bracket

Drawn by: Alain Touchette
Material: Steel

Drawing no: I-BME2A

No required: 1
NOTE: THE TWO GROOVES ON THE INSIDE DIAMETERS ARE BOTH .079 WIDE x .056 DEEP AND THEY HAVE A 1/16" WALL ON BOTH SIDES.
NOTE: THE FOUR 1/4" UNC HOLES ARE CENTERED ON PLATE THICKNESS
PLATE IS 1/2" THICK
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UBC

Frame Pin

drawn by
Alain Touchette
drawing no
I-FR3A

material
Steel
no required
4
NOTE: PLATE IS 1/2" THICK

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Eccentricity Control Plate

drawn by
Alain Touchette

material
Steel

I-FR4A

1

DRILL φ3/4"

DRILL φ9/32"

8 HOLES
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UBC

Spacer
drawn by Alain Touchette
drawing no I-FR05A
material Steel
no required 2
THREAD 1/2' UNC 8 HOLES

DRILL Ø1/8" 2 HOLES

MILL 1" WIDE x .250 DEEP GROOVE

MILL 2" WIDE x .125 DEEP GROOVE
NOTE: INSIDE SURFACE OF RING MUST BE GROUND AND HARDENED.
NOTE: THIS SURFACE MUST BE FLAT FOR ENTIRE LENGTH OF ASSEMBLY

<table>
<thead>
<tr>
<th>Department of Mechanical Engineering UBC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ring Frame Assembly</td>
</tr>
<tr>
<td>Material: Aluminum 6061-14</td>
</tr>
<tr>
<td>Tolerance: ±0.018</td>
</tr>
</tbody>
</table>
NOTE: THE 3/4" UNC HOLE IS ONLY NEEDED ON ONE OF THE TWO PIECES.
NOTE: A 3" HIGH BAR WILL GIVE SUFFICIENT MATERIAL TO MACHINE ARC.
APPENDIX E

Test Rig Specifications

Electric Motor

- Brand name: Hampston
- Type: variable speed, DC
- Mounting: trunnion mounted
- Speed: 0-2400 RPM
- Power: 25 hp at 2400
- Control: tachometer feedback speed regulation
- Shaft dia: 1.600"

Motor Controller

- Brand name: Randtronics
- Type: regenerative DC motor controller
- Model: TB 750 series
- Rated Power: 25 hp
- Line Voltage: 240 V, 3 phase
- Output: 0-240 V DC, 0-55 Amps
- Field: 100 V
- Mode: tachometer feedback speed regulation

Synchronous Belt

- Brand Name: Browning
- Type: gearbelt
- Model: HPT 8M
Drive Pulley:  B348M50SH
34 teeth
3.409 pitch diameter

Low Speed
Driven Pulley:  B1928M50E
192 teeth
19.195 pitch diameter
5.56 speed ratio
20.03" center distance with 2000 mm long belt (2000 8M50)
17.46 hp rating at 205 RPM driven pulley speed

High Speed
Driven Pulley:  B1128M50E
112 teeth
11.175 pitch diameter
3.29 speed ratio
19.62" center distance with 1600 mm long belt (1600 8M50)
22.91 hp rating at 530 RPM driven pulley speed

Flywheel
Outside dia:  16"
Width:  4"
Weight:  45 kg
APPENDIX F

Instrumentation Specifications and Calibration

The purpose of this Appendix is to describe the transducers used in the measurement of the intensifier performance. Details of the calibration of these instruments and the calibration file used by the data acquisition software are also included.

The following transducers are examined:

- Loadcell
- Thermocouple
- Strain gage pressure transducer
- Mass flow rate meter
- Tachometer

Loadcell

<table>
<thead>
<tr>
<th>Model:</th>
<th>Interface SM-250 (serial # C26470)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type:</td>
<td>Strain gage force transducer</td>
</tr>
<tr>
<td>Capacity:</td>
<td>± 250 lbf</td>
</tr>
<tr>
<td>Output:</td>
<td>3.218 V/V</td>
</tr>
<tr>
<td>Excitation:</td>
<td>10 VDC</td>
</tr>
<tr>
<td>Signal Conditioning:</td>
<td>5B38 isolated strain gage input</td>
</tr>
</tbody>
</table>
The calibration of the load cell was done with the load cell installed in its usual position (figure 4.4). A load arm, which measured 16 in, was attached to the motor shaft. The rotor and casing of the electric motor were linked so that a torque applied to the shaft would be transmitted to the casing (and thus measured by the load cell). The following calibration curve was generated:

![Loadcell Calibration](image_url)

**Figure F.1**  
Loadcell Calibration

In this graph, as well as all the calibration curves to follow, the square points represent the values read and the line is a fitted curve which was used in the calibration file. Also the output voltage was read after all signal conditioning.
Thermocouples

Model: Omega Thermocouple
Type: J type (iron+ constantan-)
Capacity: 0-760 °C
Output: 0-40 mV
Excitation: 10 VDC
Signal Conditioning: 5B37-J-01 isolated thermocouple input

There were four thermocouples used in the test rig to measure inlet, interstage, outlet and wall temperature. The calibration was done by placing the thermocouples in boiling water which was allowed to cool. The voltage after signal conditioning was read after each 10 °C drop in the water temperature. The calibration curves are found in figures F.2-F.5.
**Figure F.3**
Interstage Temp. Transducer Calibration

**Figure F.4**
Outlet Temp. Transducer Calibration
There are three pressure transducers used to measure inlet, interstage and outlet pressure. The calibration curves can be found in figures F.6-F.8. The calibration was done with a dead weight gage tester. Different hydraulic pressures covering the operation range were applied to the transducer while the voltage output was read after the signal conditioning.
Figure F.6
Inlet Pressure Calibration

Figure F.7
Interstage Pressure Calibration
Tachometer

Model: Signer
Type: Generator
Output: 7V / 1000 RPM
Signal Conditioning: 5B38 isolated strain gage input

The tachometer was calibrated with a hand held optical tachometer trained on the shaft of the motor. The same method was used to verify the speed ratio of the gearbelt transmission. The calibration curve generated is:
Mass Flow Rate Meter

Model: Micro Motion DH012S100
Transmitter Model: Micro Motion RFT9712
Type: Coriolis meter
Capacity: 0-300 kg/hr
Pressure Rating: 393 bar
Temperature Rating: -240 to 204 °C
Signal Conditioning: 5B32

The calibration of the mass flow rate meter is done with a supplied calibration computer.
CALIBRATION FILE FOR INTENSIFIER DATA ACQUISITION

Last modified 02-18-1994

Intensifier Version 2.00, 2-Stage Rotary-Reciprocating, Mechanical Drive, E-Motor

Torque
0
Nm
-.284, -.023662, .284, 0, 0, 0, 0, 0, 0, 0
-29.3625, 0, 29.3625, 0, 0, 0, 0, 0, 0, 0

Wall Temperature
6
°C
.54172, .99573, 0, 0, 0, 0, 0, 0, 0, 0
10, 90.556, 0, 0, 0, 0, 0, 0, 0, 0

Motor Speed
5
RPM
.0013, .1851, 3.86651, 0, 0, 0, 0, 0, 0, 0
0, 104, 2171, 0, 0, 0, 0, 0, 0, 0

Interstage Press
0
bar
-5, -.199, .05291, 4.7061, 0, 0, 0, 0, 0, 0
0, 0, 6.8948, 189.6058, 0, 0, 0, 0, 0, 0

Outlet Pressure
8
bar
-10, -2.865, -1.5102, 5.79822, 0, 0, 0, 0, 0
0, 0, 6.8948, 241.3165, 0, 0, 0, 0, 0

Inlet Temperature
7
C
.5447, .98312, 0, 0, 0, 0, 0, 0, 0
10, 90.556, 0, 0, 0, 0, 0, 0, 0

Outlet Temperature
6
C
.544034, .983173, 0, 0, 0, 0, 0, 0, 0
10, 90.556, 0, 0, 0, 0, 0, 0, 0

CNG dm/dt
0
kg/hr
-5, 0, 5, 0, 0, 0, 0, 0, 0
-50, 0, 50, 0, 0, 0, 0, 0, 0

Inlet Pressure
4
bar
0, .9783, 1.1583, 5.981, 0, 0, 0, 0, 0
0, 0, 6.8948, 206.843, 0, 0, 0, 0, 0

Int. Stg. Temp
6
C
.5447, .98312, 0, 0, 0, 0, 0, 0, 0
10, 90.556, 0, 0, 0, 0, 0, 0, 0, 0

Eccentricity
5
mV
0, 5, 0, 0, 0, 0, 0, 0, 0, 0
0, 5000, 0, 0, 0, 0, 0, 0, 0, 0
APPENDIX G

Data Acquisition Hardware Specifications

The data for the following components was taken from C. Aichinger's thesis [2]. Since no modifications were made to the data acquisition hardware, this data is still valid.

Signal Processing Modules

- Tachometer: 5B31 Isolated Voltage Input
- Mass flow: 5B32 Isolated Current Input
- Thermocouple: 5B37-J-01 Isolated Thermocouple Input - Type J
- Pressure Transducer: 5B38 Isolated Strain Gage Input
- Torque Load cell: 5B38 Isolated Strain Gage Input

A/D Board

- Model: Advantech, PCL 818, PC-LabCard Series
- Type: high speed, high performance I/O card for IBM A/D conversion
- Analog Input:
  - Channels: 16 single ended or 8 differential
  - Resolution: 12 bits
  - Converter: ADC-774
  - Conversion Time: 8 μs
  - Input Range Bipolar: ± 10V, 5V, 2.5V, 1V, 0.5V
  - Input Range Unipolar: 0-10V, 0-5V, 0-2V, 0-1V
  - Input Range Selection: Programmable
  - Trigger Mode: by software
Digital Output:
   Channels: 16
A/D Pacer and Counter:
   Device: Intel 8254
   Pacer: 32 bit with 10 MHz or 1 MHz time base
   Pacer max. rate: 2.5 MHz
   Counter: 16 bit counter with 100 kHz time base

Personal Computer
   Brand Name: ANO
   Type: IBM compatible
   CPU: Intel 80286
**Performance Data Calculation Subroutine**

SUB PerfCalc

' ************************************************************
' ** Subroutine to calculate performance values **
' ************************************************************

'Set constant values:

rho.air = 1.205 kg/m³ Density of air @ 20øC, 1.0133 bar
R.GEN = 8314.34 J/Kmol*K 'GENERAL GAS CONSTANT

'STANDARD VALUES FOR NATURAL GAS: FOR MORE INFORMATION
'density:
    rho.STD = .713142 kg/m³ PROPERTIES SEE:
'Molecular weight:
    M.STD = 17.1456999# kg/Kmol
'Gas constant:
    R.STD = R.GEN / M.STD J/kg*K

'B.C. HYDRO VALUES FOR NATURAL GAS:
'density:
    rho.BCH = .690351 kg/m³
'Molecular weight:
    M.BCH = 16.597737# kg/Kmol
'Gas constant:
    R.BCH = R.GEN / M.BCH J/kg*K

Kelvin = 273.15
k.isentr = 1.3

'Set variables:

Torque = EngData(VAL(chan$(1)))' N-m

RPM = EngData(VAL(chan$(2)))' Motor Speed in RPM
IF speed.rat = 0 THEN
  speed.rat = 1
END IF

intens.rpm = RPM / speed.rat

in.temp = EngData(VAL(chan$(3)))' CNG inlet temperature in °C
out.temp = EngData(VAL(chan$(6)))' CNG end temperature (after 2nd stage) in °C
in.press = EngData(VAL(chan$(4)))' CNG inlet pressure in bar
out.press = EngData(VAL(chan$(7)))' CNG end pressure (after 2nd stage) in bar
wall.temp = EngData(VAL(chan$(8)))' wall temperature of intensifier cylinder in °C
CNG.mass = EngData(VAL(chan$(5)))' CNG massflow in kg/hr
inter.temp = EngData(VAL(chan$(9)))' interstage temperature after 1st stage in °C
inter.press = EngData(VAL(chan$(10)))' interstage pressure in bar

IF LCASE$(chan$(11)) = "none" THEN
  stroke = 2 * VAL(cnst$(11)) * 2.54' Stroke in cm (2* eccentricity)
ELSE
  stroke = 2 * EngData(VAL(chan$(11))) * 2.54' Stroke in cm
END IF

'Calculate Intensifier Displacement:
*FIRST STAGE:
disp.1 = (bore.1 ^ 2) * pi / 4 * stroke * ncylinders.1' in cm³

*SECOND STAGE:
disp.2 = (bore.2 ^ 2) * pi / 4 * stroke * ncylinders.2' in cm³

Conversion of temperatures to Kelvin
in.tempK = in.temp + Kelvin' inlet temperature (K)
out.tempK = out.temp + Kelvin' outlet temperature (K)
inter.tempK = inter.temp + Kelvin' interstage temperature (K)
wall.tempK = wall.temp + Kelvin' wall temperature (K)

temp.diff.1 = inter.temp - in.temp' first stage in °C
temp.diff.2 = out.temp - inter.temp' second stage in °C
temp.diff.total = out.temp - in.temp ' total temperature difference in °C

IF in.tempK < 250 OR inter.tempK < 250 OR out.tempK < 250 THEN
  temp.ratio.1 = 999
  temp.ratio.2 = 999
ELSE
  temp.ratio.1 = inter.tempK / in.tempK
  temp.ratio.2 = out.tempK / inter.tempK
END IF

'Pressure ratios:
IF in.press < .1 THEN
  p.ratio.1 = 999
  p.ratio.total = 999
ELSE
  p.ratio.1 = inter.press / in.press     'first stage
  p.ratio.total = out.press / in.press  'overall pressure ratio
END IF
IF inter.press < .1 THEN
  p.ratio.1 = 999
ELSE
  p.ratio.2 = out.press / inter.press   'second stage
END IF

'Theoretical temperatures
theoret.out.tempK.1 = in.tempK * p.ratio.1 ^ ((k.isentr - 1) / k.isentr)
theoret.out.tempK.2 = inter.tempK * p.ratio.2 ^ ((k.isentr - 1) / k.isentr)

theoret.out.temp.1 = theoret.out.tempK.1 - Kelvin
theoret.out.temp.2 = theoret.out.tempK.2 - Kelvin

'Theoretical mass flow:

'FIRST STAGE:

IF intens.rpm < 5 THEN
  theoret.mass.1 = 999
ELSE
  rho1 = in.press / (R.BCH * in.tempK) ' density of intake gas (10*kg/cm^3)
  theoret.mass.1 = rho1 * disp.1 / 10 * (intens.rpm * 60) ' kg/hr
END IF
'Volumetric Efficiency:

'OVERALL:
vol.eff.tot = CNG.mass / theoret.mass.1 * 100' in %

'Power:

'FIRST STAGE:  [kW theoretical (isentropic) power consumption]
A = (k.isentr / (k.isentr - 1)) * CNG.mass * R.BCH * in.tempK / 3600 / 1000
isentr.power.1 = A * ((p.ratio.1 ^ ((k.isentr - 1) / k.isentr)) - 1)

'SECOND STAGE:  [kW theoretical (isentropic) power consumption]
A = (k.isentr / (k.isentr - 1)) * CNG.mass * R.BCH * inter.tempK / 3600 / 1000
isentr.power.2 = A * ((p.ratio.2 ^ ((k.isentr - 1) / k.isentr)) - 1)

'OVERALL:
isentr.power.tot = isentr.power.1 + isentr.power.2

act.power = Torque * RPM * 2 * pi / 60 / 1000' kW actual power consumption

IF CNG.mass <= .001 THEN
   power.per.mass = 999
ELSE
   power.per.mass = act.power / CNG.mass ' power per mass (kW/kg/hr)
END IF

'Isentropic Efficiency:

IF act.power <= .01 THEN
   isentr.eff = 999
ELSE
   isentr.eff = isentr.power.tot / act.power * 100 'isentropic efficiency in %
END IF

'Put calculated values in an array for subsequent use
'See list of Calcname$() in SUB SetCalcNames

CalcData(1) = p.ratio.1
CalcData(2) = p.ratio.2
CalcData(3) = p.ratio.total
CalcData(4) = power.per.mass
CalcData(5) = act.power
CalcData(6) = isentr.power.1
CalcData(7) = isentr.power.2
CalcData(8) = isentr.power.tot
CalcData(9) = vol.eff.tot
CalcData(10) = isentr.eff
CalcData(11) = intens.rpm
CalcData(12) = disp.1
CalcData(13) = disp.2
CalcData(14) = temp.ratio.1
CalcData(15) = temp.ratio.2
CalcData(16) = theoret.out.temp.1
CalcData(17) = theoret.out.temp.2
CalcData(18) = theoret.mass.1

END SUB

*Last edited: Dec. 6, 1993*
**Simulation Program**

***simulation program for rotary intensifier***

***dimensioning variables***

_crank angle variables_
DIM x1#(360), Vol1#(360), p1#(360), t1#(360), M1#(360)
DIM x2#(360), Vol2#(360), p2#(360), t2#(360), M2#(360)
DIM Frad1#(360), Ftan1#(360), Ftot1#(360)
DIM Frad2#(360), Ftan2#(360), Ftot2#(360)
DIM torq1#(360), torq2#(360), torqtwo#(360), torqfour#(360)
DIM Mres#(360), pres#(360), tres#(360), alpha#(360)

_revolutions variables_
DIM MFlow1#(200), MFlow2#(200), MFlowbyps#(200)

_intake pressure variables_
DIM MFlow#(20), Pmax#(20), pr1#(20), pr2#(20), prt#(20), power#(20)

***getting compressor dimensions***

stroke = (1) * 25.4
eccmax = stroke / 2  'maximum eccentricity
Rinner = (6.875) * 25.4  'distance from top of cyl to center
Router = (6.8125) * 25.4  'outer ring radius
bore1 = (1.625) * 25.4
bore2 = (1) * 25.4
rodlen = Router - Rinner + eccmax  'connecting rod length

***setting constants***

pi# = 3.141592665359#
R = 500000  'gas constant
k = 1.3  'isentropic coefficient
ncom = 1.25  'polytropic coefficient
nexp = 1.4  'polytropic coefficient
Vres# = 375000  'intercooler volume
clvol1# = pi# * 2 * (bore1 ^ 2) / 4 + 4950  'clearance volume of 1st stage
clvol2# = pi# * 2 * (bore2 ^ 2) / 4 + 1500  'clearance volume of 2nd stage
a1# = pi# * (bore1 ^ 2) / 4  '1st stage piston area
a2# = pi# * (bore2 ^ 2) / 4  '2nd stage piston area
tblk# = 320  "bulk temperature"

***default values***

Pintmin# = 2
Pintmax# = 20
nopoints# = 1
Pexh# = 20
Tint# = 290
ecc# = eccmax
rpm# = 200
drive$ = "b:\"
filename$ = "dataout.sim"
graph$ = "y"

***inputs***

CLS

PRINT "Intake tempreature (K) " , Tint#;
INPUT temp$
IF temp$ <> "" THEN Tint# = VAL(temp$)

PRINT "Minimum intake pressure (MPa)" , Pintmin#;
INPUT temp$
IF temp$ <> "" THEN Pintmin# = VAL(temp$)

PRINT "Maximum intake pressure (MPa)" , Pintmax#;
INPUT temp$
IF temp$ <> "" THEN Pintmax# = VAL(temp$)

PRINT "Exhaust pressure (MPa) " , Pexh#;
INPUT temp$
IF temp$ <> "" THEN Pexh# = VAL(temp$)

PRINT "Number of points " , nopoints#;
INPUT temp$
IF temp$ <> "" THEN nopoints# = VAL(temp$)

PRINT "Eccentricity (inches)  " , ecc# / 25.4;
INPUT temp$
IF temp$ <> "" THEN ecc# = VAL(temp$) * 25.4

PRINT "RPM " , rpm#;
INPUT temp$
IF temp$ <> "" THEN rpm# = VAL(temp$)
PRINT "data output drive", drive$;
INPUT temp$
IF temp$ <> "" THEN drive$ = temp$

PRINT "data output filename", filename$;
INPUT temp$
IF temp$ <> "" THEN filename$ = temp$

PRINT "do you want graphical output of P and T (y/n)", graph$;
INPUT temp$
IF temp$ <> "" THEN graph$ = temp$

***variable intake pressure loop (counter is m)***

m = 1
DO
  Pint# = (m - 1) * ((Pintmax# - Pintmin#) / npoints#) + Pintmin#

***first iteration guesses***

  pl#(0) = Pint#
  p2#(0) = Pint#
  pres#(0) = (bore1 / bore2) ^ 2 * Pint#
  t1#(0) = Tint#
  t2#(0) = Tint#
  tres#(0) = tblk#

***determining values of - x (distance from BDC) for every crank angle***

  i = 0
  x1#(0) = 0
  DO
    i = i + 2
    theta# = i * pi# / 180
    alpha#(i) = ecc# * SIN(theta#) / Router
    asn# = alpha#(i) + (1 / 6) * alpha#(i) ^ 3 + (3 / 40) * alpha#(i) ^ 5 + (15 / 336) * alpha#(i) ^ 7 + (105 / 3456) * alpha#(i) ^ 9 + (945 / 42240) * alpha#(i) ^ 11 + (10395 / 599040) * alpha#(i) ^ 13 + (135135 / 9676800) * alpha#(i) ^ 15 + (1051325 / 175472640) * alpha#(i) ^ 17
    x1#(i) = ((-Router * SIN(pi# - theta# - asn#)) / SIN(theta#)) + Rinnner + rodlen
    IF i = 180 THEN x1#(180) = (stroke / 2) + ecc#
    IF i = 360 THEN x1#(360) = (stroke / 2) - ecc#
    Vol1#(i) = (pi# * (stroke - x1#(i)) * (bore1 ^ 2) / 4) + clvol1#
  LOOP UNTIL i = 360
  i = 0
DO
    i = i + 2
    ii = i - 90
    IF ii <= 0 THEN ii = i + 270
    x2#(i) = x1#(ii)
    Vol2#(i) = (pi# * (stroke - x2#(i))) * (bore2 ^ 2) / 4 + clvol2#
LOOP UNTIL i = 360
Vol1#(0) = Vol1#(360)
Vol2#(0) = Vol2#(360)
Pmax#(m) = 0

M1#(0) = p1#(0) * Vol1#(0) / (R * t1#(0))
M2#(0) = p2#(0) * Vol2#(0) / (R * t2#(0))
Mres#(0) = pres#(0) * Vres# / (R * tres#(0))

***revolutions loop (counter is j)***

j = 0
DO
    j = j + 1

***crank angle loop (counter is i)***

IF graph$ = "y" THEN
SCREEN 12
VIEW (1, 1)-(638, 420), 0, 1
WINDOW (-1, 0)-(360, 50)
LINE (0, -1)-(0, 50), 7
LINE (-1, 0)-(360, 0), 7
LINE (90, -1)-(90, 1), 7
LINE (180, -1)-(180, 1), 7
LINE (270, -1)-(270, 1), 7
LINE (-1, 2)-(1, 2), 7
LINE (-1, 4)-(1, 4), 7
LINE (-1, 6)-(1, 6), 7
LINE (-1, 8)-(1, 8), 7
LINE (-1, 10)-(1, 10), 7
LINE (-1, 12)-(1, 12), 7
LINE (-1, 14)-(1, 14), 7
LINE (-1, 16)-(1, 16), 7
LINE (-1, 18)-(1, 18), 7
LINE (-1, 20)-(1, 20), 7
LINE (-1, tblk#/10)-(360, tblk#/10), 5
LINE (-1, 30)-(360, 30), 5
LINE (-1, 21)-(360, 21), 1
LOCATE 2, 5: PRINT "Temperature"
LOCATE 17.5, 5: PRINT "Pressure"
i = 0
DO
  i = i + 2

***first stage***

IF x1#(i) > x1#(i - 2) THEN
  'compression
  p1#(i) = p1#(i - 2) * (Vol1#(i - 2) / Vol1#(i)) ^ ncom
  IF p1#(i) > pres#(i - 2) THEN
    p1#(i) = pres#(i - 2)
    valve1 = 1
  ELSE
    valve1 = 0
  END IF
  t1#(i) = t1#(i - 2) * ((p1#(i) / p1#(i - 2)) ^ ((ncom - 1) / ncom))
ELSE
  'expansion
  p1#(i) = p1#(i - 2) * (Vol1#(i - 2) / Vol1#(i)) ^ (nexp)
  IF p1#(i) < Pint# THEN
    p1#(i) = Pint#
  END IF
  t1#(i) = t1#(i - 2) * ((p1#(i) / p1#(i - 2)) ^ ((nexp - 1) / nexp))
  valve1 = 0
END IF
M1#(i) = p1#(i) * Vol1#(i) / (R * t1#(i))
IF valve1 = 1 THEN
  Mres#(i) = Mres#(i - 2) + (M1#(i - 2) - M1#(i))
  tres#(i) = ((t1#(i) * (M1#(i - 2) - M1#(i))) + (tres#(i - 2) * Mres#(i))) / (Mres#(i) + M1#(i - 2) - M1#(i))
  pres#(i) = Mres#(i) * R * tres#(i) / Vres#
  mtoT1# = mtoT1# - M1#(i) + M1#(i - 2)
ELSE
  Mres#(i) = Mres#(i - 2)
  pres#(i) = pres#(i - 2)
  tres#(i) = tres#(i - 2)
END IF
IF (-M1#(i - 2) + M1#(i)) > .0000001# AND x1#(i) < x1#(i - 2) THEN
  t1#(i) = ((t1#(i - 2) * M1#(i - 2)) + Tint# * (M1#(i) - M1#(i - 2))) / M1#(i)
END IF

***first stage forces and torque***

Fr1adl#(i) = p1#(i) * a1#
Ftan1#(i) = Fr1adl#(i) * TAN(alpha#(i))
\[ F_{tot1}(i) = \sqrt{\left(F_{rad1}(i)\right)^2 + \left(F_{tan1}(i)\right)^2} \]
\[ \text{torq1}(i) = F_{tan1}(i) \times \frac{\text{Router} + \text{ecc} - x_1(i)}{1000} \]

***reservoir bypass***

IF \( \text{pres}(i) > \text{Pexh} \) THEN
\[ \text{pres}(i) = \text{Pexh} \]
\[ \text{mb} = \text{pres}(i) \times \frac{\text{Vres}}{\text{R} \times \text{tres}(i)} \]
\[ \text{mtotb} = \text{mtotb} + \text{Mres}(i) - \text{rnb} \]
\[ \text{Mres}(i) = \text{mb} \]
ENDIF

***second stage***

IF \( x_2(i) > x_2(i-2) \) THEN
'compression
\[ p_2(i) = p_2(i-2) \times \left( \frac{\text{Vol2}(i-2)}{\text{Vol2}(i)} \right) \]
IF \( p_2(i) > \text{Pexh} \) THEN
\[ p_2(i) = \text{Pexh} \]
ENDIF
\[ t_2(i) = t_2(i-2) \times \left( \frac{(p_2(i)/p_2(i-2))^\left((n\text{com}-1)/n\text{com}\right)}{(n\text{com}-1)} \right) \]
\[ \text{valve2} = 0 \]
ELSE
'expansion
\[ p_2(i) = p_2(i-2) \times \left( \frac{(\text{Vol2}(i-2)/\text{Vol2}(i))^\left(n\text{exp}\right)}{(\text{Vol2}(i-2)/\text{Vol2}(i))^\left(n\text{exp}\right)} \right) \]
IF \( p_2(i) < \text{pres}(i) \) THEN
\[ t_2(i) = t_2(i-2) \times \left( \frac{(p_2(i)/p_2(i-2))^\left((n\text{exp}-1)/n\text{exp}\right)}{(p_2(i)/p_2(i-2))^\left((n\text{exp}-1)/n\text{exp}\right)} \right) \]
\[ p_2(i) = \text{pres}(i) \]
\[ \text{valve2} = 1 \]
ELSE
IF \( \text{tres}(0) \times \frac{(\text{Vol2}(90)/\text{Vol2}(270))^\left(n\text{com}-1\right)}{(\text{Vol2}(90)/\text{Vol2}(270))^\left(n\text{com}-1\right)} > t_2(270) \)
THEN
\[ t# = t_2(270) \]
ELSE
\[ t# = \text{tres}(0) \times \frac{(\text{Vol2}(90)/\text{Vol2}(270))^\left(n\text{com}-1\right)}{(\text{Vol2}(90)/\text{Vol2}(270))^\left(n\text{com}-1\right)} \]
ENDIF
\[ t_2(i) = t_2(i-2) \times \left( \frac{\text{Vol2}(i-2)/\text{Vol2}(i)}{(n\text{exp}-1)} \right) \]
\[ \text{valve2} = 0 \]
ENDIF
ENDIF
\[ M_{2}(i) = p_2(i) \times \frac{\text{Vol2}(i)}{(R \times t_2(i))} \]
IF \( \text{valve2} = 1 \) AND \( x_2(i) < x_2(i-2) \) THEN
\[ \text{Mres}(i) = \text{Mres}(i) - (M_{2}(i) - M_{2}(i-2)) \]
\[ \text{pres}(i) = \text{Mres}(i) \times R \times \text{tres}(i) / \text{Vres} \]
\[ \text{mtot2} = \text{mtot2} + M_{2}(i) - M_{2}(i-2) \]
\[ t_2(i) = \left( \left( t_2(i-2) \times M_{2}(i-2) \right) + \left( \text{tres}(i) \times (M_{2}(i) - M_{2}(i-2)) \right) \right) / M_{2}(i) \]
END IF

***Second stage forces and torque***

\[ ii = i - 90 \]
\[ \text{IF } ii \leq 0 \text{ THEN } ii = i + 270 \]
\[ \text{Frad2}(i) = p2(i) \times a2 \]
\[ \text{Ftan2}(i) = \text{Frad2}(i) \times \tan(\alpha(i)) \]
\[ \text{Ftot2}(i) = \sqrt{\text{Frad2}(i)^2 + \text{Ftan2}(i)^2} \]
\[ \text{torq2}(i) = \text{Ftan2}(i) \times \left(\frac{\text{Router} - \text{ecc} \times x2(i)}{1000}\right) \]
\[ \text{torqtwo}(i) = \text{torql}(i) + \text{torq2}(i) \]
\[ iii = i - 180 \]
\[ \text{IF } iii \leq 0 \text{ THEN } iii = i + 180 \]
\[ \text{torqfour}(i) = \frac{\text{torqtwo}(i)}{\text{torqtwo}(iii)} \]
\[ \text{torqavg} = \text{torqavg} + \text{torqfour}(i) \]

***Graphing***

IF graph$ = "y" \text{ THEN}\n\[ \text{PSET}(i, \text{t1}(i)/10), 11 \]
\[ \text{PSET}(i, \text{t2}(i)/10), 12 \]
\[ \text{PSET}(i, \text{tres}(i)/10), 10 \]
\[ \text{PSET}(i, \text{p1}(i)), 11 \]
\[ \text{PSET}(i, \text{p2}(i)), 12 \]
\[ \text{PSET}(i, \text{pres}(i)), 10 \]
ENDIF

***Heat transfer from bulk***

IF \( t1(i) < \text{tblk} \) THEN
\[ t1(i) = t1(i) + 0.005 \times (\text{tblk} - t1(i)) \]
\[ p1(i) = (M1(i) \times R \times t1(i)) / \text{Vol1}(i) \]
ENDIF
IF \( t2(i) < \text{tblk} \) THEN
\[ t2(i) = t2(i) + 0.005 \times (\text{tblk} - t2(i)) \]
\[ p2(i) = (M2(i) \times R \times t2(i)) / \text{Vol2}(i) \]
ENDIF
IF \( \text{tres}(i) \leq \text{tblk} \) THEN
\[ \text{tres}(i) = \text{tres}(i) + 0.005 \times (\text{tblk} - \text{tres}(i)) \]
\[ \text{pres}(i) = (Mres(i) \times R \times \text{tres}(i)) / \text{Vres} \]
ENDIF

***End off crank angle loop***

LOOP UNTIL \( i = 360 \)

***Mass flow rate calculation***
MFlow1#(j) = mtoT1# * 2 * rpm# * 60
MFlow2#(j) = mtoT2# * 2 * rpm# * 60
MFlowbyps#(j) = mtotb# * 2 * rpm# * 60
MFlow#(m) = (MFlow1#(j) + MFlow2#(j) + MFlowbyps#(j)) / 2
trqavg# = torqavg# / 180
power#(m) = trqavg# * rpm# * 2 * pi# / 60 / 1000

torqavg# = 0
mtoT1# = 0
mtoT2# = 0
mtotb# = 0

***continuity***

t1#(0) = t1#(360)
p1#(0) = p1#(360)
M1#(0) = M1#(360)
t2#(0) = t2#(360)
p2#(0) = p2#(360)
M2#(0) = M2#(360)
Mres#(0) = Mres#(360)
pres#(0) = pres#(360)
tres#(0) = tres#(360)

***verifying equilibrium criterion and max flow rate req't***

IF ABS(MFlow1#(j) - MFlow2#(j) - MFlowbyps#(j)) < .2 AND j > 10 THEN
    j = 120
ENDIF

***end of revolution loop***

LOOP UNTIL j = 120

***calculation of output data***

pintersta# = 0
i = 0
DO
    i = i + 2
    pintersta# = pintersta# + pres#(i)
LOOP UNTIL i = 360
pintersta# = pintersta# / 180
prt#(m) = Pexh# / Pint#
pr1#(m) = pintersta# / Pint#
pr2#(m) = Pexh# / pintersta#
***end of intake pressure loop***

    m = m + 1
LOOP UNTIL m = (nopoints# + 1)

***data output to disk***

OPEN drive$ + filename$ FOR OUTPUT AS #2
m = 1
DO
    PRINT #2, prt#(20), pr1#(m), pr2#(m), MFlow#(m), power#(m)
    m = m + 1
LOOP UNTIL m = (nopoints# + 1)
CLOSE #2
END