THE SYNTHESIS OF A FREE-PISTON POWER SAW

by

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The object of this study was to apply the technique of synthesis to the design of a small power saw. The study produced experimental data on optimum chain speeds, engine vibrations, noise levels, and heat transfer coefficients for reciprocating cylinder heads, and led to a simple free-piston configuration in which a piston oscillated between a mixture of air and fuel in one end of a closed cylinder and a spring in the other. The feasibility of developing the configuration into a practical reciprocating engine was verified by designing, building and testing a prototype.

The prototype incorporated such novel features as instant, effortless starting and stopping, automatic throttling, self-cooling, compression ignition of a carbureted air-fuel mixture, and a balanced engine. Uncontrolled ignition timing reduced engine efficiency, and the lack of inertia made engine stalling easy and carburetor adjustment difficult.

The computed results suggest that a developed 3 lb free-piston power saw will produce 1.0 hp at 6,400 cpm and have a specific fuel consumption of .9 lb/shp-hr.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Chapter</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. FORMULATION</td>
<td>1</td>
</tr>
<tr>
<td>1.1 Introduction</td>
<td>1</td>
</tr>
<tr>
<td>1.2 Specification for the Design Envelope</td>
<td>26</td>
</tr>
<tr>
<td>2. SYNTHESIS</td>
<td>63</td>
</tr>
<tr>
<td>2.1 Wood Cutting Devices</td>
<td>63</td>
</tr>
<tr>
<td>2.2 Evaluation of Existing Engines</td>
<td>77</td>
</tr>
<tr>
<td>2.3 Synthesis of Alternatives</td>
<td>91</td>
</tr>
<tr>
<td>2.4 Synthesis of the Free-piston power Saw</td>
<td>107</td>
</tr>
<tr>
<td>3. DETAILING</td>
<td>118</td>
</tr>
<tr>
<td>3.1 Dimensional Analysis</td>
<td>118</td>
</tr>
<tr>
<td>3.2 Automatic Throttling</td>
<td>145</td>
</tr>
<tr>
<td>3.3 Design of Components</td>
<td>156</td>
</tr>
<tr>
<td>3.3.1 General Considerations</td>
<td>156</td>
</tr>
<tr>
<td>3.3.2 Bounce Spring</td>
<td>158</td>
</tr>
<tr>
<td>3.3.3 Synchronizing Mechanism</td>
<td>172</td>
</tr>
<tr>
<td>3.3.4 Arresting Mechanism</td>
<td>184</td>
</tr>
<tr>
<td>3.3.5 Cooling System</td>
<td>190</td>
</tr>
<tr>
<td>3.3.6 Combustion System</td>
<td>197</td>
</tr>
<tr>
<td>3.4 Fabrication of Parts</td>
<td>206</td>
</tr>
<tr>
<td>4. EVALUATION</td>
<td>216</td>
</tr>
<tr>
<td>4.1 Performance Characteristics of the Free-piston Saw</td>
<td>216</td>
</tr>
</tbody>
</table>
4.2 Conclusions ................................... 237
4.3 Summary ...................................... 241
REFERENCES ...................................... 247

APPENDICES

I Cutting Speed Test Data ................. 257
II Chain Saw Vibration Test Data ......... 258
III Chain Saw Noise Test Data ............ 259
IV Data from the Questionnaire on the Use of Power Saws .................. 260
V Typical Power Saw Performance Data . . . 261
VI Data for Heat Transfer from Reciprocating Heads ...................... 262
VII FPS Prototype Test Data ................. 263
### LIST OF TABLES

<table>
<thead>
<tr>
<th>Table</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>Specifications for a small power saw</td>
<td>61</td>
</tr>
<tr>
<td>II</td>
<td>Characteristics of the conventional and Wankel engines</td>
<td>81</td>
</tr>
<tr>
<td>III</td>
<td>Scaling characteristics for similar engines</td>
<td>129</td>
</tr>
<tr>
<td>IV</td>
<td>Size of material required to store 175 in-lb of energy</td>
<td>166</td>
</tr>
<tr>
<td>V</td>
<td>Capacity and optimum size of bands</td>
<td>179</td>
</tr>
<tr>
<td>VI</td>
<td>Materials suitable for cylinder heads and blocks</td>
<td>197</td>
</tr>
<tr>
<td>VII</td>
<td>Materials suitable for pistons</td>
<td>202</td>
</tr>
<tr>
<td>VIII</td>
<td>Performance characteristics of the free-piston power saw</td>
<td>237</td>
</tr>
</tbody>
</table>
### LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1</td>
<td>Outline of the comprehensive design engineers task</td>
<td>10</td>
</tr>
<tr>
<td>1.2</td>
<td>Cutting with one of the original one-man chain saws</td>
<td>17</td>
</tr>
<tr>
<td>1.3</td>
<td>Sketches of the chisel-tooth and chipper-tooth chains</td>
<td>18</td>
</tr>
<tr>
<td>1.4</td>
<td>Typical list of chain saw specifications, circ. 1950</td>
<td>19</td>
</tr>
<tr>
<td>1.5</td>
<td>Typical tree harvesting machine</td>
<td>24</td>
</tr>
<tr>
<td>1.6</td>
<td>Apparatus used in chain speed tests</td>
<td>33</td>
</tr>
<tr>
<td>1.7</td>
<td>Schematic representation of muffler and electrical equivalent</td>
<td>53</td>
</tr>
<tr>
<td>1.8</td>
<td>Accidents reported to Quebec Pulp and Paper Association</td>
<td>56</td>
</tr>
<tr>
<td>1.9</td>
<td>Results of questionnaire on the importance of saw characteristics</td>
<td>60</td>
</tr>
<tr>
<td>2.1</td>
<td>Diagrams illustrating hot-gas cycle</td>
<td>85</td>
</tr>
<tr>
<td>2.2</td>
<td>Stirling thermal engine schematic drawing</td>
<td>88</td>
</tr>
<tr>
<td>2.3</td>
<td>Sketches of the unbalanced lever engine</td>
<td>94</td>
</tr>
<tr>
<td>2.4</td>
<td>A sketch of the oscillating free-piston engine</td>
<td>101</td>
</tr>
<tr>
<td>2.5</td>
<td>Typical reciprocating-blade saws</td>
<td>115</td>
</tr>
<tr>
<td>2.6</td>
<td>Existing free-piston configurations</td>
<td>117</td>
</tr>
<tr>
<td>3.1</td>
<td>Free body diagram of the free-piston power saw</td>
<td>157</td>
</tr>
<tr>
<td>3.2</td>
<td>Photographs of heat transfer test apparatus</td>
<td>195</td>
</tr>
<tr>
<td>3.3</td>
<td>Free-piston power saw assembly sketch</td>
<td>207</td>
</tr>
<tr>
<td>Figure</td>
<td>Description</td>
<td>Page</td>
</tr>
<tr>
<td>--------</td>
<td>-----------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>3.4</td>
<td>Photograph of the transfer port machining operation</td>
<td>209</td>
</tr>
<tr>
<td>3.5</td>
<td>Photographs of synchronizing and arresting mechanism assemblies</td>
<td>212</td>
</tr>
<tr>
<td>3.6</td>
<td>Photographs of FPS components</td>
<td>214</td>
</tr>
<tr>
<td>3.7</td>
<td>Photograph of all FPS parts</td>
<td>215</td>
</tr>
<tr>
<td>4.1</td>
<td>Photographs of the instrumented FPS</td>
<td>222</td>
</tr>
<tr>
<td>4.2</td>
<td>Photographs of the FPS, with a fixed-throw crankshaft</td>
<td>222</td>
</tr>
<tr>
<td>4.3</td>
<td>Photographs of the FPS with an oscillating crankshaft</td>
<td>230</td>
</tr>
<tr>
<td>4.4</td>
<td>Subassembly photographs of a conventional power saw and the FPS</td>
<td>239</td>
</tr>
</tbody>
</table>
## LIST OF GRAPHS

<table>
<thead>
<tr>
<th>Graph</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1 Power saw performance trends</td>
<td>22</td>
</tr>
<tr>
<td>1.2 Optimum power as a function of tree size</td>
<td>31</td>
</tr>
<tr>
<td>1.3 Importance of sprocket size on cutting rates</td>
<td>35</td>
</tr>
<tr>
<td>1.4 Importance of joint on cutting rates</td>
<td>35</td>
</tr>
<tr>
<td>1.5 Cutting rate variations</td>
<td>35</td>
</tr>
<tr>
<td>1.6 Importance of bar length on cutting rates</td>
<td>35</td>
</tr>
<tr>
<td>1.7 Importance of wood species on cutting rates</td>
<td>36</td>
</tr>
<tr>
<td>1.8 Gear drive cutting rates</td>
<td>36</td>
</tr>
<tr>
<td>1.9 Specific energy as a function of chain speed</td>
<td>36</td>
</tr>
<tr>
<td>1.10 Vibration damage levels as a function of frequency</td>
<td>40</td>
</tr>
<tr>
<td>1.11 Vibration amplitude while cutting versus counterweight size</td>
<td>44</td>
</tr>
<tr>
<td>1.12 Vibration amplitude while cutting and while unloaded</td>
<td>44</td>
</tr>
<tr>
<td>1.13 Effect of bar on amplitude with several different counterweights</td>
<td>45</td>
</tr>
<tr>
<td>1.14 Effect of bar on vibration amplitude</td>
<td>46</td>
</tr>
<tr>
<td>1.15 Effect of speed on vibration amplitude</td>
<td>46</td>
</tr>
<tr>
<td>1.16 Effect of rubber insulation on vibration amplitude</td>
<td>47</td>
</tr>
<tr>
<td>1.17 Noise level readings of typical power saws</td>
<td>50</td>
</tr>
<tr>
<td>Graph</td>
<td>Description</td>
</tr>
<tr>
<td>-------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>1.18</td>
<td>State of Washington standard for industrial noise</td>
</tr>
<tr>
<td>1.19</td>
<td>Tentative Swedish noise level limits for power saws</td>
</tr>
<tr>
<td>2.1</td>
<td>Specific energy required by various wood cutting devices</td>
</tr>
<tr>
<td>2.2</td>
<td>Power of a Wankel engine compared with conventional engines</td>
</tr>
<tr>
<td>2.3</td>
<td>Computed piston position for the unbalanced lever engine</td>
</tr>
<tr>
<td>2.4</td>
<td>Blade and piston position as a function of compression ratio</td>
</tr>
<tr>
<td>3.1</td>
<td>Power as a function of piston area for typical power saws</td>
</tr>
<tr>
<td>3.2</td>
<td>Specific power as a function of bore size</td>
</tr>
<tr>
<td>3.3</td>
<td>Specific weight as a function of bore size</td>
</tr>
<tr>
<td>3.4</td>
<td>Specific power as a function of piston speed</td>
</tr>
<tr>
<td>3.5</td>
<td>Specific power as a function of engine speed</td>
</tr>
<tr>
<td>3.6</td>
<td>Port areas of typical power saws versus stroke/bore ratios</td>
</tr>
<tr>
<td>3.7</td>
<td>Port heights versus square root of stroke/bore ratios</td>
</tr>
<tr>
<td>3.8</td>
<td>Specific power and BMEP versus port area ratios</td>
</tr>
<tr>
<td>3.9</td>
<td>Flow area versus stroke for ideal FPS</td>
</tr>
<tr>
<td>3.10</td>
<td>Piston strokes as a function of time for ideal FPS</td>
</tr>
<tr>
<td>3.11</td>
<td>Stroke variation caused by sudden load changes</td>
</tr>
<tr>
<td>Graph</td>
<td>Page</td>
</tr>
<tr>
<td>----------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>3.12 Maximum load transmitted by bands</td>
<td>181</td>
</tr>
<tr>
<td>3.13 Heat transfer curve for reciprocating cylinder heads</td>
<td>196</td>
</tr>
<tr>
<td>3.14 Calculated thermal losses from combustion chambers</td>
<td>200</td>
</tr>
<tr>
<td>4.1 Prototype engine traces (experimental)</td>
<td>223</td>
</tr>
<tr>
<td>4.2 Prototype engine traces without combustion (computed)</td>
<td>224</td>
</tr>
<tr>
<td>4.3 Effect of leakage, friction and damping on traces (computed)</td>
<td>225</td>
</tr>
<tr>
<td>4.4 Prototype engine traces with combustion</td>
<td>226</td>
</tr>
<tr>
<td>4.5 Crank engine performance (experimental and computed)</td>
<td>229</td>
</tr>
<tr>
<td>4.6 Experimental engine position traces (experimental)</td>
<td>232</td>
</tr>
<tr>
<td>4.7 Experimental engine position traces (computed)</td>
<td>233</td>
</tr>
</tbody>
</table>
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1. FORMULATION

1.1 Introduction

This study evolved a configuration for a small prime mover to be used in a class of applications such as power saws. Starting with a formulation of the basic power saw requirements, the study continued with a synthesis of several alternatives to the conventional reciprocating engine. Synthesis was followed with an optimization of the most promising alternative and a detail design of the components. The study concluded with a verification of the idea by testing an actual prototype. The end result was a light-weight, vibrationless, quick-starting, self-throttling, free-piston machine that departed "radically" from existing power saws.

In the configuration optimized and built, a piston bounces between a spring in one end of a closed cylinder and a volume of fuel and air in the other end. A blade is attached directly to the oscillating piston. By igniting the mixture during the compression stroke, a simple prime mover is theoretically possible.

Engineering design is an iterative decision-making process applying scientific principles and utilizing practical techniques in order to define a device, a process, or a system in sufficient detail to permit its physical real-
ization [1.1, 1.2]. A good design permits manufacture by the most economical method and in the shortest possible time and results in a functional, often patentable product that not only meets the stipulated conditions but also incorporates aesthetic appeal [1.3, 1.4, 1.5]. A good creatively-designed product is a unique, often startling combination of engineering principles and known data. The device is useful and beneficial and very likely to be dramatic, spectacular and newsworthy. It satisfies the maker and fascinates the user [1.6, 1.7].

A good designer is a generalist who is motivated by very broad concepts of human activity, thought and behaviour. He communicates well. He understands the creative process. He maintains a delicate balance between his ability to synthesize, to analyze, and to evaluate. He is thoroughly familiar with the environment in which his product will be made, sold, used and serviced. Arnold [1.8] called this generalist a "comprehensive designer".

Although each design project has its own history, the sequence of events common to all projects forms a pattern that can be studied profitably. By examining this pattern engineers have obtained an insight into the methodology of design by which thoughts about needs are

*References in parenthesis refer to Bibliography at end of this thesis.
projected into ideas about things [1.9, 1.10].

The pattern leads to the standard recipe: analyze, theorize, delineate and modify [1.11]. The use of this recipe results in a conventional solution that is usually physically realizable, economically worthwhile, and financially feasible but not necessarily the best one possible.

To arrive at a more optimal solution, especially to a complex problem, it is necessary to add a creative or inventive process to the recipe. The inventive process has the following steps [1.12, 1.13, 1.14, 1.15]:

1. preparation - gathering skills and formulating the problem,
2. perspiration - thinking deliberately and intensely about the problem,
3. incubation - a period of mental rest after deliberate thinking has failed,
4. inspiration - the sudden idea or reorganization which is the solution,
5. verification - following through with generalizations, evaluations and elaborations.

As the scope of his creative work depends upon his store of knowledge, the progressive designer endeavours to be perceptive and to enlarge his viewpoints through study, experience, and observation. By being aware of his emotional blocks and conscious of his predisposition to
particular images, methods or way of thinking, he overcomes his biggest obstacle to originality because ability to recall the right images and the ability to modify images effectively are two essential ingredients of creativity [1.12, 1.16].

In contrast to the creative mental processes are these non-creative processes [1.15]:

1. observation - studying perceived objects and circumstances,
2. reflection - reviewing the content of the mind,
3. remembering - recalling past experiences and previously acquired ideas,
4. reasoning - determining the consequences of assumed conditions and courses of action,
5. judgment - formulating decisions.

But no new solutions are possible without stimulation. What stimulates us? Niemann [1.17] suggests excitement over a new phenomenon, a new realization, a new requirement, the fruitful anger over an incomplete thing or the lively difference of opinion with other experienced people. Less influential, but nevertheless satisfactory, is the stimulation that comes through reading with an alert attitude and the unrest that comes through criticism of the present, by presentation of new points of view and by hearing others express their wishes.
Creativity is only a fairly recent inclusion in engineering courses. Perhaps the earliest formulation of the creative design process was for General Electric Company's "Creative Engineering Program" established in 1937 [1.18, 1.19]. The following steps were formulated:

1. Definition of the problem,
2. Manipulation of elements bearing on solution.
3. Period resulting in the intuitive idea,
4. The idea is shaped to practical usefulness.

[1.19, p. 9]

From about 1940 to 1960, creativity was emphasized in the hope that better technological designs would result. Roadblocks to creative activity were investigated not only by designers but by authors generally. Erich Fromm [1.20] clearly demonstrates that although each person has a strong need for security, to be a member of a group, to belong to something, he expresses this need positively or negatively. He becomes a free productive individual or he forms psychological blocks. These blocks form filters that distort the information he receives from the outside world, inhibit free association within his brain, and prevent clear communication to others.

In the late fifties and early sixties, thoughtful individuals began to realize that there was considerably more to the design process than analysis or even creativity. Observing the actions of his fellow man, Tielhard de Chardin [1.21] felt that man is obsessed by the need to depersonal-
ize all that he most admires, partly because of

analysis, that marvellous instrument of scientific research to which we owe all our advances but which, breaking down synthesis after synthesis, allows one soul after another to escape, leaving us confronted with a pile of dismantled machinery. [1.21, p. 283]

Arnold [1.7, 1.18] was perhaps one of the first engineers to recognize the need for a more comprehensive view of the design process when in 1959 he included such topics as psychology of the mind, aesthetics, decision theory and operations research in his theory of design.

This comprehensive view has now become an important concept in the philosophy of design. The solution to the rapidly changing and increasingly complex problems of engineering design requires a flexible approach. To achieve flexibility yet at the same time to arrive at a concrete solution, the engineer is encouraged to generate several alternatives, choose the best, and then optimize the configuration using rational techniques. Information feedback continually challenges his decisions, making the design process not an "open information loop" system where the solution hypothetically does not affect the environment, but a "socio-technical feedback" system where the solution anticipates the dynamic interaction of design, man and environment [1.22]. The dynamic feedback system greatly increases the number of variables and the number of decisions required. The result is that "the skill of the designer is
measured by his ability to identify limits and to make appropriate compromises" [1.21, p. 9].

Even as optimization techniques, decision theory, systems analysis, operations research, cybernetic imagination, value engineering, reliability engineering and program evaluation review techniques are being investigated and applied, some social scientists are considering the effect that too much emphasis on methods has on the individual because the process of synthesis is essentially individualistic. Jacques Ellul [1.23] suggests that optimization techniques when applied to managerial organizations lead to standardization and rationalization of economic and administrative life. He quotes Antoine Mas to expand his point:

> standardization means resolving in advance all the problems that might possibly impede the functioning of an organization. It is not a matter of leaving it to inspiration, ingenuity, nor even intelligence to find a solution at the moment some difficulty arises; it is rather in some way to anticipate both the difficulty and its resolution. From then on standardization creates impersonality, in the sense that organization relies more on methods and instructions than on individuals. [1.23, P. 11]

But some engineering educators are optimistic about the changing nature of engineering design. Thimm [1.22] believes that:

> systematic training in operations research and decision theory will improve overall engineering performance; it might even save the world from technological pollution and ecological catastrophe. [1.22, p. 12]
Society is expecting the designer to take more responsibility not only against errors but also for effluents of an unhealthy nature. At the third annual McMaster University design seminar, F.R. Duncan [1.24] closed his remarks on the legal responsibility of the designer with the remark that, "Its just not safe to leave design errors floating around any more" [1.24, p. 36]. At the same seminar Allcut [1.25] mentioned that while ethics relate entirely to people and design deals only with things, the relationship between the two is essentially that of cause and effect and "considerable judgment based on experience is required to combine these two factors in the right proportions" [1.25, p. 38].

The emphasis on creativity is associated with the Gestalt theory of perception developed around the 1920's [1.26]. The crux of the theory is that it is important to examine the totality of a problem, not merely its components. In perception the whole is greater than the sum of its parts. To perceive the totality or implication of the solution requires a "total" approach to problem solving.

What is called the "total" approach is an alliance between intellect and intuition, between vertical thinking based on logic leading to a conventional solution and lateral thinking based on intuitive awareness leading to radical solutions. This approach integrates what C.P. Snow calls the "two cultures" [1.27] and leads to new discoveries,
unusual solutions, and comprehensive understanding of the problems.

In the "total" approach to creative design, illustrated on Figure 1.1, the engineer goes through five stages [1.28, 1.2]:

1. Problem formulation,
2. Concept synthesis,
3. Configuration optimization,
4. Element design, and
5. Performance evaluation.

If it is initiated after the problem is clearly understood and the design envelope is explicitly formulated, synthesis can lead to a number of concepts which tentatively satisfy the requirements of function, environment and economy. Each of the concepts is subjected to critical scientific analysis and judged on sound engineering principles. For the most promising concept, an optimum configuration is devised and the elements of the configuration are designed using the best engineering practice. Finally a working model is constructed and the performance evaluated. Throughout the design, feedback is used to correct decisions made at any of the earlier stages.

In applying the five steps, conflicts occur between creative synthesis and logical analysis. One difficulty is that the creative step requires a breadth of knowledge and understanding and an ability to relate diverse elements,
Experimental General Knowledge Personal Experience Field Study

Analytic Ability

Formulation of Design Envelope

Scientific Principles

Creative Ability

Concept 1

Method 2 Method 3

Concept 2

Method 1 Method 2

Step 1. Formulation

Step 2. Synthesis

Step 3. Optimization

Step 4. Delineation

Step 5. Evaluation

Figure 1.1 Outline of the Comprehensive Design Engineer's Task
whereas the analytic step requires a depth of specialized knowledge, an ability to use mathematics, and an ability to recognize and remember specific facts. To be effective the creative engineer must be able to oscillate freely between the creative step and analytical step, between imagination and reason. Imagination evolves new combinations of ideas and reason evaluates each combination.

In order to be creative and evolve new ideas the mind must be free to alternate between all aspects of the problem whereas in order to evaluate logically the mind must not depart from a systematic step-by-step sequence. This conflict between creative synthesis and logical analysis is resolved by allowing the mind complete freedom to produce ideas, solutions, possibilities and guess work at any time while employing a system of notation to record all design information for logical analysis. Thus imagination is unrestricted in the mind and logic is preserved on paper. Once started, the system of notation must be used flexibly as a guide, not dogmatically as a ritual.

One engineering heuristics technique used to generate solutions to problems is to postulate a generalized model of the design process in the form of a "tree" [1.28]. Each solution to some design concept will, in general, give rise to a series of subsidiary problems each of which has (usually) more than one possible solution.
The design is complete when all branches of the tree terminate. This happens when the following rules are followed:

1. When a number of alternative solutions are presented, any one may be accepted and the rest ignored.
2. All the problems dependent on the choice of a particular alternative solution, must, however, be solved.
3. A particular branch of the tree must be followed until a solution is reached which does not have a dependent problem. Provided that this solution is the preferred one at this point, the branch terminates. [1.28, p. 54]

Factor analysis is another interesting technique which attempts to solve the basic difficulty of combining logical analysis with creative thought [1.28, 1.29, 1.30]. This technique requires a table or matrix where all parameters concerned with features or functions desired are listed vertically. All possible means of achieving each function are listed horizontally with reference to the conflicting demands of the other functions.

The partial solutions are then combined by permutation to give several alternative whole solutions. This procedure is the reverse of conventional design methods in which a single solution is conceived as a whole with the details being worked out later.

As the work proceeds, many more ideas are added. Incomplete, questionable or conflicting information is substantiated by a literature search, by consulting experienced persons or by performing experiments. New ideas are encour-
aged by looking at existing designs, viewing the problem from a new viewpoint, comparing the problem with other solutions, identifying with the object being designed, fantasizing an ideal solution, using free association, and brainstorming [1.14, 1.28, 1.31]. This latter method is best suited to a design team with well defined responsibilities and occurs when each person uncritically records any idea or solution that occurs to him after being confronted with the problem, or after looking at examples, drawings and reports of existing designs.

Relationships between solutions can be clarified with trend plots or new solution plots. A trend plot shows how shape and performance have changed over the years together with the reason for the changes. A new solution plot is a means of comparing the range of existing solutions, or new proposals in relation to shape or performance. These plots can be used to find areas where new combinations of shape and performance can be sought. More logical and careful thought is given to aspects of the problem where no solutions have appeared. If the imagination comes to a halt and no solution seems possible, Jones suggests the following procedure:

1. Write down the conditions which would make a solution possible,
2. Write down a phrase describing the difficulty and substitute alternatives for each word,
3. Write down the consequences of not finding a solution.

[1.28, p. 64]
Because it safeguards against wasting time on a system that is not feasible or goes outside the design envelope, a continuous check that the possible solutions are indeed acceptable is kept up in parallel with all other design activity. If an unacceptable solution is detected work on it ceases.

From among the alternatives synthesized the best one is chosen for optimization and detail designing. The optimum design is only achieved when experience and ingenuity are blended with design philosophy, material selection and the production methods available. This blending initially involves rough sketches and order-of-magnitude calculations and finally uses detailed analysis.

The facts of life in a mass-production economy make the introduction of new concepts very costly and risky. It is natural to prefer stability and to seek it. This preference for stability means that innovations and inventions, if they are to become production items, will have to fit in with the current market capabilities and the established means of production, maintenance and servicing as well as financial resources available [1.4, 1.32].

Where it is possible to vary the properties of the elements or of inputs or some aspects of environment so as to effect the properties of the system, it is possible to choose a combination of variables which yield the best system performance. For the case in which a mathematical relationship connects overall cost and the variables to be optimized, cal-
culus, linear programming or dynamic programming methods are possible. Where variables are not mathematically connected, other techniques to find the best "strategy of search", that is, the pattern of moves likely to lead most quickly to the summit, are used [1.33].

The crucial elements and uncertain features of the optimum solution are identified and an attempt is made to determine how the design can be modified to yield a workable system in case the crucial elements cannot be realized. The result will be a hierarchy of possible designs, starting with the most favourable and proceeding to less favourable alternative arrangements.

In cases where a certain crucial element has few or no alternative realizations so that no adequate path of retreat is available in the event that the element proves unavailable experimental work is initiated in order to demonstrate the feasibility of this element. This is to safeguard against failure of the whole design at a later date. It is unwise to work on some branches of the design tree to the point of fine detail while there are still unresolved problems at a 'higher' (that is, more abstract) level, although for small projects, the information required may be obtained from tests on an actual prototype.

The design tree and other approaches to systematic design can be profitably studied and applied in engineering design courses. Typical of the creative design projects under-
taken on a graduate level is de Pencier's Ph.D. thesis, *Genesis of a Machine: A Production Paper Cutter* [1.34]. In his thesis he describes the evolution of the machine from an idea to an operating prototype:

The emphasis is on the decisions, methods, criteria, and results incorporated in the machine rather than on analytical aspects of the design. The primary criteria guiding the design was to provide a machine more economical for users to own.

[Abstract, ref. (1.34)]

The usual industrial approach to small engine design is not systematic but by trial and error. Solutions are sought only to urgent problems without a study of the totality of the design. Pressure is exerted on the designer to get specific solutions immediately. Consequently concepts change slowly and breakthroughs seldom occur. Such has been the case with the development of the power saw. When the first German saws were imported into British Columbia in 1937, they weighed about 120 lbs and produced about 5 hp (rated at 8 hp). When World War II cut the supply of saws and spare parts and the dealer, D.J. Smith Equipment, found it necessary to make his own units, he copied the imported saw. In 1943 he brought out a 90 lb version and then a limited number of one-man chain saws. By 1945 the dealer had become Industrial Engineering Limited, Power Machinery Limited had formed, and Vancouver was the center of the North American chain saw industry.
When they brought out the successful one-man chain saw shown on Figure 1.2, Power Machinery Limited became a strong competitor with Industrial Engineering Limited for the power saw market. With competition came more rapid progress. The problem areas on existing saws were investigated by both companies independently. The manual rope and pulley starting system was replaced with an automatic recoil rewind mechanism in 1946.
When the Cox chipper chain came on the market in 1948, its value in reducing friction and saw blade loading was immediately recognized. Whereas the standard chain required an external force to feed the teeth much like the force required to cut with the conventional table saw, the chipper chain required no such force, because the self-feeding action originated from the shape of the teeth, as shown on Figure 1.3.

Figure 1.3 Sketches of the chisel-tooth (left) and the chipper-tooth (right) chains

1.3. Because the self-feeding reduced bar and chain friction, the need for a foolproof automatic oiling system, in use since about 1946, was not as critical.

The self-energizing centrifugal clutch was added to the saws in 1949. This clutch automatically engaged when the engine speed reached a predetermined level and disengaged when the speed dropped below this level; the amount of pre-
compression of the springs acting against centrifugal force determined the engagement speed.

By this time some attention was given to aesthetics. As well as being painted for the first time, the units were shaped more attractively. Also emphasized were the automatic systems, as shown by the last four items of a typical list of specifications, Figure 1.4.

**SPECIFICATIONS**

- **Motor:** Single cylinder, 2 cycle, air cooled
- **Power:** 4 H.P. at 4000 R.P.M.
- **Cylinder:** Aluminum, chrome plated and honed
- **Crankshaft:** Forged alloy steel, machined, heat-treated and precision ground.
- **Connecting Rod:** Same as above
- **Piston:** Aluminum - 3 ring
- **Carburetor:** Float type - Tillotson
- **Ignition:** Flywheel type - Wico
- **Lubrication:** Oil and gasoline mixed
- **Main Bearings:** Ball and needle - Standard makes
- **Connecting Rod Bearings:** Needle and Bronze
- **Guide Bar:** Alloy steel, heat treated, hard tipped
- **Cutting Chain:** Alloy steel, heat treated, chipper type
- **General Construction:** Cast Magnesium
- **Starting Mechanism:** Automatic recoil
- **Drive:** Gilmer Belt - no lubrication necessary
- **Clutch:** Automatic
- **Chain Oiler:** Automatic

*Specifications for the P.M. "Rocket" chain saw, ref. [1.35].*
they eliminated the need for Gilmer belts, chain drives, bevel gears, spur gears or other speed reducers.

When Industrial Engineering Limited brought out their direct drive chainsaw, Power Machinery Limited was behind in their chain technology, having supported the development of a Tillotson all-position carburetor. The lead gave Industrial Engineering Limited a name and a market which their competition could not match even with the introduction of an all-position carburetor a year later.

The all position diaphragm carburetor with a built-in diaphragm fuel pump ended many fuel metering problems. When the Tillotson diaphragm fuel pump became available in June 1954, Power Machinery Limited [1.36] announced that the new carburetor would be sold at the option of the buyer. By November the new carburetor was standard equipment.

The quick changeover can be better understood if one considers the problems encountered with the two types of older carburetors—the float type using gravity feed and the diaphragm type using a pressurized fuel tank. The float type carburetor operated only when upright; in one design this requirement was met by manually swivelling the carburetor or when the engine was turned for bucking, and in the other design by manually swivelling the blade and chain while keeping the engine upright. The pressurized tank type carburetor required continual adjustment because the inlet and outlet check valves controlling the tank pressure fre-
quently plugged or leaked, causing the pressure to exceed or fall below the proper working pressure.

Since the introduction of the all position carburetor no major change has occurred, although continual refinement of design, materials, and manufacturing techniques have reduced the weight of the saw and increased the speed of the engine. In 1945 when they were introduced, the one-man saws weighed about 34 lbs and produced about 2 1/2 hp at 3800 rpm from a 4.5 in$^3$ displacement. Ten years later a 22 lb saw produced 4 hp at 6000 rpm from a 6.2 in$^3$ displacement. In 1963 a 16 lb saw produced 4 1/2 hp at 7500 rpm from a 5.8 in$^3$ displacement. Now in 1969 the small 6 1/2 lb saws produce 2 hp at 9000 rpm from a 2.8 in$^3$ displacement. These trends, plotted on Graph 1.1, show that engineers have increased the power per unit displacement by gradually raising the reliable operating speed although at the expense of brake mean effective pressure. Nevertheless, since the power saws were introduced, the method of cutting has not changed.

Although the introduction of the power saw to woodlands operation in Eastern Canada dates from 1929 when a 4 hp German machine weighing 80 lbs was tested, it was not until the one-man saw became commercially available that the use of the power saw became general. For example, Brown [1.37] reports that in the pulpwood limits of Eastern Canada during the 1950-51 season less than 4% of the wood was cut
Graph 1.1 Power Saw Performance Trends

- **SPEED (RPM)**
  - 3000
  - 5000
  - 7000
  - 9000

- **BMEP (PSI)**
  - 20
  - 30
  - 40
  - 50

- **DRY WEIGHT (LBS)**
  - 0
  - 60
  - 120

YEAR
- 1935
- 1945
- 1955
- 1965
- 1975

- **TWO MAN**
- **ONE MAN**
- **LIGHTWEIGHTS**
with power saws. This increased to 55% during the 1954-55 season and to 97% during the 1957-58 season. Over the same eight year period the production went up from 1.67 to 2.44 cords per man per day, largely due to the extended use of the power saw.

Now because the totality of the wood cutting operation is being considered, a new revolution is sweeping the pulpwood lots of Eastern Canada. Instead of felling pulpwood trees and then bucking them into 8 foot lengths with a portable power saw, the operators are turning to a semi-mechanized or totally mechanized harvesting system. In the semi-mechanized system the trees are felled and topped with power saws; further processing is done by machines. In the totally mechanized system shears mounted on tractors or special harvesting machines delimb, cut, and buck the trees. The speed at which the totally mechanized system is expected to be introduced in the limited pulpwood areas in Eastern Canada, is shown by the following statistics [1.38]: whereas only 8% of the pulpwood produced in 1966 was harvested by the totally mechanized system, by 1970 this proportion is forecast to be 20% and by 1975 it is to be 75%.

As an example of the fully mechanized cutting system, the tree harvester of Figure 1.5 walks up to a tree, wraps a deliming head around the trunk and sends the deliming head flying up the tree riding on a telescoping mast. As the head moves up it removes all branches and shears the top
The operator places open bottom shear on the tree at ground level. He then encircles the tree with the delimbing arms. As the tree decreases in diameter, cutting edges follow. When the top diameter measures about three inches, operator tops the tree by actuating the hydraulically operated topping shear. After trunk is sheared at the bottom, the tree rests on the shears cupped by the curved shoe that holds it in position while the operator swings it to bunching position. The operator tilts the mast forward, opens the delimer grapple jaws and the tree drops to the ground.

Figure 1.5  Typical tree harvesting machine [1.39]
off at any desired point. The head then returns part way and holds the trunk while some larger shears snip the tree off at ground level. The head and bottom shears deposit the tree on the ground in a pre-established pattern and the cycle is complete. Although this machine is capable of 80-100 cords per day, in practice it has averaged 22 cords per man-day. This production compares with 2.6 cords per man-day for a good power saw timberjack and 1.67 cords per man-day for a good axeman [1.37, 1.40, 1.41]. These production figures could suggest the value of the "total" approach to the problem of harvesting wood. Even though the technology of shear cutting is not new, the harvesting systems were not developed until all harvesting requirements were considered as one concept. One wonders what the outcome would have been had the "total" approach been applied before the power saw was introduced to the professional logger. Would the power saw be now relegated to the casual user?

The value of applying the "total" approach to wood harvesting is evident from the brief history just given. How the formulation of a design envelope of a power saw engine was based on the "totality" of the problem is the subject of the next section.
1.2 Specifications for the Design Envelope

An accurate formulation of the design problem in the light of all the complex and conflicting requirements of real life is an important first step in creatively solving any technical problem. As long as the first stage of the analysis is incomplete, and the problems are not correctly stated, it is useless to proffer solutions. And before the problems can be correctly posed, an exact description of the expected requirements and imposed conditions must be given.

The analysis of what saw operators expect in a chain saw, what government guidelines suggest and enforce, and what factors experience has shown to be important, resulted in the following list of characteristics which were considered to be part of the design envelope:

1. speed and power,
2. environmental control,
3. allowable noise levels,
4. allowable vibration levels,
5. spark arrestor regulations,
6. safety precautions,
7. importance of cost, appearance, reliability, weight, ease of starting, ease of handling, ease of maintenance, low fuel and oil consumption, and long operating life.

Wallace [1.11] suggests that the designer should always consider the worst set of circumstances which can occur
and design accordingly because "sooner or later any machinery may be loaded to the limit of its capabilities". [1.11, p. 23]. Even though the worst set of circumstances should be considered, a good formulation of the design envelope will insure against overdesigning.

Before undertaking a new design project it is of course necessary to know if a market exists for the proposed item. If the item is to compete with existing units, it must possess some unique characteristic highly desired by prospective users. Not only must this characteristic be suitable for the environment in which it will operate, but it must also be economically and aesthetically attractive. In addition to possessing a unique characteristic, the new item must compare favourably with features and characteristics of existing units. If the market is very competitive, it may be necessary to design a unit for a specific application before it becomes saleable.

To achieve a low-cost product, the design should always be related to mass-production techniques. A simple, mass-produced part will cost less to make than a manually machined part. If it can be made and assembled on an automatic assembly line, a new saw can have a bright future.

Because how and where the product is used determines many of the requirements, a decision to design the saw for the casual user was made early in the design study. The machine was to be of specific interest to 4 groups of
people who require a small portable device:

1. tree pruners such as orchardists, gardeners,
2. construction workers such as carpenters, plumbers, electricians,
3. demolition crews,
4. casual operators such as campers, farmers, hunters.

By assuming that the time between cuts is independent of the size of the cut and power, and by using average values of cost, wages, specific cutting rates, and ratios of cutting time-to-idle time, it was possible to set up an equation linking cost to the size of cut and power supplied. More specifically, the following assumptions were used:

1. a capital cost of sixty dollars per horsepower (hp),* a working load of "N" days per year, and a depreciation in "W" working days with 6% annual interest,

   \[ \frac{(60)(hp)}{W} + \frac{(0.06)(60)(hp)}{N} \]

2. an operating cost equal to 1.3 times the annual depreciation cost \[\frac{(1.3)(60)(hp)}{W}\]

3. an average specific cutting rate (SCR) of 3 in²/sec-hp \[1.43\].

*Based on the list price for Canadian 270 with 30 inch bar and chain, April 1, 1964.
4. a 1:3 ratio of cutting time-to-idle time for a 5 hp machine in 14 in diameter timber [1.44], [cutting time = \( \frac{hp}{SCR} \) A, idle time = \( \frac{3(hp)A}{SCR} = 40 \)],

5. an operator's wage of thirty dollars a day.

Adding the depreciation, interest, operating expenses and wages, the formula for the daily cost is:

\[
\text{COST} = 30. + \frac{3.6 \text{hp}}{N} + \frac{138 \text{hp}}{W} \quad [\$/\text{day}]
\]

Taking the cycle time to be equal to the sum of cutting time based on the assumed specific cutting rate and idle time which is assumed to be constant, the production rate equation is:

\[
\text{RATE} = \frac{(\text{Area/cycle})}{(\text{time/cycle})} = \frac{A}{3 \text{ hp} + 40} \quad [\text{in}^2/\text{sec}]
\]

where A is the area in sq in per cut.

The cost of generating a surface is obtained by dividing the cost by production rate:

\[
\text{PRODUCTION COST} = \frac{0.0417}{N} + \frac{5 \text{ hp}}{NA} + \frac{41.7}{A} + \frac{.347}{\text{hp}} + \frac{1.92 \text{ hp}}{WA} + \frac{1.6}{W} \quad [\frac{\$}{1000 \text{ in}^2}]
\]
The equation for optimum power based on minimum cost is obtained by differentiating the above equation and setting the result equal to zero:

\[
(hp)_{\text{optimum}} = \frac{1}{\sqrt{\frac{14.4}{NA} + \frac{552}{WA}}}
\]

For trees and round wood the equation becomes

\[
(hp)_{\text{optimum}} = \frac{\text{diameter}}{\sqrt{\frac{18.3}{N} + \frac{702}{W}}}
\]

This equation is plotted on Graph 1.2.

If the pruner cuts branches that average 3 in diameter, or if the carpenter cuts boards that average 1 x 8 in, or if the farmer averages 7 in\(^2\) between cuts, then a 1 hp saw will be the least expensive size. As well as costing more to purchase and operate, a larger machine is less flexible to handle and harder to control.

Below 1 hp a chain saw may become more of a toy than a tool, especially since 50% of the brake horsepower never reaches the cutting point but is absorbed by the sprocket, chain and guide-bar [1.43]. The need for ensuring that transmission losses on a small saw are kept to a minimum and that the maximum amount of engine power is available for actually cutting the wood, is evident from these considerations.
The engine speed should be as low as convenient and the torque as high as practical. In addition a curve of the power as a function of speed should be flat. The flat power curve ensures that the saw cuts at a constant rate over a wide speed range. This specification is especially significant for casual users who lack the experience required to keep the saw running at a constant speed even if they knew what the optimum speed was.

Usually casual users do not cut wood in extremes of weather. Except in specific areas and isolated cases the
saw will not be used if the temperature drops below 0° F or goes above 100° F. Likewise its use in snow, or heavy rain will be quite limited.

Optimum power is not the only performance specification required; the chain speed or the shaft torque must also be specified. The values specified will depend on the kinematic ratios of the chain and sprocket, the construction of the chain and of the teeth, and the physical properties of the wood.

To approach the optimum combination of factors, the author set up a series of tests on typical chain saws. The data for each test included the time required by each cut, the area of the cut, the type of work, the engine speed, and the specification of the chain, bar and sprocket. Power was obtained from engine performance curves drawn from dynamometer data taken before and after the test. The apparatus is shown on Figure 1.6 and the data is in Appendix I.

The following variables were investigated:

1. types of wood - maple, hemlock, and cedar,
2. chain pitches - .375, .404, and .500 in,
3. sprocket sizes - 7, 8, and 9 tooth
4. joints - .025, .030, and .040 in,
5. bar lengths - 15 in and 24 in,
6. types of saw - direct drive and gear drive,
7. speed range - 4500 - 7000 rpm.
The results, drawn on Graphs 1.3 to 1.9, led to the following conclusions:

1. The minimum specific energy required to cut hemlock was 1,600 in-lb/in² and to cut maple it was 3,700 in-lb/in² (Graph 1.3). The minimum energy occurred at the lowest speed tested and coincided with the speed for most rapid cutting (Graph 1.5).

2. When cutting hemlock, the specific energy was not affected very much by the sprocket size but when cutting maple, the specific energy was lower for the smaller sprocket (Graph 1.3).
3. Because it precedes the cutter tooth into the kerf, the depth gauge tooth determines the depth of cut or "bite" (Figure 1.3). The distance the depth gauge teeth are below the cutter teeth is known as joint. The tests showed that the minimum joint produced the most efficient cuts (Graph 1.4).

4. The shorter bar cuts faster than the longer one (Graph 1.6).

5. The cutting rate for cedar was only slightly less than that for hemlock but about double that for maple (Graph 1.7).

6. For the gear drive saw, the cutting rate depended on the number of teeth passing a given point (Graph 1.8).

7. Although the gear drive with a 24 in bar cut slightly faster than the direct drive with the same length bar, it cut 15% slower than the direct drive with a 15 in bar (Graph 1.9).

Vibration levels must also be specified to ensure that the operator will not experience damaging vibration levels when handling the saw. Prolonged exposure to excessive vibration produces a vascular disturbance of the hands known as Raynaud's phenomenon, air-hammer disease, or "white fingers". During a vascular disturbance blood
Graph 1.3 Importance of sprocket size

Graph 1.4 Importance of joint size

Graph 1.5 Cutting rates

Graph 1.6 Importance of bar length
Graph 1.7 Importance of wood species

Graph 1.8 Specific cutting rate

Graph 1.9 Specific energy as a function of chain speed
circulation decreases significantly to the extent that the fingers turn white, become stiff, ache, lose their feeling, and become useless [1.45]. This condition is only temporary, although once they are damaged the hands will never again be normal and the spasms may be triggered by conditions other than vibration, such as exposure to cold.

During some cool and damp days on the coastal areas of British Columbia, 25%* of the loggers in a particular location may experience this phenomenon. Due to the temporary nature of the condition a day or two away from the power saw, or a change in weather conditions will be sufficient to allow the fingers to return to normal.

In Australia, a medical study by Grounds [1.46] showed that in a group of 22 timber fellers using chain saws, Raynaud's phenomenon occurred in 20 of them. In all but one of these 20 cases, the first trouble occurred from 1 to 6 years after they started using chain saws. Once afflicted, these men typically experienced intermittent painful pallor of the fingers, while driving their cars, while working or hunting in cold weather, or when watching football.

This 91% incidence of the disease after an average exposure of only 3 1/2 years is indicative of the severity of the vibration problem. Not only are the medical authorities aware of this danger, but loggers themselves

*Unofficial statistic from the Workman's Compensation Board Office, Vancouver, British Columbia.
are beginning to realize the extent of the disabling and permanent damage that can occur from handling the chain saws, especially the powerful lightweight ones. In May 1968, a group of loggers in Washington State went on strike to protest the use of saws they felt were vibrating excessively. In general there is a relationship between the nature of vibration and the incidence of disease:

The evidence in general suggests that if any tool produces high amplitudes at the lower frequencies of, for example, 40 to 125 cycles per second (2400 to 7500 rpm), it is likely to produce Raynaud's phenomenon, particularly if it is in continual use . . . the salient factor is not so much the tool, as the heaviness of the work expressed in terms of speed of working exertion, and grip required. [1.46, p. 272]

Ground states that:

the damage done to the vasculature of the hands bears some relation to the energy absorbed. The energy E of a wave motion of frequency F and amplitude A is given by the equation $E = KA^2F^2$ where K is a constant. On this assumption, the noxious effects of a vibrating tool will increase greatly with increasing frequency, until a limit is reached beyond which the skin will fail to conduct the vibration, and if the handles are appropriately insulated this limit will be lower. [1.46, p. 272]

The engine vibrations are the result of periodic forces acting on the mass of the machine. The present trend to lightweight saws has decreased the vibrating mass without decreasing the periodic force to the same extent. Consequently the expectation is that the amplitude of vibration should increase. But because the newer saws are designed
to operate at higher speeds, the amplitude may not have increased while the energy imparted to the machine probably has. This energy ultimately determines the amount of damage done to the vasculature of the hands.

Thompson [1.44] at McCulloch Corporation also investigated the effects of chain saw vibration on operators' health. He concluded:

1. There is danger of physical injury to the hands and arms from large amplitude saw vibrations at frequencies above 10,000 rpm.
2. High speed vibration can be tolerated from a health standpoint if the amplitude is low enough.
3. Disagreeable vibrations and injurious vibrations are not necessarily the same . . .

The reason saws have not caused more injury appears to be because (1) the relatively low speed of most saws and (2) the fact that the operator is continually changing his grip, altering his body position and operating the saw intermittently. [1.44, p. 8]

He suggested that for continuous operation the velocity should be below .75 ips; at 10,000 cpm the permissible amplitude would be .001 in. He compared this to the Russian permissible limit which is .0032 in at 5,500 cpm with a mandatory 5-10 min rest period after 45-50 min of continuous work. This limit, Thompson's limit and the damage limit for continuous operation is drawn on Graph 1.10.

The necessity for rest periods is imperative for the lightweight saws as they are run for a greater
portion of a working day. Then also, they are frequently used at high speeds; for example during limbing, the operator may open the throttle wide and use the machine

Graph 1.10 Vibration damage levels as a function of frequency as an axe. Not only do the saws run faster but they also require less time for a cut. This means that the operator makes more cuts per day and more cuts require more control and pressure on the handles.
No study of engine vibration would be complete without at least a cursory analysis of vibration causes. In a power saw most of the vibrations are caused by engine unbalance, piston clearances, and rough cutting chains. Manufacturing and design tolerances determine piston clearances and chain operation, but the engine unbalance is an inherent characteristic of the single cylinder two-stroke engine.

The reciprocating motion of the piston sets up an unbalance along the piston centerline. The unbalance in this direction can be largely eliminated by adding a counterweight on the crankshaft. But this counterweight sets up an unbalance in a plane perpendicular to the original piston direction. Thus the addition of counterweights decreases the amplitude of vibration on one plane but increases it in another.

A series of tests were performed by the author to determine the effect of a change in the counterweight size on vibration levels. In one test, 4 similar saws with vertical cylinders were used; they were identical except for the counterbalancing mass which varied from 197 grams to 248 grams. The 197 gram mass balanced 53% of the reciprocating unbalance and the 249 gram mass balanced 76% of the unbalance. In another test 6 saws were compared. For both tests root-mean-square vibration level readings were taken with the saw cutting and with it unloaded either with or without a bar and chain. From the data obtained (shown in
Appendix II), Graphs 1.10 - 1.16 were drawn and the following conclusions were made:

1. The vibration amplitude varies from .013 to .030 in and is in the area where damage to the hands will occur if saws are used continuously without the protection afforded by gloves (Graph 1.10).

2. The effect of larger counterweights is to lower perpendicular vibrations in the rear handle, and to raise both horizontal and vertical vibrations in the front handle (Graph 1.11).

3. The cutting action contributes to the vibration amplitudes in both planes but especially to the horizontal component (Graph 1.12).

4. The guide bar lowers the horizontal component of cylinder vibration but does not affect the vertical component, probably because by moving the center of gravity forward, the additional mass increases the radius from the center of rotation to the center of the accelerometer, negating the effect of the increased mass (Graph 1.13). The change in the vibration is more pronounced than indicated because the vibrations caused by the cutting action were not accounted for.

5. Whereas in the cylinder the vertical vibration is independent of initial piston unbalance, in the
handle the perpendicular vibration is dependent on the amount of piston unbalance. If the piston is lightly overbalanced so that the unbalance is mainly horizontal, the addition of the bar reduces the perpendicular amplitude, and if the piston is balanced normally so that the vertical and horizontal imbalance are nearly equal, the bar does not change the amplitude (Graph 1.14).

6. Vibration levels reach a minimum value around 6,000 rpm and are very high during engine idle, possibly due to erratic combustion (Graph 1.15).

7. Contrary to expectations, the vibration insulation rubber on the handles actually increased the vibration amplitudes (Graph 1.16).

It is obvious that the optimum counterweight will be a compromise between keeping either the horizontal or the vertical vibrations low. The foregoing results suggest, that the largest unbalance should be in a plane with the highest inertia or in a plane where the vibration is least disturbing. In a chain saw application the highest inertia is usually in the direction of the bar and the least disturbing vibrations are perpendicular to the arm.

No chain saw exhibits only translational vibrations; all exhibit a rocking motion as well. Indeed when the unbalanced force is located at some distance from the
Graph 1.11  Vibration amplitude while cutting versus counterweight size

Graph 1.12  Vibration amplitude while cutting and running unloaded
Graph 1.13  Effect of adding bar on the amplitude with several counterweights
Graph 1.14  Effect of the bar on vibration amplitude

Graph 1.15  Effect of speed on vibration amplitude
center of inertia, a large rocking couple may be set up. This fact can be used to advantage when the unbalanced force is large in only one plane. In such a situation there exists a point which undergoes very little translational motion.

Graph 1.16 Effect of rubber insulation of vibration amplitude

Finding the optimum handle position certainly begins with an analysis of the mode of vibrations and weight distribution. It is then necessary to check where vibrations can be reduced or, if it is advantageous, to have only a vertical or a horizontal unbalance and then locate one handle at a vibration mode and the other handle at a low amplitude location but parallel to the largest component at that location. The weight must be distributed to result in the overall minimum vibration levels. No factor accentuates vibrations as much as poor weight and load distribution.
and no system absorbs vibration as much as special shock absorbing gloves, usually made of foam and leather. Also, fleece gloves keep the hand warm and dry and allow for crimping in the palm of the hand where the vibration insulating material is most effectively used.

Instead of lining a glove the insulating material could be wrapped around the handle. Because high pressure is exerted on the handle, this method is not always as effective as it could be.

A third method of reducing vibration transmission is to attach the handle not solidly to the engine frame but flexibly to some insulating material. The flexible handle supposedly reduces the vibration levels transmitted to the hand. Observations during a series of vibration level tests verified this expectation; when comparing the levels of two similar saws, the vibration in the machine with the flexible handle was lower than or equal to the vibration in the machine with the solid handle.*

Exposure to high level industrial noise "limits speech communication, changes attitudes and behaviour, and impairs hearing."**

If the operator is exposed to high level noise for a considerable length of time not only will the noise contribute to fatigue but it may permanently damage the ear.

*Author's unofficial observations during vibration level tests at Pe El, Washington, June 22, 1967.
**From a talk by B.P. Carton presented to Truck Loggers' Convention, Vancouver, January 20, 1967.
What we are concerned with here is the impairment of the workman's ability to hear and understand normal speech. The degree of impairment depends on many factors, including the type of noise, the intensity of the noise, and the duration of exposure.

The sound level is generally expressed as the logarithmic ratio of the energy in the measured pressure wave to the energy in a standard pressure wave, usually the threshold of hearing at 1000 cps. Thus 0 dB represents the threshold of hearing, 60 dB represents conversational speech, and 140 dB represents the threshold of pain.

To reduce the possibility of hearing impairment from long-term exposure to high level noise, the Workman's Compensation Board of British Columbia adopted Accident Prevention Regulation 12.28 in 1966. This regulation states in part that

where the noise levels exceed the Board criteria, and the circumstances are such that a hazard to hearing exists, the noise shall be reduced to acceptable levels by engineering means. Where reduction by engineering means is not practical, adequate ear protection shall be provided and shall be worn. [1.47, p. 4]

The broad-band noise level criteria for hearing conservation adopted by the Workman's Compensation Board, the damage risk criteria suggested by Baranek [1.48] and the noise level readings at the operator's ear of typical saws are shown in Graph 1.17. The data for the noise level readings shown in
in Appendix III.

The noise intensity decreases with distance away from the saw. For example, where the operator is exposed to an intensity of 105 dBA, an observer at 23 ft is exposed to 82 dBA and at 50 ft he is exposed to 75 dBA [1.49].

Graph 1.17  Noise level readings of typical power saws

Because the extent of hearing impairment depends on the length of exposure, the noise criteria should accommodate varying amounts of exposure. The State of Washington
Graph 1.18  State of Washington standard for industrial noise

Graph 1.19  Tentative Swedish noise level limits
Occupational Health Standards criteria does this by specifying the noise level in terms of allowable exposure in hours per week without ear protection, Graph 1.18. But because the length of exposure does not vary a great deal from operator to operator, the Swedish criteria for chain saws specifies the noise levels in terms of the amount of protection given to the ear, Graph 1.19.

A properly designed muffler of either the reactive or dissipative type reduces the exhaust noise significantly. The dissipative muffler contains flow resistive material such as fibreglass or asbestos to absorb the energy released when the exhaust ports open. But when exhaust particles clog the perforations, not only is the silencing effectiveness lost, but the resistance to exhaust flow is increased. Since the ported two-stroke engine is very sensitive to backpressure, a small increase in flow resistance produces a large power drop. This type of muffler then, is not very suitable on a power saw.

The reactive muffler is timed to pass only the frequencies below its resonant frequency. Its effectiveness in reducing noise level is limited by the volume or size possible. Effectiveness can be understood in terms of an electrical analogue (low pass filter) as shown on Figure 1.7.

The current (volume of gas exhausted) flows from the source into the capacitor (muffler volume) and is dis-
charged through the inductance coil (tailpipe). To be effective at low frequencies, the capacitor and inductance must be large. The optimum volume for a power saw is in the order of 10 times the engine displacement and the optimum length of the tailpipe is in the order of 12 ft [1.44, 1.50]. Obviously a compromise between size and effectiveness is necessary.

It is generally assumed that engine noise comes from exhaust. But exhaust noises are only part of the noise picture. McCulloch's experience [1.44] shows that without the exhaust noise the accumulated aerodynamic noise of the cooling air fan and the mechanical noise of bearings, thrust

![Figure 1.7 Schematic representation of muffler and electrical equivalent [1.50]](image)
washers, gears, vibrating panels, piston slap, chain and sprocket noises are around 100 dB at 3 ft from the saw. Better engine balance and closer tolerances can reduce some of this noise. Other reductions can be accomplished by stiffening or insulating the vibrating panels and by aerodynamically designing the fan and cowling for quietness.

The muffler on the power saw has a dual purpose. It reduces the noise to an acceptable level and prevents the escape of sparks, carbon deposits and other substances likely to cause a fire. Retention is achieved if the large sparks impinge on a solid surface and break up before they leave the muffler. The solid surface may be a wall, a baffle, or a screen. Whatever is used as the impingement surface must be positioned in such a fashion that particles cannot travel in a straight line from the port to the muffler exhaust.

The Society of Automotive Engineers, in conjunction with the Power Saw Manufacturers Association, drafted a muffler standard [1.51]. To be acceptable the muffler must have an 80% retention of particles that can be retained on an ASTM#30 (.023 in) screen and a 100% retention of particles that can be retained on an ASTM#16 (.047 in) screen. Although in most ways similar to the SAE-PSMA, the California standard [1.52] specifies 90% retention in both categories. Australia accepts a muffler if pine needles with 6% moisture content
do not ignite after being held in contact with the muffler for 30 sec during a wide open throttle, no load engine test [1.53]. In Washington State the exhaust gas must impinge on 3 surfaces before exhausting from the muffler, or the muffler must contain two staggered solid baffles, or 2 perforated baffles with holes less than .187 in diameter or 2 or more screens with holes less than or equal to .030 in diameter, [1.54]. In the state of Maine, only one perforated baffle is required but the holes must be less than .080 in diameter and spaced around the periphery, or if screens are used, 3 are required and the holes must be less than .030 in diameter [1.55]. Most standards also require a muffler that is designed and made of a material that will not allow excessive shell temperatures; the SAE-PSMA standard limits this temperature to 850° F. The arrester should operate for a minimum of 8 hours before cleaning is necessary and have a life expectancy of 50 hours.

No electric, pneumatic, or engine power tool is as dangerous as one with rapidly moving, sharply filed teeth, such as a power saw. And yet no tool has as few protective guards. Little imagination is required to visualize how much injury can occur in a mishap and little persuasion is required to accept accident statistics.

A breakdown of the accidents reported to the Quebec Pulp and Paper Safety Association is given in Figure 1.8.
### Type of Accident

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<tr>
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</thead>
<tbody>
<tr>
<td>While clearing, saw kicked back, or deflected.</td>
<td>1</td>
<td>5</td>
<td>8</td>
<td>5</td>
<td>29</td>
<td>73</td>
<td></td>
</tr>
<tr>
<td>Branching with saw, kicked back, cut through or deflected.</td>
<td>1</td>
<td>8</td>
<td>31</td>
<td>2</td>
<td>35</td>
<td>45</td>
<td></td>
</tr>
<tr>
<td>Branch hit accelerator starting saw.</td>
<td>1</td>
<td>0</td>
<td>1</td>
<td>3</td>
<td>16</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>3</strong></td>
<td><strong>13</strong></td>
<td><strong>40</strong></td>
<td><strong>11</strong></td>
<td><strong>82</strong></td>
<td><strong>123</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Percentage</strong></td>
<td><strong>1.19%</strong></td>
<td><strong>2.34%</strong></td>
<td><strong>6.92%</strong></td>
<td><strong>4.36%</strong></td>
<td><strong>14.80%</strong></td>
<td><strong>21.28%</strong></td>
<td></td>
</tr>
</tbody>
</table>

### Miscellaneous

- Saw kicked back or chain jumped due to touching branch, tree log, other obstruction or pinching. 53 131 130
- As tree fell it pushed saw chain, which hit operator. 7 41 36
- Pushing branch hit saw or operator causing fall on saw. 10 11 6
- Hit by companion's saw while helping by pushing tree. 4 10 6
- Hit by companion's saw while getting away from falling tree. 12 2 2
- **Total** | **98** | **215** | **195** |
| **Percentage** | **38.88%** | **38.80%** | **33.73%** |

### Changing Position

- Fall with chain at rest, hit saw teeth. 4 15 17
- Fall with chain in motion. 2 12 9
- Hit self with saw in motion. 5 13 8
- **Total** | **16** | **45** | **34** |
| **Percentage** | **6.34%** | **7.22%** | **5.88%** |

### Power Saw Kick-Back, Hit by Companion's Tree

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<tr>
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</thead>
<tbody>
<tr>
<td>20 years old and under</td>
<td>154</td>
<td>148x</td>
<td>148x</td>
<td>Cut</td>
<td>106</td>
<td>70</td>
<td>106</td>
</tr>
<tr>
<td>21 to 25</td>
<td>154</td>
<td>154</td>
<td>154</td>
<td>Laceration</td>
<td>26</td>
<td>10</td>
<td>100</td>
</tr>
<tr>
<td>26 to 30</td>
<td>25</td>
<td>45</td>
<td>45</td>
<td>Amputation</td>
<td>10</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>31 to 35</td>
<td>58x</td>
<td>58x</td>
<td>58x</td>
<td>Bruise</td>
<td>15</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>36 to 45</td>
<td>78x</td>
<td>78x</td>
<td>78x</td>
<td>Fracture</td>
<td>29x</td>
<td>30x</td>
<td>30x</td>
</tr>
<tr>
<td>46 to 70</td>
<td>43</td>
<td>43</td>
<td>43</td>
<td>Foreign body in eye</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Unknown</td>
<td>2</td>
<td>9</td>
<td>12</td>
<td>Strain</td>
<td>16</td>
<td>31</td>
<td>23</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>252x</td>
<td>554(5x)</td>
<td>578(5x)</td>
<td></td>
<td><strong>252x</strong></td>
<td><strong>554(5x)</strong></td>
<td><strong>578(5x)</strong></td>
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### Part of Body Injured

- Finger | 10 | 30 | 41 | | 30 | 90x | 96 |
- Hand | 11 | 41 | 43 | | 13 | 88 | 104 |
- Arm | 18 | 30 | 34 | | 68 | 98 | 136 |
- Elbow | 2 | 10 | 10 | | 78 | 191 | 137(2x) |
- Torso | 30 | 58x | 43 | | 15x | 87(5x) | 105(5x) |
- Head | 21x | 24(4x) | 34(4x) | | **252x** | **554(5x)** | **578(5x)** |
- Eye | 7 | 11 | 7 | | | |
- Multiple | 14 | 23 | 43 | | | |
- Foot | 15 | 30 | 34 | | | |
- Leg | 72 | 147 | 146x | | | |
- Knuckle | 48 | 138 | 135 | | | |
- Toe | 4 | 8 | 14 | | | |
| **Total** | **252x** | **554(5x)** | **578(5x)** |

**Note:** (x) Indicates that figures include fatal accident.

### Length of Time Employed Before Accident

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<td></td>
<td><strong>252x</strong></td>
<td><strong>554(5x)</strong></td>
<td><strong>578(5x)</strong></td>
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### Figure 1.8

Accidents reported to Quebec Pulp and Paper Association [1.37]
An analysis of these statistics indicates that the moving chain was responsible for two thirds of the injuries and a third of the fatalities. The three fatalities as well as 40% of the accidents (1957) were caused by kickback; this reaction occurs when the top of the moving chain accidentally hits a foreign object and causes the whole machine to kick back out of control. The remainder of the accidents occurred when the chain jumped out of the guidebar, or when the operator or assistant accidentally hit the chain in trying to escape from falling branches or trees or were pushed into the chain by the branch or tree, or when the operator or assistant were too careless.

It is obvious that in the hands of an inexperienced operator the chain saw has a high potential for hurting and maiming. The statistics show that 75% of the accidents occur within 4 weeks after the operator starts work with a chain saw. Though somewhat safer than the bar type chain saw, the bow type is unpopular because it is expensive and cumbersome. Also considerably safer than the chain type saw because there is practically no chance of kickback, the reciprocating blade saw is unpopular because it does not feed as easily as does the chipper tooth chain nor does it remove the sawdust as quickly. Consequently, the operator must exert considerable effort to keep the saw cutting rapidly. The effort causes fatigue which in turn makes him more prone to accidents.
It is always difficult to avoid back fatigue when forcibly controlling a vibrating machine. Fatigue can be reduced by properly balancing the saw and properly designing the handles. Maximum operator comfort and minimum back fatigue result when the grip conforms to the shape of the hand and the location lines up with the guide bar and the engine center of gravity. Good balance and good control during vertical cutting require that the front handlebar be forward of the center of gravity. In this arrangement the front hand is used as a pivot and the rear hand as a control.

The engine controls must be grouped so that unintentional finger movements are kept to a minimum, and that quick natural movements of the index finger and thumb control the throttle, chain oiler, and ignition cut-off. The controls must also automatically reduce power when the fingers are accidentally or purposely removed.

Some operators think that the higher the power, the faster the cut, the less the fatigue. But with more power for a given weight, the operator spends less time cutting and more time moving from cut to cut. Because the saw cuts faster he must control the saw more. Therefore it cannot be maintained that a high-power-to-weight ratio means less fatigue, especially if the vibration levels increase as the saw weight decreases. Other means of reducing fatigue must be found because most experts agree that fatigue is a prime
cause of accidents.

To determine the importance of a number of controllable saw characteristics, a questionnaire was distributed to a number of professional and casual users. Responses were few, but replies were received from Ontario, New Brunswick, the Queen Charlotte Islands, and from the local area. The questionnaire asked the user to indicate to what extent the stated characteristics bothered or affected him, or how important he considered them to be. He was asked to evaluate them according to four degrees: very much, quite a lot, somewhat, and not at all. By assigning a value of 3, 2, 1 and 0 to these degrees, by totalling each category and dividing by the number of responses (7 for the professional logger cutting pulpwood), an importance factor was determined. The characteristics mentioned and the results obtained are shown on Figure 1.9. The original data is shown in Appendix IV.

The loggers considered easy starting, reliability and low weight as the most important characteristics, and appearance, smell and noise as the least important. Two loggers expressed personal suggestions for improvements to existing saws. Both suggested a fuel filtering system to keep out sawdust. Other suggestions included a more rugged air filter, a better chain, tinkerproof carburetor adjustments and self-cleaning ports.

Even though users did not consider appearance to be important, if other things are equal or unknown, they are
more likely to buy an attractive, functionally arranged saw than an unattractive one. For the casual user, the purchase price is often the main consideration. In comparison with other casual saws that sell for 100 to 200 dollars, a new saw should sell below 200 dollars. If the price is higher than for the conventional saw, the casual user will have to be convinced of the saw's merits before he pays the extra price.

To summarize what the operating characteristics of a power saw should be like, one can say that the casual user
requires and wants a simple one-handed tool that can be carried into a tree, pushed into a tight corner or carried long distances. For safety and convenience, he wants it to be powerful, self-starting and fast stopping because in awkward situations such as when a tree pruner is precariously perched on a branch or a carpenter is accurately cutting the overhang of a roof, hand starting is essentially difficult and dangerous. As with all hand tools, the operator wants to experience the minimum amount of fatigue and the maximum protection to his health so that noise levels and vibration amplitudes must be low. To make it easy to handle, the machine must be light in weight and small in size. It should be inexpensive to own and operate. Its source of power must be safe, reliable and readily available.

Table 1
Specifications for a Small Power Saw

1. **power** - 1 hp,
2. **chain speed** - less than 2000 fpm if a chain is used,
3. **weight** - less than 4 lbs,
4. **cost** - less than 200 dollars,
5. **vibration amplitude** - less than .002 in at 7000 rpm,
6. **noise level** - below 110 dBA,
7. **safety** - automatic stopping if trigger is released,
8. **appearance** - aesthetically pleasing,
9. **specific fuel consumption** - less than 1.2 lb/bhp-hr,
10. **fuel tank capacity** - 7 in³ - enough for 10 min continuous full power cutting,
11. **oil** - if mixed with fuel, rates should be at least 50:1,
The specifications for the design envelope have been summarized in Table I. The formulation of the specifications constitutes only the first step in the design process. The next step is to ascertain all possible means of achieving the desired output, within the specified limitations. This step, the analysis of suitable power sources and possible cutting devices, is the subject of the following chapter.
2. SYNTHESIS

2.1 Wood Cutting Devices

Finding the best source of power for the wood cutting device required an evaluation of the many sources available. The task of eliminating the less desirable and accepting the more reasonable was expedited with the help of the following criteria: the machine and its source of power must be reliable, safe, inexpensive and portable.

Solar energy was the first source to be evaluated. Not only is this source free and universally available, but its conversion into work has recently received much study. It is possible to use solar energy in the following ways:

1. to heat boilers, air heaters or energy absorbing materials directly,
2. to grow vegetation (especially algae) and then burn it as fuel,
3. to photolyze water and then burn the hydrogen and oxygen produced, and
4. to power an energy converter and then use the electricity.

When the sun is at the zenith the solar power reaching a horizontal plate at sea level is about 1/10 hp/ft². Because conversion efficiency is only about 10%, a collector area of 100 ft² is required to
produce 1 hp. According to Thirring [2.1], during a sunny day the total daily energy reaching the earth at the 50th parallel varies from a peak of 570 gram-calories per sq cm in June to a low of 100 gram-calories per sq cm in December. The caprice cloud cover prevalent in many areas reduces the radiation and makes solar power unreliable.

The drawbacks of caprice cloud cover can be overcome when the cloudy intervals are short. Thermal energy can be stored in melted salt or, if first converted into electrical energy, in batteries. One hp-hr can be stored in 25 lbs of Na₂SO₄·10H₂O [2.1], in a 20 lb zinc-air battery [2.2] or in 70 lbs of lead-acid storage batteries. Although it would not be very portable, even without an integrated storage device, solar power is feasible for a stationary power plant, as was shown by the small piston engine built in Italy to pump irrigation water. Using a flat plate as a collector and sulphur dioxide as a working material, it converted solar energy into work at an efficiency of 10%. Because solar power is not portable, it did not meet the design requirements so no further analysis was undertaken.

Although feasible in windy areas and practical in undeveloped countries, lacking a more suitable source, wind power is not reliable and not portable [2.3]. Power from tides and waves and from hydraulic potential is also not portable. Nuclear energy is not only too expensive but also
too heavy; the performance target for the 500 watt S.N.A.P.-
10A reactor-thermoelectric unit is 200 lbs per kilowatt-hour
[2.4]. Consequently windpower, hydraulic potential, and
nuclear energy were not analyzed.

It is possible to use fuel cells and batteries to
power an electric power saw. Although it has a weight
advantage over the storage battery and an efficiency advantage
over the heat engine, the fuel cell is too expensive for the
commercial market. The room type cells such as \( \text{H}_2 - \text{O}_2 \) require
expensive electro-catalysts, and expensive fuel, weigh 100 lbs
per kilowatt-hour and achieve 50% efficiencies [2.5]. High
temperature cells such as alkali-halogen with a 5 lb per
kilowatt-hour rating show some promise for commercial develop­
ment but containment is difficult [2.6].

Storage batteries are less expensive but heavier than
fuel cells. Conventional batteries weigh 70 - 200 lbs per
kilowatt-hour, zinc-air batteries weigh 20 lbs per kilowatt-
hour [2.2], organic-electrolyte batteries may approach 5 lbs
per kilowatt-hour [2.7], and a new experimental Lithium-
Tellurium battery weighs 2 lbs per kilowatt-hour [2.8]. Besides
fuel cells and batteries, photovoltaic, thermionic and thermo-
electric converters can be used to generate electricity. Photo-
voltaic converters operating with 15% efficiencies require a
high temperature heat source to supply radiant energy. The
converters can be compact; a production type 100 watt con­
verter using a radio-isotope as the source of heat can be
housed in a 2 in diameter sphere [2.9, 2.10]. A thermionic converter transforms thermal energy into electrical energy by utilizing the thermionic emission of electrons. Efficiencies of 4-19% can be obtained depending on the materials and the temperature used [2.11]. A thermoelectric generator utilizing the thermocouple principle, converts thermal energy into electrical energy at efficiencies of 2-10% [2.12]. The magneto-hydrodynamic generator, a device using 5000°F plasma jets to conduct electricity in a magnetic fuel, and the electro-gas-dynamic generator suitable for voltages above 50,000 volts are both still in the experimental stage [2.13, 2.14].

All the above mentioned fuel cells and batteries are in themselves heavy and in order to produce work, require an electromechanical device such as a motor which further increases the weight. A typical electric reciprocating saw, the 1 hp Wellsaw 400, with an 8 in blade, weighs 8 lbs [2.15]. Because batteries and fuel cells with an electric motor would be heavier and more cumbersome than existing power saws, they do not meet the design requirements.

Energy may be stored as potential energy due to gravity or pressure, or as kinetic energy, strain energy, internal thermal energy, or chemical energy. It is possible to compare the energy storing ability on a weight basis by assuming typical sizes as shown below:
1. **Kinetic energy:**

\[
K.E. = \frac{1}{2} (R \omega)^2
\]

for \( R = \text{radius} = 3 \text{ inch} \)
\( \omega = \text{speed} = 9000 \text{ rpm} \)

\[
K.E. = 860 \text{ ft-lb/lb mass}
\]

2. **Strain energy:**

\[
S.E. = \frac{S^2}{2Ep}
\]

for steel rods, \( S = \text{stress} = 200,000 \text{ psi} \)
\( E = \text{modulus} = 10^7 \text{ psi} \)
\( \rho = \text{density} = .28 \text{ lb/in}^3 \)

\[
S.E. = 240 \text{ ft-lb/lb mass}
\]

for steel springs, \( S.E. = 100 \text{ ft-lb/lb mass} \)

for rubber, \( S.E. = 4000 \text{ ft-lb/lb mass} \)

3. **Gravity potential:**

For 100 ft above datum, \( P.E. = 100 \text{ ft-lb/lb mass} \)

4. **Compressed air potential:**

\[
Wk = \frac{Pa}{\rho a} \left( \frac{1}{\gamma - 1} \right) \left( \frac{\gamma - 1}{\gamma} \right) - 1
\]

where \( Wk = \text{work out in expanding a volume of gas from P to Pa according to law pV\gamma = \text{constant}} \)

\( Pa = \text{pressure of atmosphere} \)

\( \rho a = \text{density of atmosphere} \)
γ = ratio of specific heats = 1.4
r = compression ratio = 9
Energy based on air weight only: \( W_k = 80,000 \text{ ft-lb/lb air} \)
If stored in aluminum sphere, (stress/press = 230):
\[ W_k = 2,600 \text{ ft-lb/lb mass}. \]

5. **Thermal energy in salt:**

For \( \text{Na}_2\text{SO}_4.10\text{H}_2\text{O} \), Heat of Fusion = 104 Btu/lb

Energy = 80,000 ft-lb/lb mass

6. **Hydrocarbon fuel:**

Heat of Combustion = 18,000 Btu/lb

With 20% efficiency, \( W_k = 2,800,000 \text{ ft-lb/lb mass} \)

Of all the energy storing devices considered, the hydrocarbon fuel has the highest energy-to-weight ratio and requires only a small storage volume.

The fundamental objective of the device is to cut a piece of wood with a minimum expenditure of energy and in the shortest possible time. By applying a force large enough to exceed the rupture strength of the wood fibres a piece of wood can be broken, by applying a shear load large enough to exceed the shear stress at failure the piece can be cut, by applying an abrasive material the wood can be eroded, and by applying a hot wire or flame the piece can be burned. Wood
exhibits pronounced visco-elastic characteristics. Under suddenly applied loads there is an immediate deformation approximating the classical deformation patterns, followed by a logarithmic increase with time [2.16]. This characteristic means that the faster the load is applied or the cut is made, the higher the stress will be. Consequently the specific energy will increase as the cutting speed is increased.

Under the influence of an applied bending moment, the outer fibres in a piece of wood can be broken if the moment is large enough. For round wood the required moment varies with the diameter cubed and for Douglas fir it is given by:

\[ M = 400 d^3 \text{[in-lb]} \]

where the diameter (d) is given in inches.

The amount of work required to break a given tree increases as the lever arm between the point of application and ground increases. The relationship between work per unit area and lever arm for Douglas fir is given by:

\[ \text{Specific Work} = 0.39 \ell \text{[in-lb/in}^2]\]

and the required force is given by:

\[ \star \]

*For dense structural wood, e.g. Douglas fir, southern pine, Marks Handbook [2.16], gives the following values: S = 2000 psi, Simpact = 4000 psi, E = 1.76 \times 10^6 psi.*
\[ F = \frac{400 \, d^3}{l} \ [\text{lb}] \]

where the lever arm \((l)\) is given in inches.

When a force is applied 8 1/2 ft from the ground on a 10 in diameter Douglas fir tree the following will be required to fell the tree:

- bending moment = 400,000 in-lb,
- force = 4,000 lb,
- specific energy = 39 in-lb/in^2

Because little energy is required and the type of surface produced is unimportant, felling trees by breaking them is an attractive way to clear land. This is done mechanically by pulling two long chains attached to a very large and heavy steel ball. Two tractors, travelling some distance apart, pull the free end of the chains and topple the trees between them. The reason for having the ball diameter large is to keep the forces low (by maintaining suitable ground clearance) and the reason for having the ball heavy is to prevent the chains from riding along the tops of the trees and breaking off only the treetops. Such a device was used at the Portage Mountain Dam site where two 385 hp tractors travelling 75-80 ft apart pulled an 8 ft diameter, 9 ton steel ball and cleared 9 acres of forest per hour. Trees up to 4 ft diameter were toppled [2.17]. Because the energy applied to the tree is distributed over a large volume, it is
not surprising that many ragged splinters protrude on the parting surface and many minute cracks occur on the inside of the wood. The wood broken by this method is therefore of a low grade.

When held against a rotating disc, wood heats up, becomes brittle and disintegrates. It is by this method that Yu [2.18] cut wood during experiments at the University of British Columbia. He found that at low feed rates friction does not generate sufficient heat to raise the wood to its ignition temperature; therefore wood is removed by attrition. He calculated that at high feed rates the ignition temperature is reached, so that the wood is removed by burning as well as by attrition. At a feed rate of $4 \text{ in}^2/\text{min}$, Yu required a loading force of 5 lbs and a specific cutting energy of 100,000 in-lb/in$^2$. At 16 in$^2$/min, he required a force of 70 lbs and a specific energy of 300,000 in-lb/in$^2$. These values are for cutting across the grain of a fir board 1 in thick, with a moisture content of 72%. The surface of the wood produced by a friction disc is smooth, polished and straight. The kerf is narrow and clean. But the friction disc is practical only if ample power is available.

The kinetic energy of a water jet can cut wood if the velocity is high enough to stress the wood tissues beyond their rupture strengths. Because the water jet receives its
kinetic energy as it passes through a very small nozzle, the usable energy is in a highly confined and controlled form. The jet is a cutter that is not liable to wear, does not require daily maintenance by highly skilled personnel, and produces high quality surfaces with a negligible loss of material. But cutting with water jets has one major disadvantage. Unlike the bending method of breaking wood and like the friction disc method, the high velocity jet method causes a complete breakdown of the material being removed. Because it requires energy in proportion to the amount of breakdown, the jet-cutting method is highly inefficient. Bryan [2.19] at the University of Michigan found that the energy required by this method was at least 50 times higher than normally obtained with the conventional power saw. This puts the specific energy required at about 100,000 in-lb/in². At a feed rate of about 5 in²/min and with a pressure of 30,000 psi he was able to penetrate 1 in of maple and generate 20 in² of surface area per min. At a feed rate of about 100 in²/min, the .010 in diameter jet generated 60 in²/min but the penetration was only .3 in. Extrapolating from the results produced by the .010 jet, he predicted that a .040 jet would cut 16 in stock.

A number of other researchers have investigated cutting with water jets [2.20, 2.21]. Some of their findings are listed below:
1. The depth of penetration is inversely proportional to the feeding speed; the area of cut first increases with increased feeding speed, reaches a maximum, and then at a certain point begins to decrease.

2. When the jet passes through the same cut many times, a "critical depth" is reached at which point the kinetic energy of the jets will be completely exhausted by the forces of friction against the walls of the cut and against the so called "water cushion".

3. In blind holes at depths greater than 10 hole diameters, the water pressure reaches a constant value of about 1/10 of the supply pressure.

4. The maximum pressure on a target plate is half of the supply pressure at a distance of about 350 nozzle diameters.

5. The pressure distribution on a target plate decreases radically from point of impact until at a radius of 2.6 jet radii the pressure is zero.

6. The destructive capability of jets drops slowly with increasing distance between headpiece and sample; when sample is moved 5-25 cms, the destructive effect is reduced 15-20%.

7. The width of kerf in wood is approximately equal to the diameter of the nozzle opening.

As well as by breaking the fibres, crushing the tissues, or wearing particles away, wood can be cut by shear-
ing the fibres with sharp blades. It is by this latter method that most of the pruning in gardens and orchards and most of the felling, limbing and cutting in a mechanized harvesting system is carried out. The thickness of the blade and its cutting angle affects the cutting force and specific energy required. Johnston [2.22] found that a .250 in thick knife at a cutting angle of 45° required a force of 5060 lbs and a specific energy of 817 in-lb/in² to cut a 4.6 in spruce cant. A .750 in thick knife required a maximum force of 9470 lbs and a specific energy of 1520 in-lb/in² to cut a similar cant. The effect of the cutting angle below 45° was insignificant. Based on his experimental results, Johnston came up with the following design formula which gives the maximum force required to cut fresh spruce in the diameter range of 3-6 ins [2.23, 2.24]:

\[
\text{Force} = 3000 + 2300 t \ d - 3400 t \ [\text{lb}]
\]

where \( t \) is knife thickness (inch) and \( d \) is log diameter (inch).

A typical example of a commercial shear is the Roanoke Tree Shear [2.25]. The shearing speed of Model TF-10, weighing 3400 lbs exclusive of the crawler tractor on which it is mounted, varies from 5-10 sec per tree depending on the tree diameter. A 6 in diameter hydraulic cylinder working on a maximum operating pressure of 2100-2500 psi supplies the force to the 1 1/4 in thick blade which opens up to cut 26 in diameter trees. At 12 sec for a 25 in diameter tree, the
feed rate is 2500 in\(^2\)/min and the maximum cutting force is 68,000 lbs.

Chopping with an axe, one of the first wood cutting methods used by man, is very efficient and quite fast. An experienced lumberjack can cut a 14 in diameter white pine log in 31.3 sec [2.26]. At this rate he cuts 300 in\(^2\)/min and if his power during this time averages .4 hp [2.27], the specific energy for chopping is 530 in-lb/in\(^2\). Even though his rate is high for short periods, the axeman cannot maintain this rate; after a short time, he must take a rest. Machines on the other hand, require no rest periods.

Another method of cutting wood (for which little information was available) is by drilling closely spaced holes. Instead of the shavings being removed as with a conventional bit, they can be pressed against the perimeter of the hole by special wing-and-spur cutters to give added forward thrust and reduce the power required [2.28].

Ultrasonics could possibly be used to cut wood. The mechanical friction produced by a vibrating tool could be sufficient to stress the fibres beyond their tensile limit and thus produce a rupture. Even though ultrasonics is used to push screws into plastics [2.29], its cutting potential in wood is yet to be determined.

The method of wood cutting requiring the smallest amount of external energy is burning. Once the ignition
temperature of the wood is reached with the aid of a flame or hot band, the cutting action can be self sustaining. If rapid combustion is maintained with a length of hot wire or steel band and a controlled supply of oxygen, the cut can be completed before combustion in the vicinity of the cut becomes self-sustaining. This concept was experimentally proved feasible by cutting with an oxygen-acetylene torch.

The most popular method of cutting wood is with a saw because the specific energy required is low and the cutting rate is directly proportional to the power available.

Johnston [2.30] found that for feed rates from 600-2,300 in²/min he required a specific energy of about 1460 in-lb/in² to cross-cut white spruce lumber. The energy required to cut white pine was 18% lower, while to cut yellow birch it was 23% higher. As has been shown in Chapter 1, the energy required to cut wood with a chain saw depends on the joint of the chain, the chain speed, the rate of cutting and the species of tree. For hemlock the minimum specific energy required was 1,650 in-lb/in² at a feed rate of 700 in²/min. This specific energy is shown on Graph 2.1, with the specific energy required by the other cutting devices.

Before the choice of a wood cutting device was finalized, the existing engines were evaluated. This evaluation is the subject of the next section.
Graph 2.1 Specific energies required by various wood cutting devices

2.2 Evaluation of Existing Engines

In the evolution of the internal combustion engine many types and configurations have emerged as possible alternatives to the conventional reciprocating engine. The Wankel, gas turbine, and Stirling engines, familiar to the engine designer as concepts that have overcome some of the disadvantages of reciprocating engines were evaluated as possible small power units. Other types proposed by inventors such
as Tschudi, Mercer, Llewellyn, Dotto, Lim, James, Yoto, Gursted, Virmel, Kauretz, Unsin, Rajakaruna and Selwood were considered but not evaluated as none of them appeared promising.

The gas turbine has the advantages of a relatively constant torque at all speeds, vibrationless operation and a high power-to-weight ratio. Although in principle the weight per horsepower varies with the square root of the horse power, in practice the turbine and compressor efficiencies deteriorate rapidly below 500 hp so that the promised gain in weight per horsepower tends to be cancelled out. Disadvantages of this engine are the high fuel consumption and the slow rate of acceleration.

A number of small gas turbines have been built or are proposed. Sollar's feasibility study of small gas turbines indicated that a 20 hp gas turbine turning at 100,000 rpm will have a specific fuel consumption of 1.04 lb/shp-hr and will cost $1,400 in quantities of 2,000 units per year [2.31]. An 80 hp turbine in a tractor with a reduction gear to 2,000 rpm weighs 90 lbs [2.32]. A two-stage turbine driven by mercury vapour and driving an alternator in a hermetically sealed package 7 in diameter by 18 in long, weighs 60 lbs and produces 4 kilowatt at 3,600 rpm [2.33]. Volvo's 250 hp automotive gas turbine, including transmission, weighs 800 lbs [2.34]. Ford's 400 hp gas turbines for trucks are to cost 10-15 $/hp. [2.35]. These examples
indicate that the gas turbine is not competitive costwise with the conventional reciprocating engine.

Of the numerous rotary engines proposed during the last decade only the Wankel has emerged as a practical alternative to the piston engine. Factories in Japan and Germany are producing them on an assembly line basis, and companies in Italy, France and England are planning to do the same.

Some recent design changes found necessary to improve the performance and reliability [2.36] show up some weaknesses of the Wankel:

1. intake ports were moved to the sides so that the overlap between intake and exhaust ports was reduced, thereby improving fuel consumption,

2. the troichoidal track was sprayed with special material and seals were made of more suitable materials to reduce apex seal chattering (Nitrited cast iron, molybdenum spray and chrome-plating had previously been tried),

3. the heat flow path from the unsymmetrically heated rotor and track was changed to reduce distortion.

To evaluate the feasibility of using the Wankel engine in a power saw, the performance characteristics of an industrial Wankel engine producing 6.6 hp at 5,500 rpm
was compared with an industrial two-stroke engine producing 3 hp at 5,000 rpm and a conventional power saw producing 5.8 hp at 7,000 rpm. The information on the Wankel and industrial engines was obtained from Go-Power Corporation [2.37] and the information on the power saw, from Power Machinery Limited. The Wankel engine used in this comparison was the first production version of a single rotor air cooled Fichtl and Sachs KM37 industrial unit. The scavenging charge in this engine heats up as it passes through the rotor before entering the cylinder, to keep the temperatures down to acceptable levels [2.38]. The governor, although very necessary on the Wankel, is an unattractive feature for loggers. The two-cycle industrial unit used in the comparisons was an air cooled, diecast, reed-valve-ported McCulloch Series 49 industrial engine, a type commonly used in chain saws, scooters and outboard motors. The chain saw was a sand cast, piston-ported Canadien 275 complete with bar and chain. The characteristics of the three engines are listed in Table II. The positive features of the Wankel engine, when used in a power saw, would be:

1. lower vibration levels,
2. smoother torque,
3. lower fuel consumption,
4. lower oil consumption,
5. lower starting torque,
6. lower noise level.
Table II
Characteristics of the Conventional and Wankel Engines

<table>
<thead>
<tr>
<th></th>
<th>Industrial</th>
<th>Wankel</th>
<th>Chain Saw</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement (in³)</td>
<td>4.9</td>
<td>6.6</td>
<td>7.4</td>
</tr>
<tr>
<td>Maximum power (shp)</td>
<td>3.0 @ 5000 rpm</td>
<td>6.6 @ 5500 rpm</td>
<td>5.8 @ 7000 rpm</td>
</tr>
<tr>
<td>Weight (lb/hp)</td>
<td>4.0</td>
<td>5.2</td>
<td>3.7</td>
</tr>
<tr>
<td>Specific fuel consumption (lbs/shp-hr)</td>
<td>1.1</td>
<td>0.7</td>
<td>1.0</td>
</tr>
<tr>
<td>List price ($/hp)</td>
<td>33</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>6</td>
<td>8.5</td>
<td>5.7</td>
</tr>
<tr>
<td>Vibration amplitude at 4000 rpm [2.38]</td>
<td>.017</td>
<td>.0025</td>
<td>.014</td>
</tr>
<tr>
<td>(typical)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ratio of maximum to mean torque [2.39]</td>
<td>7.0</td>
<td>3.5</td>
<td>7.0</td>
</tr>
<tr>
<td>Number of noise pulses per cycle</td>
<td>1</td>
<td>3</td>
<td>1</td>
</tr>
</tbody>
</table>

The negative features would be:

1. higher cost,
2. more weight,
3. greater complexity,
4. reduced reliability.

The last feature can be improved by more development work on the sealing, cooling and combustion systems. Graph 2.2
shows that the power-speed characteristic of the Wankel engine is quite close to that of a chain saw engine, although the power in the Wankel drops faster as the engine slows down. As the operator controls the engine speed by the amount of load he applies to the saw, the Wankel engine, as ported, would be harder to keep at the optimum speed, especially since the governor cuts in as the engine develops maximum power. As the negative factors are important to a power saw operator, other engines were evaluated.

Although invented by Stirling in 1816, the quiet hot-air cycle engine was not a practical high speed engine until very recently. The innovations which made the old concept practical are:
1. use of hydrogen or helium as the working gas,
2. increased knowledge of compact heat exchanges,
3. invention of the rhombic drive,
4. invention of "rollsock" positive seals.

Philips [2.40] built a 440 lb hot-gas engine that developed 50 hp at 2,500 rpm. By using light-weight materials, by operating the engine at high speeds, and by designing for low weight and high power, one would expect the weight to power ratio of the hot gas engine to be competitive with existing two-stroke engines.

The inherent characteristics of the external combustion engine give the Stirling engine the following positive features:

1. smokefree combustion with a great variety of fuels,
2. exhaust temperatures only a few degrees above the incoming air temperature,
3. low specific fuel consumption (.52 lb/shp-hr),
4. low peak torque (3.5 times mean output torque resulting partly from two pulses per revolution),
5. low piston velocities (750-800 fps),
6. low noise levels (inaudible at 100-300 ft),
7. low vibration levels (rhombic drive resulting in a balanced engine).

Some of the negative characteristics that produce a heavier and more expensive engine are:
1. more complicated mechanisms such as seals, power and speed control devices, regenerator, oil pump and an auxiliary compressor for the working gas,
2. higher forces in the drive mechanism and cylinder,
3. long delay in starting cold engine (1-2 min),
4. larger cooling system (approximately double conventional engine requirements).

The hot-gas engine is based on the well known principle of compressing gas at a low temperature (Figure 2.1B) and expanding it at a high temperature (Figure 2.1D). When a high temperature is required, the gas is forced into a hot region (Figure 2.1C) and when a low temperature is required, into a cold region (Figure 2.1A). What makes the modern engine work so well is an effective regenerator situated between the hot and cold regions. This regenerator stores most of the heat rejected at the end of the expansion stroke and returns it to the gas at the end of the compression stroke.

In order to get the most work out of a cycle, the power piston stroke and the displacer piston stroke must overlap, and the cold volume must go to zero. At a given ratio between the hot and cold space temperatures and the ratio between the maximum hot and maximum cold volumes, there exist optimum values of piston phase angle and swept volume. These conditions were analyzed by Kirkley [2.41].
Creswick [2.42] analyzed the heat and mass flow in the ideal isothermal Stirling cycle and made the following observations:

1. Heat must be rejected from all surfaces during compression and added everywhere during expansion to maintain a constant temperature.

![Figure 2.1 Diagrams illustrating the hot-gas cycle](image)

2. Because more mass is in the expansion space during expansion than during compression, more heat is added than is rejected. The reverse is true for the compression space.

3. The dead spaces reject as much heat as they take in, thus they have no useful function in the ideal isothermal cycle.
4. The peak heat flux far exceeds the average rate of heat addition; therefore, heat transfer surfaces should be designed to accommodate the instantaneous heat addition rates rather than the average net rate.

5. The total amount of heat deposited or picked up in one "blow" of the regenerator is several times greater than the net amount transferred from the environment per cycle.

At least five engine arrangements are possible: opposed, Vee, oil column, double acting and rhombic drive [2.43]. In the opposed piston configuration, the high pressures acting on the pistons (in the order of 100 atmospheres) must be balanced by introducing a high average pressure in the large crankcase. In the Vee arrangement the crankcase is more compact, but the large dead space between pistons and the lack of flow symmetry reduce thermodynamic efficiencies. Driving the power pistons with a column of oil overcomes some of these deficiencies but the oil must be circulated, cooled, degassed and filtered. In the double acting engine each of four pistons performs the work of a displacer and a power piston alternately, as the gas is forced from one cylinder to another through the regenerator; in this arrangement a limit to the variation in hot and cold volumes affects thermodynamic efficiencies. In the rhombic
drive configuration, a balanced engine without a pressurized crankcase is possible because the displacer piston is easily sealed with a rolling diaphragm; this configuration is thermodynamically favourable and technically appealing because of its simplicity. All of the basic components of a hot-gas engine in a rhombic drive configuration were evaluated with a view to designing them for a power saw application. The arrangement of these components is shown in the schematic diagram Figure 2.2.

The main component is, of course, the regenerator. Thermal stress can be avoided by dividing the annular space into a number of smaller chambers not continuous with the cylinder wall. Best performance can be obtained with a matrix consisting of a randomly packed mass of copper wires approximately .001 in diameter by 2 in long, arranged between two sleeves of compressed paper and retained on the ends by a coarse wire gauze [2.44]. The air used in combustion enters the preheater on the outside of the hot air ducts and spirals into the burner. Once in the burner the air is sprayed with atomized fuel, ignited, and the hot gas leaves through the inside of the hot-air duct to be insulated by the incoming air.

Altering the mean pressure of the working gas controls the engine output without affecting the thermal efficiency. The control system consists of a method for admitting and withdrawing gas from the working cycle and
a thermostat to control the heater temperature. An auxiliary compressor forces gas into the working cycle and a low pressure receiver tank (below the lowest maximum cycle pressure) accepts excess gas. By using a low density gas such as hydrogen, flow losses are relatively small, heat transfer coefficients are high, and the response to throttle is fast.

Figure 2.2  Stirling thermal engine schematic drawing [2.40]
The rolling diaphragm has added greatly to the efficiency of the modern hot-gas engine. It maintains a positive seal between the piston and cylinder walls so that no working gas can escape from the cylinder. As the diaphragm is supported on an oil cushion, its function is confined to separating the fluid in the cushion from the working gas above the piston. Across the diaphragm there is only a pressure difference of about 5 atmospheres. The real difference in pressures (50-100 atmospheres) is carried by a second conventional seal. This seal separates the oil cushion under the diaphragm from the space containing the driving mechanism. By designing steps in the piston and in the cylinder wall, the oil cushion volume becomes independent of the stroke, and by designing suitable orifices, the pressure can be made self-regulating. The endurance of the rolling diaphragms depends directly on the pressure across the diaphragm and inversely on the thickness, temperature and piston clearance; (Polyurethane rubber diaphragms last for about a year) [2.46].

A power saw engine should be air-cooled. To achieve the high cooling rate required in a hot-gas engine, a large surface area and a large blower are required. If the temperatures are not kept low, efficiency and power is low.

Scaling laws indicate that for similar engines, weights are proportional to displacement. If a constant piston speed is maintained, then the power-to-weight ratio
goes up as the displacement goes down. Also, the ratio of the surface area-to-volume goes up so that very small cylinders will have a relatively high heat transfer (a characteristic desired in the hot gas engine.)

Based on these observations, it is expected that a small engine would have a high power-to-weight ratio (if clearances can be scaled down.) Based on the Philips 40 hp engine (scaling factor \( m = 0.36 \)), a 5 hp engine running at 4,500 rpm would weigh 20 lbs and occupy the same volume as a cylinder 5 in diameter by 12 in long.

A Stirling cycle power saw would run quietly, smoke-free, nearly vibrationless and with a cool exhaust. The machine would cost more and be more complex and heavier than a conventional two-stroke engine. The delay in starting would be inconvenient, but the delay would be very short when the saw was hot. Secondary factors such as its ability to burn a variety of fuels efficiently, to operate at a lower speed and to ignite fuel without high pressure fuel pumps or timed magnetos, would present some advantages, but factors such as special seals, complicated power and speed controls, and the requirement of a pressurized gas system would present maintenance and cost disadvantages.

A chain saw is very often abused. For this reason it should be simple, rugged and foolproof. If any of the complicated controls or seals on the hot-gas engine were to malfunction, the engine would not run and would require a
competent mechanic for repairs. Because the saw would cost at least twice as much as the conventional engine and weigh about 50% more, it is doubtful that it would be accepted in the general competitive market. Nevertheless, for limited markets, where multifuel capacity and freedom from noise and vibration is paramount, the Stirling cycle power saw might be the answer. But for the purpose of this project, the Stirling cycle engine did not meet the design requirements so that the analysis was discontinued.

2.3 Synthesis of Alternatives

As the evaluation of the gas turbine, Wankel and Stirling engines failed to indicate an engine that would be competitive with the two-stroke cycle engine presently used extensively in portable power saws, the next stage in the design program was to synthesize some new engine arrangements. The synthesis started with a consideration of the basic problem areas in the existing power saws and concluded with a feasibility check.

No part in high-speed engines is as important as the connecting rod bearings. And yet no part fails as often. To produce as much power as possible from a given weight, designers have gone to higher and higher speeds, until the fatigue life of the connecting rod bearings is often reached within the useful life of the saw. The high loadings and large number of cycles cause the bearings to fail and the engine to stop. An engine without a crank would be ideal.
The crankless rotating free piston engine concept, based on the idea that a weight at the end of a horizontal level causes an unbalanced moment, meets this requirement. If the lever were allowed to rotate and if the weight were moved up whenever the weight reaches its lowest point (i.e. when the lever is vertical), then the lever would continue to rotate. By using a piston as the weight and a cylinder as a lever, an engine is theoretically possible. An analysis of the energy present in such a system yielded the following equation for the kinetic energy:

\[ \text{K.E.} = (I_c + I_p + M_p R^2) \frac{\omega^2}{2} + \frac{M_p V_p^2}{2} \]

Where

- \( I_c \) is the inertia of the cylinder about center of rotation,
- \( I_p \) is the inertia of the piston about center of rotation,
- \( V_p \) is the velocity of the piston,
- \( M_p \) is the mass of the piston,
- \( R \) is the distance from the piston center of gravity,
- \( \omega \) is the angular velocity of cylinder.

Assuming that the piston experienced only a centrifugal force (gravitational force and gas force negligible) then the differential equation is:
This equation indicates that the cylinder would slow down as the piston moved away from the center of rotation and speed up as the piston moved towards the center. In order to get work out, a planetary gear and a one-way clutch that allowed the cylinder to accelerate only if the output arm also accelerated was postulated. On closer examination it became obvious that the system would slow down and stop unless the work put into the system through combustion could accelerate the cylinder. This realization led to an analysis of the energy involved in moving the piston against the centrifugal force. The observation was made that if the system could be arranged so that while the piston was moving away from the center of rotation the cylinder rotated more slowly than when the piston was moving toward the center of rotation, net work out could be possible. Combustion must occur when the piston was furthest from the center of rotation, so that the energy released in combustion would move the piston against the centrifugal force. But this arrangement lacked a reaction force and without one no system could work. Therefore the next step was to postulate an arrangement whereby gravity would give the required reaction force. In this simple arrangement the piston moved against a gravity force, as in Figure
2.3. As long as the piston remained on the left side of the center of rotation, the output torque was positive.

Figure 2.3 Sketches of the unbalanced lever engine

In principle the engine operated as follows: as the cylinder rotated counterclockwise on shaft "0", the piston moved toward end "A" as shown in Figure 2.3. In this position the piston weight accelerated the cylinder. At the end of the stroke the compressed mixture in end "A" ignited and the pressure accelerated towards end "B". But the cylinder was also rotating, so when the piston had passed through the center of rotation, the gravitational pull on the piston again exerted a positive torque on the shaft. If the cycle continued on end "B" as it had done on end "A", and if the earth's gravitational pull was augmented by swinging the cylinder about another center, a compact engine was theoretically possible.
A simple calculation showed that for a cylinder assembly rotating at 1,000 rpm about an axis 6 in from the center of the cylinder rotation, the centrifugal acceleration was approximately 150 g. If combustion raised a 1 lb piston a distance of 1 in and if the cylinders rotated at 6,000 rpm, a 4 cylinder engine would produce about 10 hp.

An analysis of the system produced two simultaneous equations; one described the piston acceleration and the other described the cylinder acceleration:

\[
\frac{d^2 R}{dt^2} = A(P_x - P_r) + Lw^2 \cos \theta + R(\omega + \omega_o)^2 + \mu [2 \frac{dR}{dt} (\omega + \omega_o) + L\omega_o \sin \theta] - (Lw^2 \sin \theta + \mu [Rw + Lw^2 \cos \theta]) \frac{d^2 \theta}{dt^2}
\]

\[
\frac{d^2 \theta}{dt^2} = \left[2 \frac{dR}{dt} (\omega + \omega_o) + Lw^2 \sin \theta \right] \left[ \cos \theta + \frac{R}{L} + \frac{R}{Lw} \pm \mu \sin \theta \right] + A(P_x - P_x) \sin \theta - \frac{T}{I_O W + I_q + \frac{I_q}{W} - R_r}
\]

Where:

\[R_r = [R + Rw + Lw^2 \cos \theta] [\cos \theta + \frac{R}{L} + \frac{R}{Lw} \pm \mu \sin \theta]\]

\[R = \frac{r}{s}\] is dimensionless piston position,
\[r\] is piston position (from center of rotation),
\[s\] is reference stroke
\[\omega\] is angular speed of cylinder = \(\frac{d\theta}{dt}\),
\[\theta\] is angular position of cylinder (from rotating arm),
\[ \omega_o \text{ is angular speed of rotating arm} = \frac{d\theta_o}{dt} \]

\[ L = \frac{\ell}{s} \text{ is dimensionless length of rotating arm}, \]

\[ \ell \text{ is length of rotating arm}, \]

\[ t \text{ is time}, \]

\[ W = \frac{\omega_o}{\omega} \text{ is dimensionless angular speed ratio}, \]

\[ \mu \text{ is coefficient of friction}, \]

\[ A = \frac{A'P_a}{M_p} \text{ is piston area ratio}, \]

\[ A' \text{ is piston area}, \]

\[ P_a \text{ is atmospheric pressure}, \]

\[ M_p \text{ is mass of piston}, \]

\[ P_r = \frac{P_r'}{P_a} \text{ is pressure ratio in right cylinder}, \]

\[ P_l = \frac{P_l'}{P_a} \text{ is pressure ratio in left cylinder}, \]

\[ T = \frac{T_{out}}{L M_p^2} \text{ is output torque ratio}, \]

\[ I_o = \frac{I_o'}{LM_p^2} \text{ is inertia of rotating arm assembly}, \]

\[ I_q = \frac{I_q'}{LM_p^2} \text{ is inertia of cylinder-piston assembly}. \]

Because the gas forces present in these equations are highly non-linear, the equations were solved by step-wise computer integration using the Runge-Kutta subroutine.
The number of design choices available to the engineer is reflected in the number of parameters in the equations as well as the combustion conditions. In choosing appropriate parameters, trends from previous results and qualitative predictions based on existing and expected force changes were used. But even after all parameters were varied within practical limits and a large number of combinations were investigated, no stable cycles were discovered. The computed results indicated the kinetic energy of the piston would increase until the cylinder head would be blown off, as shown by typical results drawn on Graph 2.3.

The final step before abandoning the crankless rotating engine concept was to linearize the system. This meant that the gas force was replaced by a linear spring. The resulting differential equations were solved using Duhammel's Integral to give:

\[ R = \frac{L\omega^2}{p^2 - \omega^2} (\cos \omega t_1 - \cos \omega t) \]

where \( p^2 = \frac{k}{M_p} - (\omega + \omega_o)^2 \)

\( k \) = spring rate

The following are results of the linear system:

1. If \( (\omega + \omega_o)^2 \) is greater than \( k/M_p \), the system is unstable,
Graph 2.3  Computed piston positions - unbalanced lever engine
2. If \((\omega^2 + \omega_0^2)^2 = k/m\), then \(p = 0\) and \(R = L(1 - \cos \omega t)\), the cycle is sinusoidal.

3. If \(p = \omega\) and \(k/m = (\omega^2 + \omega_0^2)^2 + \omega^2\), the cycle frequency of the piston is the same as that of the cylinder; the solution by double integration is \(R = \frac{V_i}{\omega} \sin \omega t + (R_i + \frac{1}{2}) \cos \omega t\).

4. If \(p = \omega\), the maximum piston displacement is a periodic function increasing or decreasing from cycle to cycle. It is possible to obtain a stable cycle for a particular value of \(p\) and \(\omega\) if \(p < \omega\) for part of the cycle and \(p > \omega\) for the remaining part. In an engine the value of \(p\) is increased when combustion suddenly raises the gas pressure, and is decreased when the exhaust port opens.

Since the cycle must be stable for at least a small variation in speed, conditions 2 and 3 are not suitable and condition 4 becomes the only workable solution. To achieve a stable cycle the ignition timing and pressure increase must be related to the cylinder position when the piston is at the top dead center. The optimum angular position depends on the speed and the required pressure increase depends on the work removed from the cycle. These requirements are difficult, if not impossible, to achieve in practice. The concept of a crankless, rotating, free-piston engine was therefore abandoned.
As has been shown, the pursuit of a practical rotating free-piston engine required many branches of the decision tree. Some led to dead ends but others opened up alternate possibilities. When it became obvious that stable cycles were difficult to achieve in the rotating free cylinder engine, the alternate possibility of using reciprocating motion directly was investigated. Experience had shown that a piston bouncing in a closed cylinder could be made into a workable engine if the piston and cylinder were synchronized and if automatic throttling were incorporated. No major problem with sealing and combustion was envisioned as long as conventional pistons, cylinders and rings were to be used.

A number of alternates for automatic throttling were conceived but discarded because they were too complicated or too unreliable. Connecting the throttle directly to the piston was not suitable because the difference between no load and full load strokes was small. A wedge or a rotating valve complicated the engine. A device sensitive to the scavenging chamber pressure was too unreliable. Finally a lever arrangement that was simple and sensitive to stroke was conceived. A description of this arrangement with reference to Figure 2.4 follows.

In the position drawn, the scavenging charge enters the top cylinder from the top scavenging chamber, while the intake charge enters the bottom scavenging chamber from
Figure 2.4  A sketch of the oscillating free-piston engine
the intake passage, and combustion occurs in the bottom cylinder. The gas force in the bottom cylinder pushes the cylinder assembly down and the piston assembly up. The resulting motion moves the blade against a load and uncovers the throttle port to admit a fresh charge into the intake passage. Further motion of the piston and cylinder uncovers the bottom exhaust port and then the top intake and the bottom transfer ports. Then it closes the throttle port. The motion continues until all the kinetic energy in the piston and cylinder has been absorbed by the compression of the gases. Combustion then occurs in the top cylinder and the cycle repeats.

When the angle (θ) between the connecting rod and piston rod is less than 45°, a small movement in the piston results in a corresponding but larger movement in the blade. Because it is an inherent part of the blade, the throttle could be designed to remain closed for a greater portion of the cycle when the piston movement becomes larger. A plot of the piston stroke and blade stroke as a function of the compression ratio illustrates the sensitivity of the arrangement, Graph 2.4. For example, when the load is suddenly removed, and the piston stops at a compression ratio of 50 instead of 10, the piston stroke increases 13% and the blade stroke increases 57%.

After the concept was refined, a computer program using the Runge-Kutta subroutine was set up to solve the two
Graph 2.4 Blade and piston position as a function of compression ratio

coupled equations derived from a force analysis without friction. The analysis resulted in the following equations:

\[
\frac{d^2Y}{dt^2} = (P_r - P_c) A - \frac{F}{\sqrt{\left(\frac{L}{Y}\right)^2 - 1}}
\]

\[
\frac{d^2X}{dt^2} = (P_r - P_c) A m
\]

Where \( Y \) is the dimensionless blade position (\( = \frac{Y}{s} \)),

\( X \) is the dimensionless cylinder position (\( = \frac{X}{s} \)),
L is the connecting rod length (= $\frac{L'}{s}$),
s is a reference stroke,
A is the reduced piston area (= $\frac{A_p}{p_s}$),
F is the net load on the saw (= $\frac{f}{p_s}$),
$P_r$ is the gas pressure in right cylinder (= $\frac{r'}{p_a}$),
$P_l$ is the gas pressure in the left cylinder
(= $\frac{p_l'}{p_a}$),
m is the mass ratio (= $\frac{m_p}{m_c}$),
$m_p$ is the mass of the piston,
$m_c$ is the mass of the cylinder,
$p_a$ is the atmospheric pressure.

The pressures in the cylinders were calculated with
the isentropic equation:

$$P_r = 4.74 \text{(SF)} (\text{CR})^{1.4}$$

Where \text{CR} is the compression ratio (function of x+y),
\text{SF} is the scavenging factor.

The following observations were made from the
computed results:

1. When no throttling, no bounce and no synchronization
   was used and the applied load was below the full
   load:
(a) the engine stalled when the applied load exceeded the full load,
(b) the cycles were quite stable for a very heavy cylinder, e.g. cylinder mounted to heavy base.

2. When the combustion pressure depended on the blade stroke:
   (a) the cycles stabilized for a heavy cylinder at compression ratios that depended on the load applied (for no load the left compression ratio was 55 and the right was 87),
   (b) the cycles were unstable when the cylinder and piston weights were equal,
   (c) the compression ratio fluctuated for a medium weight cylinder and a heavy load.

3. When automatic throttling and auxiliary bounce chambers were used:
   (a) the cycle was unstable for cylinder weights less than 10 times the piston weight,
   (b) the addition of the bounce chamber made the cycles more unstable.

4. When the cylinder and piston were synchronized but no throttling was used:
   (a) for a large load the cycles were stable,
   (b) for a small load the compression ratio kept increasing.
When the results of the ideal cycle are applied to a practical engine, the following two conditions are required for stable cycles:

1. the cylinder must be very heavy, fastened to a heavy frame or synchronized with the piston,
2. the intake must be throttled during part load operation.

These requirements were met by synchronizing the piston and cylinder with a rack and pinion gear arrangement. A detailed force analysis yielded the following equation for the acceleration of the piston:

\[
\frac{d^2y}{dt^2} = A(P_e - P_r) + \frac{f_2 - F}{[1+d] \sqrt{(\frac{L}{Y})^2(1+d) - 1}} - f_1 - \frac{f_3}{1+d}
\]

Where \( f_1 = \frac{f_1'}{m_s} \) is the sum of the friction force between the piston and cylinder, and between the frame and cylinder,

\( f_2 = \frac{f_2'}{m_s} \) is the friction force between the blade and frame,

\( f_3 = \frac{f_3'}{m_s} \) is the friction force between the piston and frame,

\( F = \frac{F'}{m_s} \) is the load on blade,

\( d \) is the ratio of cylinder-to-piston gear diameters \( = \frac{m_p}{m_c} \) for balanced engine,
\( m \) is the reduced mass. 
\[
\left( \frac{m_p m_c}{m_p + m_c} \right)
\]

The normal force against the bladeguide is given by:

\[
F_n = \frac{F - f_2}{\sqrt{(\frac{L}{Y})^2 - 1}}
\]

When the piston is at the end of its stroke so that \( \theta \) approaches 90°, the normal force on the guide is very high if the saw load is high. But \( \theta \) only approaches 90° when the saw is unloaded, so the normal force is not as high as might first be expected. When the force is high, the stroke is short. The blade makes two cutting strokes for every piston cycle and is connected to the piston with a connecting rod containing two bearings. It was surmised that if these two bearings could be eliminated, the arrangement would have even fewer moving parts. Consequently basic engine requirements were again considered in hopes of achieving even a simpler design.

2.4 Synthesis of the Free-Piston Power Saw (FPS)

The synthesis of the new power saw started with a simple concept and ended with a practical arrangement. The concept was based on a piston bouncing in a closed cylinder filled with gas. By adding energy to the gas, the piston bounce can be maintained and work removed. By providing
the cylinder with intake and exhaust ports and by using the exhaust pulses to scavenge the cylinder, a simple internal combustion engine is theoretically possible.

Though possible in theory and attainable in practice, the idea of tuning the intake and exhaust manifolds to scavenge the cylinder is associated with long manifolds and low air flows. Higher flows are possible if a scavenging pump is used. A pump can be incorporated into the design without increasing the number of moving parts if one end of the cylinder is designed as a scavenging chamber while the other remains as a combustion chamber.

Since it is not connected to the cylinder, the piston is free to reciprocate within the cylinder and to accelerate more rapidly when the cylinder motion is impeded by an external load. Because a reaction force is required for stable cycles, the piston could be mounted so that the mount transmits the gas force, the cutting force, the loading force, and the bending moment caused by the feeding force. With the self-feeding teeth, the bending moment is small but the reaction force caused by the gas pressures is still very high. Inversion suggests that the high reaction force is due to an unbalanced mass. This problem is overcome by allowing the cylinder mass to counter-balance the piston mass. The piston and cylinder can be synchronized with a rack and pinion, belt and pulley, or connecting rod and crankshaft arrangement.
If rigidly attached to the moving cylinder, the carburetor makes fuel metering difficult. It is possible to eliminate this problem by mounting the carburetor jet on the stationary base and then allowing the venturi to move over the jet, or by mounting the complete carburetor on the base and connecting it to the intake manifold with a flexible or telescopic tube.

If the saw blade is connected to the piston instead of to the cylinder, the weights are brought closer together, so that smaller counterweights are required. This arrangement has a potential for efficient scavenging, good carburetion, and vibrationless operation. It lacks, however, a satisfactory throttling system. When the amount of energy added during each cycle just equals the amount of work removed, the cycle is stable. When the amount added exceeds the amount removed, the piston-cylinder acceleration increases until the extremely high gas pressure causes so much gas to leak from the cylinder that the piston impacts the cylinder head. Because failure can occur within 2 or 3 cycles (1/30 sec) manual throttling is too slow and the engine must be designed to throttle itself instantaneously.

To a limited extent the arrangement is inherently self-throttling, because the length of time the ports remain open is inversely proportional to the piston cylinder velocity which increases when the energy added exceeds the work removed. As the velocity increases, the ports remain
open for shorter durations, so that less charge enters. Also more energy is dissipated through friction and lost through increased heat transfer. The end result of increased throttling, friction, and heat transfer is that there is an upper limit to the piston velocity even without external throttling. Maximum torque in conventional engines occurs at about 4500 rpm and free wheeling speed occurs at 9,500 rpm. This observation concurs with a calculated result based on the assumptions that the energy released varies inversely with mean piston speed and friction removes 10% of kinetic energy. At the upper speed limit (2.2 times higher than at maximum torque) the ideal gas equation for isentropic compression predicts that the maximum compression ratio will be 180 and that the maximum pressure will be 800 atmospheres. The upper speed limit and therefore the maximum pressures can be reduced by increasing the sensitivity of throttling to speed, the friction at higher speeds, and the heat transfer at higher speeds. Of the three means listed, only throttling limits the speed efficiently.

The stroke becomes more sensitive to the kinetic energy in the piston-cylinder assembly if the bounce chamber is replaced by a spring, since the energy absorbed by the spring is proportional to the deflection squared. For example, to increase the energy absorbed to 5 times the original value, a spring will deflect 123% more, whereas the stroke of an ideal gas will increase only 16%. The
engine becomes self-throttling if the piston skirt can cover the transfer ports when the stroke is long. This means that for long strokes the ports will be open only for a short time. For very long strokes, the piston can uncover an auxiliary bleed port to recirculate the scavenging charge into the carburetor and so reduce the scavenging charge flow and increase the fuel-air ratio. The synthesis of the bleed port created confidence in the concept because it was reasoned that even if the piston skirt failed to throttle the intake properly, the bleed port would act as a safety valve.

A force analysis of the arrangement yielded the following differential equation:

\[ \frac{d^2Y}{dt^2} = A P - \frac{k}{m} Y + r \left( \frac{dY}{dt} \right)^2 + \left( \frac{F+\mu N}{1+d} \right) \]

Where \( Y \) is the dimensionless piston position (= \( \frac{Y}{s} \)),

\( s \) is the reference stroke,

\( A \) is the piston area (= \( \frac{A P a}{ms} \)),

\( P \) is the resultant pressure ratio on piston \( \left( \frac{P L-P_{r}}{P a} \right) \),

\( m \) is the reduced mass (= \( \frac{m P m c}{m P + m c} \)),

\( m_p \) is the mass of piston assembly,

\( m_c \) is the mass of cylinder assembly,
k is the spring rate,

r' is the damping coefficient (\(= \frac{r's}{m}\)),

F is the external load on saw (\(= \frac{F'}{ms}\)),

N is the normal load on saw (\(= \frac{N}{ms}\)),

\(\mu\) is the coefficient of friction,

d is the ratio of cylinder-to-piston gear diameters (\(= \frac{d'}{D}\)).

This equation was used in a computer program to solve for the piston position as a function of time. Pressures during compression were calculated using the following isentropic equation:

\[
P = \left( CR_{r} \right)^{1.4} - \left( CR_{\ell} \right)^{1.4}
\]

Where \(CR_{r}\) is the compression ratio in the combustion chamber,

\(CR_{\ell}\) is the compression ratio in the scavenging chamber.

Combustion was assumed to occur when the compression ratio in the combustion chamber reached 7.4. After this time and until the exhaust port was uncovered, the pressures were calculated according to the isentropic equation:

\[
P = 4.74 \times (SF) \left( CR_{r} \right)^{\gamma} - \left( CR_{\ell} \right)^{1.4}
\]
Where $\gamma$ is the ratio of specific heats and $SF$ is the scavenging factor.

The scavenging factor was inserted so that the combustion could be related to the scavenging and combustion efficiencies. When the throttle was completely open and all the exhaust gases were scavenged, the scavenging factor was 1.0 and the cylinder pressure, when the exhaust port opened, was 69.6 psia. When the throttle was completely closed so that no fresh charge was admitted to the cylinder, or when combustion did not occur, the scavenging factor was 0.211. The load on the blade was either applied gradually or held constant. The friction and damping forces were added after the program was debugged.

Values of spring sizes for the program were based on the assumed output of a 0.8 in$^3$ engine (1 in bore by 1 in stroke). When the mean effective pressure was 50 psi, each cycle of the engine released 40 in-lb of energy. Assuming that the compression of the fresh charge requires one fourth of the energy released and that this energy is stored in the spring, the spring rate should be 20 ppi. When it deflects to 1.5 in, the spring will absorb 22 in-lb of energy. The free length of a typical spring (.95 in O.D., .10 in wire diameter) is 2 1/2 in. If a short overload spring (.62 in O.D., .12 in wire, free length 1.33 in, rate 400 ppi) is inserted inside the main one, all of the combustion energy released in one cycle can be absorbed during a
piston stroke of 1.5 in. But an evaluation showed that the optimum spring (minimum weight) would have no pre-compression but would be long enough to remain in contact with the piston.

The following types of throttling were investigated:
(a) directly proportional to stroke,
(b) directly proportional to stroke with random combustion pressure variations (because combustion in two-stroke engines is quite erratic as shown by the results of an earlier thesis [2.47],)
(c) proportional to area of port open with random pressure variations.

The following conclusions were drawn from the computed results:
(a) the exact spring rate is not critical,
(b) the friction forces are not critical,
(c) overloading immediately stalls the engine,
(d) the engine is self-starting even under full load,
(e) the exact throttling rate and erratic combustion are not critical,
(f) conditions are more critical if the saw cuts on the return stroke,
(g) a bounce chamber is not necessary,
(h) a synchronizing mechanism is required,
(i) stiffer springs produce higher speeds
(j) the speed is nearly independent of load (or stroke),
(k) the engine can go through a number of cycles before starting.

The results showed that a free-piston reciprocating blade saw was theoretically possible and a comparison with the commercial power blade saws available on the market (Figure 2.5) showed that the new saw could be competitive. Of the ones available, only the Wright two-stroke power saw is completely portable. The electric-powered types such as the electric jig saw, sabre saw and reciprocating blade saws require a cord and a power supply while the pneumatic or the hydraulic types require a hose and a tank.

![Typical reciprocating blade saws](image)
A large number of publications on the free piston engines are available; the National Research Council's Bibliography on Free-Piston Engines (April, 1964) lists 282 references. The standard conventional free-piston engine has a stationary cylinder and two reciprocating pistons which come together during the compression stroke and move apart during the power stroke, Figure 2.6 [2.99]. This concept has been used to drive an air compressor, a linear generator, a fluid pump, and a turbine. In another configuration more closely related to the reciprocating blade saw it has been used in a pile driver, Figure 2.6 [2.50]. The head of the pile driver is similar in many ways to the head of the portable Motorburr rockdrill, but the rockdrill piston is connected to a conventional two-stroke connecting rod and crankshaft, [2.51]. The free piston engine has been used as a refrigerant compressor, [2.52]. The cylinder assembly is pivoted so that the centerline of the engine compressor passes through the center of percussion of the suspended mass. It uses fuel injection for satisfactory starting on the first stroke and a proximity plug for positive ignition.

Although the free piston engine concept is not new, its application to a reciprocating blade saw is. Indeed many of the free-piston engine features such as vibration-free operation, instant starting and stopping, multifuel capacity, and simplicity are present in the blade saw application and make the saw very unique.
How piston and hammer operates

When the explosion takes place, the piston is driven upwards; it goes through its cycle at constant speed. The hammer is driven downwards at the same time, and strikes the drill. When the piston moves upwards, it uncovers the gas duct, and gas from the combustion chamber flows into the space below the hammer flange. The gas pressure acting here on the underside of the hammer flange, together with the recoil from the drill, throws the hammer back up to its starting position. The mean pressure in the gas chamber below the hammer flange is automatically regulated by a spring-loaded valve so that the hammer is always kept in time with the piston.

Figure 2.6 Existing free-piston configurations
3. DETAILING COMPONENTS

3.1 Dimensional Analysis

Although it contributes little to the understanding of the mechanisms taking place inside an engine, similitude facilitates the interpretation of test results and extends the range of sizes engines can be scaled by correlating performance in terms of dimensionless variables and ratios. The two-stroke engine designer uses the following ratios:

1. scaling factor (specified bore/reference bore) \( m \)
2. stroke/bore \( s/b \)
3. exhaust port area/piston area \( A_e/A_p \)
4. intake part area/piston area \( A_i/A_p \)
5. exhaust port area/transfer port area \( A_e/A_t \)
6. exhaust height/stroke \( E_{xHt}/s \)
7. transfer height/stroke \( T_{rHt}/s \)
8. intake height/stroke \( I_{nHt}/s \)

He determines what sizes to use by:

1. observing how size variation affects engine performance,
2. correlating performance in terms of dimensionless variables that can be extrapolated to new sizes,
3. deducing from the correlation new dimensions that will give rise to the required performance.
In small engine design, elaborate performance predictions are usually not warranted because by building and testing an actual engine more accurate results are obtained at less cost.

A family of similar engines has the same bore/stroke ratio, mean piston speed and geometry. The deduction that the volumetric efficiency remains the same for similar engines can be shown to be valid, provided the inlet and exhaust conditions remain the same.* Under

\[\text{Flow/cycle} = K_1 \int A_0 \sqrt{\Delta P \Delta t} = K_2 \bar{A} \bar{t} = K_3 \bar{A} \frac{s}{C_m}\]

\[= K_3 (\text{ExHt})(\text{width}) \frac{s}{C_m} = K_3 (K_4 s)(K_5 \pi b) \frac{s}{C_m}\]

\[= K_b s^2 b\]

\[\text{Vol. Eff.} = \left(\frac{\text{Flow/cycle}}{\text{Displacement}}\right) = \frac{K_b s^2 b}{\rho \frac{\pi b^2}{4} s} = K_7 \left(\frac{s}{b}\right) = \text{constant}\]

Where \(\Delta P\) is pressure across port, 
\(\rho\) is density upstream, 
\(t\) is time, 
\(C_m\) is the mean piston speed, 
\(A\) is port area 
\(K's\) are constants 
\(s\) is piston stroke 
\(b\) is cylinder bore
such conditions the ratio of the amount of gas trapped in
the cylinder to the amount entering remains constant. For
a constant pressure drop across the ports and a constant
orifice coefficient, the mass of charge flowing through a
port is proportional to the length of time the port is open.
When the mean piston velocity remains constant (valid for
similar engines), the length of time the port remains open
is proportional to the length of the stroke so that

\[
\text{volumetric efficiency} = \text{constant}
\]

The weight of gas displaced per cycle and there­
fore the weight of fresh charge supplied is proportional to
the engine displacement. At a constant mean piston velocity
the engine speed varies inversely with the stroke so that
the weight of the mixture supplied, as well as the energy
supplied per unit time (if the combustion efficiency remains
constant), varies with the bore diameter squared so that

\[
\text{Energy supplied} \propto m^2
\]

where \( m \) is the scaling factor and is defined as the ratio of
any two similar dimensions.

\[
\frac{\text{Weight}}{\text{time}} = \frac{K_6 bs^2}{1/N} = K_6 \cdot bs^2 \cdot \frac{C_m}{2s} = K_8 bs
\]

\[
\frac{\text{Energy}}{\text{cycle}} = (\text{Heating Value})(\text{Eff.})(\text{Weight/time})
= K_9'(K_8 bs) = K_9 bs
\]
If the air/fuel ratio, friction mean effective pressure and volumetric efficiency are unaltered as the engine size is changed, then the indicator diagrams will be similar and the brake mean effective pressure will be constant:

\[ \text{bmep} = \text{constant}. \]

Power will be proportional to the ratio of the diameters squared so that

\[ \text{shp} \propto m^2. \]

On Graph 3.1 the shaft horsepower of typical power saw engines is plotted as a function of piston area.**

**

\[
\text{shp} = \frac{(\text{BMEP})(\text{Vd})(N)}{33000} = k_{10} (\text{BMEP}) (\text{bs}^2) (N)
\]

\[= k_{10} (\text{BMEP}) \frac{b^2 c_m}{2} = k_{11} b^2 \]

Where \( \text{Vd} \) is engine displacement,

\( \text{BMEP} \) is brake mean effective pressure, and

\( \text{N} \) is engine speed

**

Unpublished data from Power Machinery Ltd., shown in Appendix V.
Graph 3.1  Power as a function of piston area for typical power saws

When considered on a displacement basis, the power of a geometrically similar engine is inversely proportional to the bore:

\[
\frac{\text{shp}}{\text{Vd}} \propto \frac{1}{m^*}
\]

When compared on a weight basis, a small engine performs better than a big one because the weight of an engine is proportional to 3 dimensions (volume), whereas power is proportional to only two dimensions (area). This

\[
\frac{\text{shp}}{\text{Vd}} = \frac{K_{11}b^2}{\pi b^2 s} = \frac{K_{13}}{s}
\]
means that the weight per unit horsepower for a geometrically similar engine varies directly with the bore:

\[ \frac{Wt}{shp} \propto m^* \]

Though it is possible in medium and large bore engines, the idea of reducing the weight-to-power ratio by decreasing the bore diameter cannot always be utilized for bores less than 2 in. Such small engines are inefficient for the following four reasons:

1. Very small cylinders have a high relative heat loss because the ratio of exposed surface area to volume is very high during combustion. Consequently a large percentage of heat is transferred to the cylinder head and block.

2. The inlet manifold is short. As it can mix with the air for only a short time, much of the fuel goes through the engine unevaporated and unburned.

3. The cylinder wall temperatures will be lower. This results in higher oil viscosity. The high viscosity oil and high relative tolerances contribute to an abnormally high friction mean effective pressure.

\* \[ \frac{Wt}{shp} = \frac{K_1^4 b^3}{K_{14}b^2} = K_{14}b \]
4. The effective Reynolds number corresponding to gas flows may be so low that viscous forces add an increment to forces resisting gas flow [3.1].

Because small bore engines are inefficient, the power per unit displacement of typical power saws does not depend on bore size (Graph 3.2) and the weight per horsepower actually increases as the cylinder bores are made smaller (Graph 3.3).

Stresses due to gas pressure and inertia of parts will be the same at similar piston positions provided the mean piston speeds are the same, the indicator diagrams are the same, and no serious feedback of vibratory forces occurs. Under these conditions

\[
\text{mechanical stress} = \text{constant}^*.
\]

\[
\begin{align*}
\text{Stress due to gas forces} &= \frac{\text{gas force}}{\text{area}} \\
&= \frac{p \pi b^2}{K_{15} b^2} = K_{15}
\end{align*}
\]

\[
\begin{align*}
\text{Stress due to accelerating forces} &= \frac{(\text{mass})(\text{acc})}{K_{15} b^2} \\
&= \frac{(K'_1 b^3)(\frac{c_m}{t})}{K'_{15} b^2} = K_{16}(\frac{b}{s})
\end{align*}
\]

* Stress due to gas forces = \( \frac{\text{gas force}}{\text{area}} \)

\[
= \frac{p \pi b^2}{K_{15} b^2} = K_{15}
\]

\[
\begin{align*}
\text{Stress due to accelerating forces} &= \frac{(\text{mass})(\text{acc})}{K_{15} b^2} \\
&= \frac{(K'_1 b^3)(\frac{c_m}{t})}{K'_{15} b^2} = K_{16}(\frac{b}{s})
\end{align*}
\]
Even though the applied stresses remain constant, higher stress ratings are possible for small cylinder sizes because the one piece construction and thinner sections have higher allowable working stresses.

The gas side heat transfer coefficient varies inversely with the bore when the Reynolds number remains constant [3.1].

\[ h \propto \frac{1}{m} \]  

\[ h = \frac{K}{\ell} = K' \frac{16}{\ell} \text{Re} \frac{K}{\ell} = \frac{K_{16}}{\ell} \]
(if Reynolds number increases because the piston speed increases, the Nusselt number and therefore the heat transfer coefficient will be larger.) To keep the thermal efficiency high and cooling system capacity low, it is desirable to hold this heat transfer coefficient to a minimum.

Unless design changes are incorporated to keep flow paths short, the temperature of the parts \( T_w \) exposed to the hot gases will rise as the cylinder size is increased so that

\[
T_w \propto m^* .
\]

Wall temperatures are limited by considerations of strength and durability of the material and in the case of the cylinder bore, by the necessity of maintaining a bearing surface free from excessive friction and wear. Not only does the wall temperature go up as the bore increases but the temperature difference across the wall also increases:

\[
T_w - T_c \propto m^{**} .
\]

---

* Convection to wall = \( \frac{Q}{A} = \frac{K_{16}b^2}{K_{15}b^2} = K_{17} \) also = \( h(T_g - T_w) \)

\[
= \frac{K_{16}}{\ell} (T_g - T_w) ; \quad T_w = T_g - K_{16} \ell = K_{17}b
\]

** Conduction through wall = \( \frac{Q}{A} = K_{17}' \frac{\ell}{K} (T_w - T_c) \)

\[
T_w - T_c = K_{17}' \frac{\ell}{K} = K_{18}b
\]
Given that the stresses due to thermal expansion in the solid body of a given material are proportional to the difference in temperature between two points in a body, the thermal stress will vary with the bore:

\[
\text{thermal stress } \propto m^*. 
\]

Lower wall temperatures also mean lower compression temperatures and consequently lower compression pressures as well as lower local hot spot temperatures. The lower temperature and shorter flame travel length will reduce the tendency for the mixture in the cylinder to detonate. Smaller cylinders can thus be operated at a higher compression ratio before self-ignition occurs, so that

\[
\text{compression ratio } \propto \frac{1}{m}. 
\]

Centrifugal forces in rotating parts such as the crankshaft are inversely proportional to engine size,

\[
\text{centrifugal force } \propto \frac{1}{m^**}. 
\]

Deflections of parts due to mechanical forces are proportional to stresses and part length,

\[
\text{deflection } \propto m^{***}. 
\]

---

* Thermal stress = \(K_{19}(T_w - T_c) = K'_{19}K_{18}b = K_{19}b\)

** Centrifugal force = \(mrw^2 = \frac{mv^2}{s/2} = \frac{K_{20}}{s}\)

*** Deflection = \(\ell(\text{strain}) = \frac{\ell}{E}(\text{stress}) = K_{21}\ell\)
Natural frequency of vibration is inversely proportional to the length of the part

\[ \text{frequency } \propto \frac{1}{\text{m}.} \]

Wear in engines is caused by foreign matter, corrosion, and in some cases, direct metallic contact. The depth of corrosive wear per unit time is independent of the bearing size. However, the depth of wear which can be tolerated is proportional to the size of the part in question:

\[ \text{wear damage } \propto \frac{1}{\text{m}.} \]

For a family of similar engines (constant stroke/bore ratio) the effect of the cylinder size on engine performance are summarized in Table III.

For a constant displacement engine the following list of characteristics summarizes the effect of the bore/stroke ratio on engine performance:

1. A long-stroke engine will have a higher thermal efficiency than a short-stroke engine. The amount of surface area exposed to the extremely high gas temperature at the beginning of the power stroke

\[ \text{Frequency of vibration of uniform beam} = C \sqrt{\frac{gEI}{Wl^3}} \]

\[ = K_{22} \sqrt{\frac{4}{3}} \frac{m}{m} = K_{22} \frac{m}{m} \]
Table III
Scaling Characteristics for Similar Engines

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Formula</th>
</tr>
</thead>
<tbody>
<tr>
<td>energy supplied</td>
<td>$\alpha m^2$</td>
</tr>
<tr>
<td>shp, fhp</td>
<td>$\alpha m^2$</td>
</tr>
<tr>
<td>weight per shp</td>
<td>$\alpha m$</td>
</tr>
<tr>
<td>deflection of parts</td>
<td>$\alpha m$</td>
</tr>
<tr>
<td>wall temperature</td>
<td>$\alpha m$</td>
</tr>
<tr>
<td>temperature across wall</td>
<td>$\alpha m$</td>
</tr>
<tr>
<td>thermal stress</td>
<td>$\alpha m$</td>
</tr>
<tr>
<td>fmep, bmep</td>
<td>= cst</td>
</tr>
<tr>
<td>bearing pressures</td>
<td>= cst</td>
</tr>
<tr>
<td>mechanical stresses</td>
<td>= cst</td>
</tr>
<tr>
<td>power per unit displacement</td>
<td>$\alpha 1/m$</td>
</tr>
<tr>
<td>maximum compression ratio</td>
<td>$\alpha 1/m$</td>
</tr>
<tr>
<td>wear damage</td>
<td>$\alpha 1/m$</td>
</tr>
<tr>
<td>heat transfer coefficient</td>
<td>$\alpha 1/m$</td>
</tr>
<tr>
<td>natural vibration frequency</td>
<td>$\alpha 1/m$</td>
</tr>
<tr>
<td>centrifugal force</td>
<td>$\alpha 1/m$</td>
</tr>
</tbody>
</table>

depends largely on the piston force and cylinder head areas and controls much of the heat transmission from the combustion volume. The piston area to cylinder displacement ratio varies inversely with the stroke:

$$\frac{\text{combustion surface area}}{\text{displacement}} \propto \frac{1}{s}$$

The advisability of using a low bore/stroke ratio to decrease surface area is obvious, but what is not so obvious is the advantage gained through a shorter
flame travel distance. Because combustion is completed sooner and therefore closer to the optimum crank angle, combustion pressures are higher. Higher pressures and lower heat transmission result in higher thermal efficiencies. (The conventional techniques of using a domed or conical combustion chamber to reduce the surface-to-volume ratio and to shorten the flame travel distance, and using a squish area to promote turbulent combustion cannot be used in the free-piston engine.)

2. A small bore engine will operate with a cooler piston than one with a large bore. The heat flow path from the piston crown to the nearest cooled surface shortens as the bore decreases so that the temperature difference also decreases:

\[ T_h - T_c \alpha b \]

For a detonation limited spark ignition engine, the cooler piston and cylinder, and the shorter flame travel distance, allow the engine to run at a higher compression ratio. For a compression ignition engine, the cooler cylinder necessitates a higher compression ratio before ignition occurs.

3. The forces due to gas pressures will be lower for a small bore than for a large bore. Since force is proportioned to area, a smaller piston area results
in a smaller force. In terms of stroke, the relationship is

\[
\frac{\text{force}}{\text{displacement}} = \alpha \frac{1}{s}
\]

4. If the stroke is initially very short, more fuel will probably be trapped in the cylinder as the stroke is increased. Since the charge is deflected upwards into the chamber, the longer stroke produces a longer flow path so it seems probable that a long stroke will increase fuel-residue mixing and decrease short circuiting.

5. As the bore/stroke ratio decreases, higher piston speeds will be required to maintain the same output. If the displacement and brake mean effective pressure remain the same, the rpm must also remain constant. A longer stroke at a constant speed means the average piston velocity will be higher. Because it depends on velocity and not on rpm, the friction mean effective pressure and the heat generated by the ring friction will be higher. If the piston velocity is already critical, the higher velocity may cause the lubrication between the rings and cylinder to break down. But if the mean piston velocity remains constant, the power of a long stroke, constant brake mean effective pressure engine will decrease as the stroke increases.
A plot of specific power as a function of stroke for typical power saws did not verify this relationship. This is not surprising since the brake mean effective pressure and mean piston speed varies from manufacturer to manufacturer and from engine to engine. The power of typical saws as a function of the mean piston speed and the rpm is shown on Graph 3.4 and 3.5. These curves also include a small model aircraft engine (.10 in³, .14 hp) and a small utility engine (1.26 in³, .5 hp).

6. A longer stroke permits the design of larger port areas. As a long stroke engine has a high wall surface/volume ratio, the usable area for ports is also large. When the ratios of the port width/circumference and port height/stroke remain constant, the ratio of the port area/piston area is a function of the stroke/bore ratio,
Graph 3.4  Specific power as a function of piston speed

Graph 3.5  Specific power as a function of engine speed
A plot of the exhaust areas as a function of the stroke/bore ratios, verified the above supposition but a plot of the transfer port areas did not verify it, Graph 3.6.

An unsuccessful attempt was made to correlate power as a function of flow through ports by first determining the factors influencing flow. The derived relationship,

\[ \frac{V_f}{V_d} \alpha \left( \frac{A}{C} \right) \left( \frac{s}{B} \right) \left( \frac{H_t}{s} \right)^2 \]

was then set equal to a constant so that:

---

* Define:

\[
\frac{\text{port width}}{\text{width + bridge}} = \frac{w}{w+y} \equiv K_{23}
\]

\[
\frac{\text{port width + bridge}}{\text{circumference}} = \frac{w+y}{\pi b} \equiv K_{24}
\]

\[
\frac{\text{port width}}{\text{circumference}} = \frac{w}{\pi b} = K_{23}K_{24}
\]

\[
\frac{\text{port area}}{\text{piston area}} = \frac{A}{A} = \frac{w(H_t)}{\pi b^2} = 4 \frac{(H_t)}{b^2} \frac{(w)(s)}{b} = K_{25} \left( \frac{s}{b} \right)
\]

** Ratio of flow through orifice \((V_f)\) to displacement \((V_d)\):

\[
\frac{V_f}{V_d} = \frac{A_a \phi t}{A_p s} = \left( \frac{A}{A_p} \right) \left( \frac{a}{C_m} \right) \left( \frac{H_t}{s} \right) \phi = K_{26} \left( \frac{s}{B} \right) \left( \frac{H_t}{s} \right)^2 \left( \frac{a}{C_m} \right)
\]

where \( \phi = \sqrt{\frac{2}{\gamma-1}} \left( \frac{r_p^{2/\gamma} - r_p^{\gamma+1}}{r_p^{\gamma-1}} \right) \)

(continued on next page)
Graph 3.6 Port areas of typical power saws versus stroke/bore ratio

EXHAUST AREA

\[
\frac{A_e}{A_p} = 0.28 \left( \frac{s}{b} \right)
\]

TRANSFER AREA

\[
\frac{A_t}{A_p} = 0.35 \left( \frac{s}{b} \right)
\]

INTAKE AREA

\[
\frac{A_i}{A_p} = 0.28 \left( \frac{s}{b} \right)
\]

Legend:
- ■ PORTED
- ○ REED VALVE
- ▲ SPECIAL
1. when piston velocity \( (C_m) \) is constant,

\[
\frac{Ht}{s} \propto \sqrt{\frac{b}{s}}
\]

2. when rpm is constant,

\[
\frac{Ht}{s} \propto \sqrt{\frac{b}{s}}
\]

These variables were used to correlate the port heights of typical saws as shown on Graph 3.7. What this graph verifies is that the port height decreases as the bore/stroke ratio decreases.

The next step in the port design was to ascertain the shape and sizes of the intake, transfer and exhaust ports. For effective scavenging, the design objective should be to secure high values of flow coefficient and scavenging

\*(continued from last page)\n
\( a \) is velocity of sound = \( 49\sqrt{T} \)

\( r_p \) is pressure ratio across port

\( C_m \) is mean piston velocity

\*(a) when \( \frac{V_f}{V_d} = K_{27}' \) and \( \frac{a}{C_m} = K_{28}' \)

then \( \frac{Ht}{s} = K_{27}'\sqrt{\frac{b}{s}} = K_{27} \frac{A_p}{A} \)

\((b) \) when \( \frac{V_f}{V_d} = K_{27}' \) and \( N = K_{29}' \)

then \( \frac{Ht}{s} = K_{30}'\sqrt{b} \)
Graph 3.7  Typical port heights versus the square root of bore/stroke ratio
efficiency. This objective requires that the port areas be large. But for expansion stroke utilization, the design objective should be to sacrifice only a small portion of the expansion stroke to the scavenging process, and a small portion of the intake stroke to the charging process. This objective requires a low value of the exhaust port height/stroke ratio and a low value of the intake port height/stroke ratio, giving rise to small port areas. Since these objectives oppose each other to quite a degree, the two-stroke cylinder port design involves a great deal of compromise between efficient scavenging and a high power/displacement ratio. But since power was more important than efficiency, the design objective was to achieve a high power/weight ratio.

The compromise can be more easily understood and the effect of stroke/bore ratio on port areas/piston area ratio can be more easily predicted if we assume that the total width of a set of ports will be proportional to the circumference and to the fraction of this circumference devoted to porting. Based on this assumption the area ratio is proportional to the port height/bore ratio.

\[ \frac{A}{A_p} = K_5 \left( \frac{H_t}{B} \right) = K_5 \left( \frac{H_t}{S} \right) \left( \frac{S}{B} \right) \]

It is evident that for a fixed ratio of port height/stroke, the port area/piston area ratio tends to be proportional to the stroke/bore ratio as already shown on Graph 3.6.
Based on the considerations previously discussed, the bore/stroke ratio for the free-piston reciprocating blade engine was chosen as

$$\frac{b}{s} = 1.35$$

When the bore is 1.250 in the stroke should be equal to 0.93 in. For this bore/stroke ratio, Graph 3.7 suggests that the exhaust height/stroke ratio should be

$$\frac{\text{ExHt}}{s} = .30$$

Graph 3.6 suggests that the exhaust area/piston area ratio should be

$$\frac{A_e}{A_p} = .21$$

The exhaust gases flow out through the ports not only during the blowdown period but also during the scavenging period. Scavenging should start when the pressure in the cylinder has dropped to the pressure in the scavenging chamber. If the transfer ports open too soon, burnt gases will rush into the chamber and prevent proper scavenging.

As most of the burnt gases will have exhausted by the time the transfer ports are uncovered, the fresh charge forces out only a fraction of the original mass of burnt gases. On the compression stroke the cylinder pressure will build up and exceed the crankcase pressure if the
transfer ports are too high; when this happens the gas will flow from the cylinder into the crankcase and reduce the scavenging efficiency. If the transfer port area is too small, the cylinder will not be completely scavenged by the time the transfer port is closed. Here again the requirements of high flow and high scavenging efficiency are in conflict and a compromise must be made.

A trend was observed when typical values of power per unit displacement and bmep were plotted as a function of the exhaust port area/transfer port area ratio as shown in Graph 3.8. The graph indicated that as the exhaust area increased, the bmep and horsepower per cubic inch decreased. A similar trend was observed by Taylor [3.1] when he reduced the exhaust area ratio of a loop-scavenged engine from

\[ \frac{A_e}{A_t} = 1.2 \text{ to } \frac{A_e}{A_t} = 0.6. \]

Based on these observations, a suitable area ratio was assumed to be

\[ \frac{A_e}{A_t} = .70 \]

By combining this choice with the previous exhaust area ratio, the transfer area/piston area ratio becomes

\[ \frac{A_t}{A_p} = .30 \]
and the reduced area ratio becomes

\[ \frac{A_r}{A_p} = \frac{1}{\sqrt{\frac{(A_p)^2}{A_e} + \frac{(A_p)^2}{A_t}}} = \frac{A_e}{A_p} \frac{1}{\sqrt{1 + \frac{(A_e)^2}{A_t}}} = 0.20 \]

The period of time after the exhaust ports open and before the transfer ports open is called the exhaust blowdown. A plot of the blowdown/stroke ratio versus the square root of bore indicated that the blowdown ratio is independent of the bore when the ratio is varied from 0.094

Graph 3.8  Specific power and bmep versus port area ratio
to .140, Graph 3.7. This graph suggested that:

\[
\frac{\text{TrHt}}{s} = .20
\]

Combining this choice with the ratio of the exhaust port height yields

\[
\frac{\text{Blowdown}}{s} = .10
\]

In designing for the intake port area and height, a compromise must be made between a large flow and minimum amount of blowback. A large flow is achieved by making the ports large; blowback through the carburetor is minimized by making port areas small and port heights low. If the areas are too small, excessive throttling will take place at high speeds; if the areas are too large, excessive blowback through the carburetor will occur at slow speeds. The optimum area and height will result in the maximum quantity of mixture being trapped at the operating speed.

The areas can be made large without sacrificing blowback if a one-way reed valve is used. But reed valves fail, cost money, add complexity to the engine and increase the bmep only slightly. For a small power saw the advantage of a higher torque (bmep) probably will not outweigh the disadvantages of increased engine complexity and decreased reliability.

With the decision made to use ports instead of
reed valves, there remains the question of what intake port area and height to use. As a plot of specific power versus area indicated no trends, the area chosen was the same as the 1.25 in diameter engine:

\[
\frac{A_i}{A_p} = 0.19
\]

For a bore/stroke ratio of 1.35, Graph 3.7 gives

\[
\frac{\text{InHt}}{s} = 0.28
\]

The port shape controls the rate of port opening. The rate for squared ports is higher than that for round ports, but a square hole is more difficult to machine and tends to snag the ends of the piston rings unless the rings are pinned so that the ring ends are always supported by the cylinder wall. Since round holes have the width equal to the height, the required small port height necessitates a large number of small holes which result in a large perimeter/area ratio and a high friction loss.

To reduce the perimeter/area ratio of round holes, two techniques were considered. One technique was to expose only a portion of a larger hole and the second technique was to drill the hole at an angle. The first technique is standard practice on many engines with round exhaust ports. The ports are drilled so that the top edge is at the required height; the drill size chosen is one to give the required
cross-sectional area (when the port is uncovered to the
specified height) without exceeding the allowable portion
of the cylinder circumference. The second technique, often
used for the transfer port, is to drill holes at an angle
so that elongated holes result.

As well as having a high perimeter-to-area ratio,
an elongated hole imparts a directional momentum to the
charge moving through it. This can be used to advantage in
the transfer port by directing the fresh charge to the back
of the cylinder. The transfer port is so placed that the
flows from two opposing holes meet and are deflected upward.
When the flows meet, the horizontal velocity components cancel
and the resulting pressure wave aids cylinder scavenging.
The directional momentum may also be created in a square port
by slanting the top edge of the port so that the port is
uncovered gradually from the back to the front, causing the
charge to flow toward the back of the cylinder.

Because round holes can be easily drilled in sand-
cast cylinders, they were specified for all ports. Had
squared ports been specified, the machining operation would
have involved a more complex milling procedure. Had the
cylinder been diecast instead of sandcast, the closer toler-
ances possible would have allowed any shape of port to be
 integrally cast.

Drilled holes were used in this design to produce
large perimeter/area ratio holes. Based on the specifications
determined earlier, the limitations of space around the circumference, and the requirement of efficient scavenging, the following holes were specified:

1. exhaust ports - two .406 in diameter holes exposed to a height of .28 in to give an exposed area of .19 in$^2$,

2. transfer ports - two .281 in diameter holes on each side drilled 45° to the radial line and exposed to a height of .18 in to give an exposed area of .24 in$^2$.

3. intake port - eight .172 in diameter holes, exposed to a height of .12 in to give an exposed area of .15 in$^2$. (It should be pointed out that the exposed height of the intake port is small because the relative stroke between the mount containing the holes and the cap exposing them is half of the relative stroke between the cylinder containing the transfer and exhaust ports and the piston exposing them.) Strength and mount circumference size considerations influenced the choice of the number of holes for the intake port.

3.2 Automatic Throttling

The flow of air and fuel through the engine is caused by pressure variations in the scavenging chamber under the piston (also called precompression chamber). In
moving towards the cylinder head on its compression stroke, the piston expands the scavenging chamber volume, thereby lowering the chamber pressure. The pressure continues to decrease until the cylinder cap uncovers a row of intake holes in the circumference of the stationary roller mount. These holes connect with a passageway leading to the carburetor. If the atmospheric pressure is higher than the scavenging chamber pressure, air rushes through the carburetor, mixes with the fuel, and flows into the chamber. The air continues to enter the chamber through the intake port until either the chamber pressure reaches atmospheric pressure or until the cap on the cylinder, returning on the power stroke, covers the port. Since the chamber volume decreases during the power stroke, some charge may escape through the intake port. See Figure 3.3.

On the top side of the piston the cylinder volume expands during the power stroke and the cylinder pressure drops nearly isentropically until the piston uncovers the exhaust port to start the blowdown process. The rapid drop in cylinder pressure associated with the blowdown continues until the piston uncovers the transfer port or until the pressure reaches atmospheric pressure. If not enough burnt gases have escaped through the exhaust ports by the time the transfer ports open, the pressure in the cylinder will still be greater than the pressure in the scavenging chamber and burnt gases will flow into the chamber. This action
reduces scavenging efficiency. On the other hand, if the blowdown time is too long so that the cylinder pressure reaches atmospheric pressure before the transfer ports open, part of the stroke remains unutilized and the percentage of the fresh charge escaping during the subsequent compression stroke is increased. For properly designed porting the fresh charge will flow from the scavenging chamber through the transfer ports and into the cylinder for as long as the ports are open.

While the transfer ports are opening, the throttle ports are closing. Optimum stroke is reached when the transfer port area equals the throttle port area. If the piston does not stop at the optimum stroke but continues to move on its power stroke, the throttle port will continue to close. If the power stroke is long enough and the auxiliary bleed port opens soon enough, the throttle port will be completely closed and the bleed port will be completely open. The open bleed port allows the fresh charge to escape from the scavenging chamber before the piston reaches bottom dead center. At the end of a very long stroke most of the fresh charge will have escaped, causing the compression stroke to create a vacuum in the chamber. This draws some of the charge which had already entered the cylinder, into the scavenging chamber as soon as the throttle ports are uncovered. Consequently, only a small amount of charge will remain in the cylinder, so that the
intake is effectively and automatically throttled.

Automatic throttling must accomplish two things:

1. allow the maximum quantity of fuel-air mixture to enter the cylinder when the piston stroke ends at the maximum power position,
2. throttle the mixture according to the amount of energy required.

The intake, exhaust, and transfer port dimensions, to satisfy the first requirement, were obtained by scaling existing engines as described in the previous section. The throttle port size and shape, to satisfy the second requirement, could not be obtained from existing engines, so a computer program was set up to predict the engine response as the sizes and shapes of the throttle port were changed.

In the first computer trials, a hypothetical relationship between the stroke and energy released was used. This relationship was based on the assumption that throttling was directly proportional to the piston position at the bottom dead center. No throttling occurred when the dead center coincided with the maximum power position and full throttling occurred when the dead center reached a preset position. For example, in one series of tests no throttling occurred when the dimensionless stroke (at dead center) was 1.2 and full throttling occurred when the stroke was 2.7. These two points were connected by a straight line as shown
on Graph 3.9. On this graph is plotted the port area-time curve based on rectangular ports and three bleed port size variations. The values were obtained by measuring the area under the computer-calculated position-time curve. The computer calculations were based on a hypothetical straight line relationship between the amount of throttling and the piston position. It is possible to change the shape of the area time curve not only by changing the size of bleed port.
areas as shown, but also by altering the port timing and by modifying the port shape.

After the port area-time factors were obtained graphically, the computer program was modified to integrate the area-time directly. Throttling for the program was still based on the hypothetical straight line relationship.

The intake port also contributes to throttling as is shown by the intake area-time integration. Because the piston dwells near the top for shorter periods when the high compression pressure produces high accelerations, the intake port, as reflected in the integration, is not open as long.

The spring rate affects the integrated port area. Because a stiffer spring results in a higher acceleration near bottom dead center, the integrated port area decreases as the spring rate increases. The top curve on Graph 3.10 depicts the plot of the piston stroke as a function of time for 3 values of the spring rate. The cycles become more stable as the spring rate increases. For the weak spring \((k/m = 40,000)\), the third stroke is higher than the second and the fourth is shorter than the effective stroke. This can be explained as follows: for the first stroke the amount of throttling was less than the amount of energy absorbed by the spring so that the second stroke was too long, causing too much throttling. The third stroke was too
short and consequently did not cause enough throttling. The unstable cycles continued until the engine stalled. By increasing the spring stiffness, the spring absorbed more energy and produced more uniform cycles. (The same effect could have been produced by decreasing the rate of throttling.)

The second curve depicts the piston position at the end of the power stroke when throttling depends on an integrated port area. Again combustion pressures were assumed to vary at random but less than ±10% from the values calculated. The operating speed depended on the spring stiffness; for a weak spring, the piston oscillated at about 4,250 cpm, for a medium spring, 5,150 cpm, and for a stiff spring, 6,000 cpm.

The effect of the piston weight on the piston position is illustrated by the third curve. The medium weight piston produces the most stable cycles. It should also be noted that the engine speed at full load is only 5% higher than at no load (6,000 cpm versus 5,700 cpm).

The bottom curve indicates what effect throttling and erratic combustion have on stability. Although the erratic combustion did not affect the stability significantly when the engine was throttled linearly, it did cause the engine to stall earlier when the engine was throttled in proportion to the integrated port area.
Graph 3.10  Piston strokes as a function of time for the ideal oscillating power saw
The next stage in the optimization of the throttle port size was to calculate the flow rate using an ideal gas model. By employing the techniques used by London [3.2] in his solution of a receiver blowdown problem, more exact equations were obtained for the amount of throttling the charge experiences in flowing through the ports. The temperatures in the cylinder and scavenging chamber depend on the amount and temperature of the gases entering and leaving, and on the amount of compression taking place. The rate of gas flowing through a port was calculated from the following equation suggested by Taylor [3.1]:

$$W_C = \frac{dM}{dt} = AC \rho a \sqrt{\frac{2}{\gamma-1} \left[ \frac{P_d^{\gamma}}{P_u} \frac{2}{\gamma} - \frac{P_d^{\gamma+1}}{P_u^\gamma} \right]}$$

Where

- $A$ is the port area,
- $C$ is the port coefficient,
- $a$ is the speed of sound,
- $P_d$ is the downstream pressure,
- $P_u$ is the upstream pressure,
- $\rho$ is the density,
- $\gamma$ is the ratio of specific heats.

The derived temperature equation used an ideal flow rate equation for each port and assumed that compression took place isentropically. For example, when the flow enters through the intake ports and leaves through the scavenging port and bleed port, the scavenging chamber temperature varies
\[
\frac{dT_p}{dt} = \left(1 + \frac{R}{m c_v} \right) T_a - T_p \frac{M_I}{M_p} - \frac{R}{m c_v} T_p \left(\frac{M_B}{M_p} + \frac{M_C}{M_p} - \frac{X}{L-X}\right) - \frac{Q}{C_v M_p}
\]

Where \( R \) is the gas constant,
\( m \) is the molecular weight,
\( c_v \) is the specific heat at constant volume,
\( T_a \) is the temperature of charge entering through intake,
\( T_p \) is the temperature in scavenging chamber,
\( w_I \) is the flow rate through intake port,
\( w_B \) is the flow rate through bleed port,
\( w_S \) is the flow rate through scavenging port,
\( M_p \) is the mass of gas inside scavenging volume,
\( X \) is the piston displacement,
\( \dot{X} \) is the piston velocity,
\( L \) is the maximum effective piston position,
\( Q \) is heat transferred out of cylinder.

When the flow reversed through any of the ports, a different though similar equation was used. Also equations for the flow of fuel and fresh air were later added to allow computation of air-fuel ratios throughout the engine.

When combustion occurred, the cylinder temperature was assumed to increase according to the following equation:

\[
T_c = T_g + \frac{F}{A} \left(\frac{19,800}{C_v}\right) M_r \frac{M_r}{M_c}
\]
Where $T_g$ is the temperature in cylinder prior to combustion,

$\frac{F}{A}$ is the fuel-air ratio of mixture entering,

$M_r$ is the mass of fresh mixture trapped,

$M_c$ is the mass of mixture in cylinder.

The specific volume calculations were based on the mass of charge present in the chamber and its volume. The pressure calculations were based on the specific volume, the chamber temperature, and an equation of state.

The throttle port diameter chosen for the actual engine design, based on the factors discussed and the need to ensure that the maximum amount of throttling would occur, was the same as the transfer port diameter. The holes were spaced so that, at the optimum stroke, the portion of the throttle port uncovered is the same as the portion of the transfer port uncovered. When the stroke is longer, the throttle port is partly or completely covered, whereas the transfer port is partly or completely open. Because the two ports are in series, both ports control the flow rate.

By specifying a width of $1/8$ in and a depth of $1/8$ in, the total cross-sectional area for two bleed ports would be $0.03 \text{ in}^2$ (or $1/6$ of the transfer port area). By specifying a height/stroke ratio of $1.8$, the effect of the bleed hole should be similar to that shown on Graph 3.9.
3.3 Design of Components

3.3.1 General Considerations

In the preceding step, concept feasibility and performance optimization were design considerations. In the present step strength, cost, material availability and manufacturing capability were general considerations. More specifically, the following practices were followed in designing the parts constituting the free piston reciprocating blade power saw:

1. right and left hand parts were eliminated wherever possible,
2. as few parts as possible were used,
3. size and weight of parts were kept at a minimum, except for the piston which had to have the same mass as the cylinder,
4. wherever practical, the machine shop facilities at the University of British Columbia were used,
5. ease of assembly and appearance determined the shape of the parts where strength was not important.

A 1 hp engine oscillating at 6,000 cpm with a .93 in stroke will produce about 65 in-lb of work per cycle. A 1.25 in diameter piston and cylinder (bmeq = 57 psi) exert a combined cutting force of 140 lbs on the blade. This
force must be matched by an equal force on the handle. The operator applying this force also exerts a feeding force whose magnitude depends on the shape and condition of the teeth (assumed to be 40 lbs as shown on Figure 3.1), and a twisting moment. If the force applied is 2.3 in ($X_2$) above the specified cutting force and 8 in ($X_1$) back from the feeding force, no bending moment is required. If forces are applied on the bottom handle ($X_2 = -3.5$ in and $X_1 = -4.0$ in) then the required twisting moment equals 950 in-lb). Because the twisting moment is smallest when the top handle is used to load the saw, the bottom handle will be used mainly to control the cut.

Figure 3.1  Free body diagram of the reciprocating power saw
The bending moment acting in the blade depends on the magnitude of the forces, the preset distance from the sawing force to the blade centroid ($X_3$), and the constantly changing distance from the feeding force to the frame ($X_4$). (If $X_4$ varies 1-7 in and averages 4 in, $X_3$ should be .86 in, so that the bending moment during the cutting stroke will be less than ±120 in-lb and the stress in a .12x.64 in blade will be less than 16,000 psi.) But the maximum stress occurs not as calculated above but during the return stroke. If the feeding force remains at 40 lbs and friction is 20% of the feeding force, the stress equals 29,000 in-lb during the return stroke, giving a factor of safety, with AISI 1040 steel, of 2.7. The shear stress in the lug and the bending stress at the pin cross-section are slightly lower at 28,000 and 27,000 psi respectively.

3.3.2 Bounce Spring

Once it was determined that the overall forces were reasonable, an analysis was made of the requirements of the power spring. Its purpose was to compress the charge and to provide a means of relating the amount of the throttling to the amount of work. The ideal relationship required a direct correlation between the amount of energy released in combustion (controlled by the throttle) and the amount of energy converted to work (controlled by load and stroke length). When the amount of work is reduced by removing
some of the load from the blade, the stroke length increases to absorb more energy. The increased stroke must throttle the intake more, so that the subsequent power strokes will release less energy. The correct amount of throttling will release just as much energy as is required by the work taken out so that the length of subsequent strokes will not change. If the amount of throttling is not correct, the stroke length will continue to change until the quantity of energy released and the amount of work performed is balanced.

The equation governing the length of stroke, derived from an energy balance taken over one cycle, is as follows:

\[ N'(x_i) + f(x_i) = W_k + f(x) + E_f \]

Where \( N'(x_i) \) is the energy released in combustion, 
\( f(x) \) is the energy stored in the storing device, 
\( W_k \) is the work removed, and 
\( E_f \) is the energy lost in friction.

When a mechanical spring with a linear throttle rate is used, the equation can be solved for \( x \) and expressed as:
\[ X = -F + \sqrt{F^2 + N + X_i^2} \]

Where \( X \) is the piston position at BDC \( (= \frac{x}{x_{o}}) \),
\( X'_i \) is the initial position at BDC \( (= \frac{x'}{x_{i}}) \),
\( x_{o} \) is a reference position,
\( F \) is the load on blade \( (= \frac{F'}{kx_{o}}) \),
\( k \) is the spring rate,
\( N \) is the work released during the power stroke minus energy lost in friction in dimensionless units \( (N = 2N^* = 2 \frac{(N'(x_i) - E_f)}{kx_{o}^2}) \).

This equation can be used to study the effect of throttling on the stability of the cycles. A general requirement for stability is that a suddenly applied full load or a suddenly removed full load should not stall the engine.

For example, consider a typical engine: it reciprocates with a stroke of 1 in, releases 100 in-lbs during the power stroke, performs 60 in-lbs of work and stores 40 in-lbs of energy in the spring \((k=80 \text{ ppi})\). When the load is suddenly removed the spring will deflect 1.6 in to absorb all the energy released (e.g., 100 in-lbs less 5% friction) and the throttle should allow no more charge to enter. In this example the equation for the energy released is given by:

*Assumptions: 1. amount of throttling varies linearly with the stroke and releases 60 in-lbs of energy when \( X_i=1 \) and releases no energy when the spring has absorbed 60 in-lbs of energy, 2. a linear spring is used, 3. work out varies with force and distance, 4. friction loss is proportional to maximum energy released, 5. effective stroke does not change.*
\[ N^* = 2.04 - 1.25 X_i \]

and the equation for the piston position is:

\[ X = -F + \sqrt{F^2 + X_i^2 + 4.08 - 1.25 X_i^2} \]

The equation giving the stroke length when the load is suddenly removed is obtained from the latter equation by setting the load \( F \) equal to zero:

\[ X_n = \sqrt{X_i^2 - 2.5 X_i + 4.08} \]

The equation giving the stroke length when the engine is suddenly loaded to capacity is obtained by setting \( F = .75 \) and can be represented by

\[ X_L = -.75 + \sqrt{X_i^2 - 2.5 X_i + 4.64} \]

The second equation on Graph 3.11 is a more accurate representation of throttling because it approximates the integrated port area more closely. If this equation,

\[ N = 1.94 - 1.19 X_i - \frac{.00075}{(X_i -.9)^3} \]

is substituted into the stroke equation and the load suddenly removed, the piston stroke is given by

\[ X_n = \sqrt{X_i^2 + N} \]
and when the load is suddenly applied (F=.49) the piston stroke is:

\[ X_\ell = -0.49 + \sqrt{(0.49)^2 + X_i^2 + N} \]

Note on this graph what happens to the piston when the load is suddenly removed. Assume that during the previous cycle the piston had stopped when \( X_i = 1.30 \). This means that .41 unit of energy is added to the engine when combustion occurs (throttling follows the \( N=2.04-1.25X_i \) curve, \( \text{1} \) on Graph 3.11). If the load is then removed so that no work is taken out of the engine during the next power stroke, the spring will deflect 1.56. The next combustion cycle will release .07 units of energy, and deflect the spring to 1.62. No energy will be released during subsequent cycles until friction reduces the stroke to its steady state, no-load value of 1.6.

If the load is suddenly increased from F=.31 to F=.75 the stroke will decrease to 1.0, \( \text{4} \) on the curve. The next combustion cycle will release the maximum energy of .78 units and the engine will be stabilized at \( X=1.0 \). If the throttling follows the \( N=1.94-1.19X_i-.00075/(X_i-.9)^3 \) curve and the load is suddenly increased to F=.49, the strokes will gradually decrease until steady state conditions are reached at \( X=1.11 \).
A typical coiled spring with a spring rate of 87 ppi absorbs 45 in-lb when compressed to 1.03 in, 105 in-lb when compressed to 1.56 in, and 175 in-lb when compressed to 2.00 in. The rate of throttling must therefore vary linearly from no throttling at a stroke of 1.03 in to full throttling at a stroke of 1.56 in.

High carbon spring steel wire (.156 in diameter), when made into a 1.0 in mean diameter coiled spring, will require 10 active coils to store 175 in-lb of energy during a 2 in deflection. It will compress to a solid height of 1.56 in, occupy a .80 in$^3$, and weigh .17 lbs. This spring
can store 1,000 in-lb/lb mass or 220 in-lb/cu in.*

The gas under the piston can be used as an energy absorber and piston bounce. The equation for the amount of work required to compress an ideal gas isentropically suggests that the energy absorbing ability of the gas increases rapidly with a decrease in volume. For example, when the maximum possible stroke of the engine is .750 in, a stroke of .652 in stores 45 in-lb of energy at a compression ratio of 7.6, a stroke of .734 in stores 105 in-lb at a compression ratio of 54, and a stroke of .746 in stores 175 in-lb at a compression ratio of 195. For this example the throttle must accomplish three things:

1. not restrict the flow when the stroke is .652 in,
2. stop the flow completely when the stroke is .734 in or longer,
3. vary the flow in an isentropic fashion between the two limits.

Because this is very difficult to accomplish, gas is not a good means of storing energy. Flat springs

* Reference [3.3] gives the following specifications for oil tempered wire (ASTM A229-41):

\[ S = \text{Elastic limit} \ 120,000 - 250,000 \text{ psi}, \]
\[ E = \text{Modulus of Elasticity} \ 30,000,000 \text{ psi}, \]
\[ \rho = \text{Density} \ 0.282 \text{ lb/cu in.} \]
or bands can be used as an energy absorber and bounce. They can also be used to synchronize the cylinder with the piston, thereby serving a dual purpose.

Steel bands are more efficient on a weight basis than a coil spring, and make possible a variable spring rate that corresponds more closely to actual throttle requirements. Bands are usually long (25 in when the cross-sectional area is .010 in$^2$) and must be securely attached to the cylinder and piston. (The attachment force required is 2,000 lbs when the length is 25 in).

Storing 175 in-lb of energy ($W_k$) in a band requires a band volume of .25 in$^3$ (Volume = $\frac{2EW_k}{S^2}$) and a weight of .070 lb.

Bands of rubber and nylon can be used to store the required energy and bounce the piston. For evaluation of rubber as a spring material, silastic rubber was chosen because it is exceptionally resistant to solvents, jet fuels, and oils and is relatively strong as compared with other fluorosilicone rubber stocks. It is easy to handle in the unvulcanized state and may be processed by milling, calendering, extruding or moulding [3.4].

For a 2 in stroke and 175 lb force, the rubber band must be .53 in long and have a cross-sectional area of .13 in$^2$. It will occupy .069 in$^3$ and weigh .0036 lb. (It can store 50,000 in-lb/lb mass and 2,500 in-lb/in$^3$)*

*Reference [3.4] gives the typical physical properties of Silastic LS-2249V fluorosilicone rubber as follows:

- Tensile strength - 1370 psi
- Tear strength - 165 psi
- Elongation - 480%
- Specific gravity - 1.46
Its elasticity and high strength make nylon a suitable material for absorbing or storing energy. In stretching 2 in, a piece of nylon 5.7 in long (.035 in\(^3\) volume) can store 175 in-lb and will weigh .0012 lb.* (It stores 150,000 in-lb/lb mass and 5,000 in-lb/in\(^3\)).

Table IV summarizes the energy storing ability of the four materials considered, by listing the volume and weights required to store 175 in-lb of energy.

### Table IV

<table>
<thead>
<tr>
<th>Size of Material Required to Store 175 in-lbs of Energy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volume Occupied (in(^3))</td>
</tr>
<tr>
<td>----------------------------</td>
</tr>
<tr>
<td>Coil spring</td>
</tr>
<tr>
<td>Steel band</td>
</tr>
<tr>
<td>Silastic rubber</td>
</tr>
<tr>
<td>Nylon</td>
</tr>
</tbody>
</table>

A comparison shows that on a per pound basis, the energy storing capacity of nylon is 140 times greater than a coiled

* Reference [3.5] gives the following specifications for Extremultus belting:

- Tensile strength - 28,500 psi at 35% stretch
- E - 78,230 psi
- Density - 30 in\(^3\)/lb
- Heat resistance to +160°F
- Cold resistance to -20°F
spring, rubber is 47 times greater, and a steel band is 2 1/2 times greater. But in spite of its poor rating, a steel coiled spring was chosen to bounce the piston for these reasons:

1. the difference in weight between any of the materials is less than 3 ounces,
2. spring deflections are proportional to the load,
3. the fatigue life is very high for low stresses (nylon and rubber have a short life if compressions are high),
4. springs are easy to design.

Even though they are more efficient than round wire in the use of space, rectangular wire coiled springs require more expensive materials and more complex manufacturing techniques. Because it is not produced in tonnage, rectangular wire has not had the refining development given to round wire, so that the available quality is not equal to that of good grades of round wire.

Because they must not fail under an infinite number of loadings, the springs must be given the best in design, material and manufacture. Consequently round wire springs were chosen.

The choice of material for springs is usually between straight carbon (ASTM A230-47) and alloy steels.
Carbon valve spring material is preferred because its ultimate strength equals that of alloy steels and its dependability at ordinary temperatures is greater. Exposing carbon steel to temperatures in excess of 350°F will cause heat setting and loss of load. Alloy steels, on the other hand, usually are more subject to seams and have a greater tendency to quench-crack.

Carbon valve spring wire (ASTM A230-47) has an ultimate strength of 200,000-230,000 psi, chrome vanadium alloy steel (SAE 6150) has an ultimate strength of 200,000-250,000 psi (elastic limit is 180,000-230,000 psi), chrome silicon alloy steel (SAE 9254) has an ultimate strength of 250,000-325,000 psi (elastic limit is 220,000-300,000 psi), and oil-tempered spring wire (ASTM A229-41) which is most generally used, has an elastic limit of 120,000-250,000 psi.

In specifying a material for a particular part, availability, performance and cost are always connected. In research and development, cost is often of secondary importance to performance and availability. The desire for quick delivery limited the choice of wire materials to oil-tempered wire available locally and chrome silicon alloy steel available on a short order basis. Because it resists heat up to 450°F well, chrome silicon wire was chosen. Shot blasting was specified because the impact of the steel or glass balls prestresses and cold works the surface and so raises the physical properties of the material on the surface where the stress is the highest and where fatigue fractures will start.
When the initial design using a single spring was modified to two springs, a better arrangement resulted. The diameter of the scavenging chamber enclosing the cylinder spring was made larger than the piston diameter so that a shorter machine was possible. This added complexity to the design. The choice of dimensions was restricted by the fact that the springs

(a) operated within preset outside diameters,
(b) operated over preset shaft diameters,
(c) had specified maximum solid heights,
(d) had the same specified deflections, and
(e) required the same spring rate.

Because all these conditions were necessary and conditions changed whenever engine sizes were changed, whenever new spring materials were considered, and whenever new characteristics were desired, a large number of calculations were performed before the final sizes were selected. A sliderule designed by Associated Spring Corporation [3.6] to aid size selection reduced the time required for calculations so that a computer program for optimizing the size was not required. The sliderule was based on the following two formulas:

\[ S = \frac{8 \cdot PDK}{\pi \cdot d^3} \]
\[
\frac{P}{\delta} = \frac{Gd^4}{8D^3N}
\]

Where \( P \) is the load on spring,

\( D \) is the mean diameter of coil,

\( d \) is the diameter of wire,

\( S \) is the torsional stress,

\( K \) is the Wahl correction formula for additional stress caused by curvature of wire and shear load,

\( G \) is the torsional modulus,

\( N \) is the number of active coils,

\( \delta \) is the deflection.

When subjected to above normal temperatures, springs often shorten (or "set") and lose load. This loss of load can be predicted and allowances can be made in cases where the load is steady. Where the load is unpredictable and the spring design is not flexible enough, the springs may be preset by exposing them to temperatures and stresses above those encountered in operation. Since the spring design is flexible in the present design, no correction factor for set was specified.

*On page 25, reference [3.6] states that SAE 6150 loses 4% load when stressed to 80,000 psi at 350°F, 7% at 100,000 psi and 13% at 120,000 psi.
A characteristic sometimes critical is the natural vibration frequency of the spring. If it is too low the spring will surge and stresses will be greatly augmented. Since the natural frequency of the designed spring is 3.5 times the operating frequency, the effect of spring surging will have to be observed if problems arise.*

Compression springs in which the free length is more than four times the mean diameter can buckle. A calculation showed that the ratio of free length to mean diameter is well below 4 and therefore the designed springs should not fail due to buckling. This was confirmed by checking the deflection/free length ratio with the limit set by The Associated Spring Corporation. Deflection/free length ratio for the piston spring is .4 (limit is .70) and for the cylinder spring it is .5 (limit is .72) [3.6, p. 24].

The increase in diameter as the spring is compressed was calculated and found to be small. The maximum increase in diameter for the cylinder spring is .025 in and for the piston spring it is .011 in.

* Reference [3.6], page 22, suggests that the vibrations per min of a spring vibrating between its own ends can be given by:

\[ N = 0.21S \]

where S is the uncorrected stress (not including the Wahl correction factor) for a deflection of 1 in. For our case S was approximately 100,000 psi so that N is 21,000 cpm.
3.3.3 Synchronizing Mechanism

The piston and cylinder can be synchronized with connecting rods and crankshafts which are simple to design and inexpensive to fabricate. The ratio of piston stroke to cylinder stroke can be easily changed and suitable tolerances for the bearings can be easily held. Either standard roller bearings requiring oil to minimize heat generation and wear, or teflon impregnated self-lubricated bearings can be used. A typical self-lubricated bearing consists of a thin porous layer of spherical bronze impregnated with a mixture of TFE fluorocarbon resin and lead power [3.7]. The Free-Piston Company of Kingston found that for oscillating shafts, teflon impregnated bearings were better than roller bearings.*

Design calculations showed that a 1 in diameter by .5 long bearing can safely carry a shock load of 1,000 lbs without exceeding Garlock's [3.7] suggested limit of 2,000 psi. The service factor for 10,000 hours at an average operating load of 25 lbs is 9,500 and well below the suggested limit of 24,000. Even though the required cross-section area for the connecting rod is quite small (.04 in² for a stress of 25,000 psi), the required space to allow oscillation is not. The cylinder connecting rods, in order to oscillate freely, require a space 2 1/2 x 1

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*Unofficial discussions, 1965.
x 5 in, and the piston connecting rods require a space 2 1/2 x 1 x 1 1/2 in.

In a rack and pinion arrangement, the racks move over or into the coil of the spring so the arrangement requires less volume than a connecting rod arrangement. The gear service factor restricts the average stress to only one-tenth of the maximum stress, but significantly, the forces involved during starting and stopping are ten times higher than the average operating forces, so the gear material is used efficiently.

A rack and pinion arrangement was used in the first design. The pinions mated with teeth on the piston skirt and on the cylinder, and a single spring fitted inside the piston. After the layout drawings of the assembly and parts had been completed, the concept of placing the pinion gears between a divided spring was formalized. In spite of all the work performed on the first design, the original drawings were scrapped and a new design undertaken when the divided spring arrangement showed that a more compact engine was possible. In the new arrangement delineated, the gear mated with teeth on the piston rod and rotated in a stationary mount. In this arrangement the latch pivoted on the frame so that the force acting in the latch when the engine was stopped, is transmitted through the gear. The force is largest when the teeth on the latch engage the teeth on the blade.
When compared to the sliding action of the connecting rod bearing, the rolling action of a gear on a rack generates less heat and requires less lubrication. Because the load changes continually from one tooth to another and reverses at the end of each stroke, the gears would be noisy unless close tolerances were maintained. The gears designed for the rack and pinion took into account tolerances, geometry, load distribution, size, dynamic overload, service life, temperature effects and pitting durability. The analysis followed the American Gear Manufacturers' Association standards [3.8, 3.9]. The resulting gears (24 pitch, 20° spur with a pitch diameter of 3/4 in and a face width of 3/8 in), occupy a volume 3/4 x 2 x 7/8 in.

On completion of the design and delineation, a manufacturer for the small rack and pinion gears was sought. None of the local companies visited had the proper facilities and most of the outside companies contacted by mail were not interested in the small job. The synchronizing mechanism was therefore reappraised and a new arrangement sought.

Again the continuous quest for a compact, low-stressed design resulted in a new and valuable idea. This idea was adopted even though again it meant scrapping the old mechanism and initiating a new arrangement. In the new arrangement the stop latch was mounted on the cylinder so that it transmitted the stopping force directly to the cylinder and the initial spring compressing mechanism was
mounted on the piston so that it transmitted the starting load directly to the cylinder. These modifications reduced the force acting in the synchronizing mechanism by a factor of 10. Nevertheless, the size of gears could not be reduced because the service load had changed very little. But calculations now based on the smaller forces showed that the connecting rod arrangement could fit into space already designed for the rack and pinion arrangement. A 3/16 in diameter connecting rod bearing (Teflon impregnated) would have to be only 3/16 in wide for a service life of 10,000 hours and a unit load of 700 psi, [3.7].

The smaller forces also made chains, timing belts and bands feasible. For example, a Morse 1/4 pitch roller chain with an average tensile strength of 875 lbs transmitting 1.4 hp at 4,000 rpm and .5 hp at 8,000 rpm could be used [3.10]. Joint galling caused by friction between the pin and bushing as the joint articulates at high speeds, limits the speed of this chain to 10,000 rpm. This limit is higher for a chain in which the motion between metal parts during articulation is a rolling action rather than a sliding action. The Morse silent chain uses such a rocker-pin joint. A 3/16 in pitch 11/32 in wide silent chain has an ultimate tensile strength of 1,250 lbs and a load carrying capacity of 0.7 hp at 7,000-9,000 rpm.

Timing belts are quieter than chains or gears because the helically wound load-carrying cables are imbedded
in a rubber or a plastic covering. The teeth, an integral part of the covering, engage a mating sprocket and function like the teeth in a rack. The Morse 1/5 in pitch, 3/8 in wide, "XL" timing belt, when used with a 12 tooth, .764 pitch diameter sprocket, transmits .34 hp at a maximum recommended speed of 5,000 rpm, and occupies the same volume as a rack and pinion assembly. When used with an 18 tooth, 1.15 pitch diameter sprocket, the belt transmits .95 hp at 10,000 rpm.

A helically wound wire rope turning over a pulley can be used to carry the load. Because the strands cross the roller at an angle, the diameter of the strands can be larger than the allowable band thickness. For example 40 strands of .007 in diameter wire wound in a .060 in diameter rope can transmit a 35 lb force.

Steel or plastic bands flexing over a roller can synchronize the piston and cylinder while transmitting only a tension load. The total stress in the band will be the sum of the flexure stress due to bending over the roller, plus the tensile stress due to the load. The derived equation of force is:

\[ F = S W t = \frac{E t^2 W}{d} \]

Where \( t \) is band thickness

\( W \) is band width,
F is tensile load,
d is roller diameter,
E is the modulus of elasticity,
S is yield stress.

The equations for maximum load capacity and optimum band thickness can be found by differentiating this equation and equating the result to zero:

\[ t = \frac{S}{E} \left( \frac{d}{2} \right) \]

\[ F_{\text{max}} = \frac{S^2}{E} \left( \frac{Wd}{4} \right) \]

These equations show that the band can carry its maximum load when the thickness is such that the bending stress is equal to half of the yield stress, and that the best band material will be the one with the highest strength-to-stiffness ratio \( \frac{S^2}{E} \). The materials which best meet this criterion include some plastics and the metals commonly used in springs. The spring steels have a high yield stress and a high modulus of elasticity while the plastics have a low yield stress but also a low modulus. Consequently some plastics can carry more load than the steels. The plastics do have the disadvantage of plastic "creep", undesirable aging characteristics, poor stability under changing temperatures and often a large hysteresis loss.
To fit into the existing space, the roller diameter was restricted to about 3/4 in and the power band width was limited to 1/4 in (the power band was to be twice as wide as the return band). When these limiting dimensions are substituted into the above equations the maximum force and optimum band thickness is given by:

\[ F_{\text{max}} = 0.0469 \frac{S^2}{E} \]
\[ t = 0.375 \frac{S}{E} \]

Table V lists, for a number of materials, the values of maximum load and optimum band thickness based on these two equations. The table suggests that spring steel is preferable for reliability and high thermal stability and nylon is preferable for high load carrying capacity.

Because of its availability as well as its load capacity, nylon was the first material used in the design even though its melting temperature was only 400°F. The nylon bands were to be preloaded so that they would remain in contact with the rollers during all load conditions. For example, by preloading the "Habasit F-1" nylon power and return bands to 3.8 lbs, the power band load increased to 16 lbs (2.8% stretch) and the return band load decreased to zero (2.8% contraction) when 64 lbs were applied to the blade.
Table V

Capacity and Optimum Size of Bands

<table>
<thead>
<tr>
<th>Material</th>
<th>Sy</th>
<th>E</th>
<th>$F_{max}$</th>
<th>$\delta$</th>
<th>t</th>
<th>Remarks</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Preformed Havar</td>
<td>320,000</td>
<td>30x10^6</td>
<td>3.0</td>
<td>.01</td>
<td>.008</td>
<td>Watch mainspring steel</td>
<td>3.11</td>
</tr>
<tr>
<td>Havar (80% C.W. &amp; aged)</td>
<td>320,000</td>
<td>30x10^6</td>
<td>16.0</td>
<td>.0107</td>
<td>.0040</td>
<td>Watch mainspring steel</td>
<td>3.11</td>
</tr>
<tr>
<td>Spring steel e.g. SAE 1074</td>
<td>200,000</td>
<td>30x10^6</td>
<td>62.0</td>
<td>.0067</td>
<td>.0025</td>
<td>Special mold for 950°F</td>
<td>3.3</td>
</tr>
<tr>
<td>Fiberglass reinforced epoxy</td>
<td>200,000</td>
<td>7.5x10^6</td>
<td>350</td>
<td>.0100</td>
<td>Vibration damper</td>
<td>3.27</td>
<td></td>
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<tr>
<td>Nylon type 8</td>
<td>57,000</td>
<td>160,000</td>
<td>950</td>
<td>.0134</td>
<td></td>
<td>Melts at 400°F</td>
<td>3.28</td>
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<tr>
<td>Preformed spring steel</td>
<td>200,000</td>
<td>30x10^6</td>
<td>124</td>
<td>.005</td>
<td></td>
<td>Preformed to 1/2 inch dia.</td>
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<tr>
<td>Extremultus nylon</td>
<td>28,000</td>
<td>78,000</td>
<td>490</td>
<td>.136</td>
<td></td>
<td>Temp -20 to 160°F</td>
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<td>Habasit nylon</td>
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<td>45,000</td>
<td>3400</td>
<td>.0102</td>
<td></td>
<td>Temp -4 to 300°F</td>
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<tr>
<td>Mylar @ 77°F</td>
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<td>550,000</td>
<td>12</td>
<td>.0081</td>
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<td></td>
<td>3.28</td>
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<tr>
<td>@ 392°F</td>
<td>1,000</td>
<td>50,000</td>
<td>1</td>
<td>.007</td>
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<td></td>
<td>3.20</td>
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<td>Kapton @ 77°F</td>
<td>14,000</td>
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<td>28</td>
<td>.013</td>
<td></td>
<td>High thermal stability</td>
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<tr>
<td>@ 392°F</td>
<td>9,000</td>
<td>260,000</td>
<td>14</td>
<td>.013</td>
<td></td>
<td></td>
<td>3.28</td>
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<tr>
<td>Nylon 6/6 glass reinforced</td>
<td>30,000</td>
<td>1.8x10^6</td>
<td>24</td>
<td>.017</td>
<td>.0063</td>
<td>Heat distortion @ 500°F</td>
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</tr>
</tbody>
</table>

Unlike nylon which has an upper temperature limitation of about 300°F, some steel alloys such as SAE 1074 are suitable as bands even up to 950°F, although the usual spring materials are reliable only up to 400°F. SAE 1074 steel with a yield stress of 200,000 psi requires a band thickness of .0025 in to carry the maximum load of 250 lbs/in width when flexing over a 3/4 in diameter roller. The flat portion of the band will experience a distributed stress of 100,000 psi and the portion in contact with the roller will experience a bending stress of 100,000 psi superimposed on the distributed stress. This interesting fact led to the discovery that by prebending the band to twice the diameter of the roller, the band thickness and load could be doubled.
without exceeding the yield stress because in being straightened out the flat portion of the band was stressed in one direction and by being bent to conform with the roller diameter, the bent portion was stressed in the opposite direction. Because the stress alternates from compression to tension, early fatigue failure may result.

Because the flexing action of the band resembles the stressing action of a watch mainspring, the materials used as watch springs could probably be used as bands. In this regard it is significant to note that not until the development of the high strength cobalt base alloys such as Havar did unbreakable mainsprings become available. It is possible that Havar may prove to be the alloy best suited for band material not only because of its high strength (300,000 psi) and high temperature stability (strength remains high even above 1,000°F,) but also because of its resistance to "set" and stress corrosion [3.11].

The bands most easily available were high strength automotive feeler gauges. For a quick check as to the yield stress, a .005 in feeler gauge was bent over a 3/4 in diameter mandrel. Since the band did not deform plastically, the yield stress was in excess of 200,000 psi and therefore suitable as a band. Before the band thickness of .004 in was chosen, the load carried by the band was plotted as a function of band thickness, Graph 3.12. The graph shows that the flat band (.004 in) can carry 150 lbs without
exceeding the yield stress of 200,000 psi. A higher load will permanently deform the band, but not until the load reaches 480 lbs is there danger of failure. A .005 in band would permanently deform with any load but the load at failure is higher (700 lbs).

Graph 3.12 Maximum load transmitted by bands

The load carrying capacity of the bands can be increased by combining several flat bands into a composite band. The equation governing the load carried by the composite, derived from a force analysis, is given by:
\[ F = \frac{S}{2} \left( Wt \right) + \frac{2}{1+3\mu} + \frac{N-2}{1+6\mu} \]

Where \( N \) is number of bands,
\( \mu \) is the coefficient of friction,
\( W \) is the band width,
\( t \) is the band thickness,
\( S \) is the yield strength of band.

If each band is separated by a layer of low-friction material such as teflon or grease, the maximum load is nearly directly proportional to the number of bands because the low coefficient of friction allows each band to act nearly independently of the other bands. But if the friction is high, the composite band acts as a single thick band and the load capacity may even be less than a single thinner band.

The ratio of the rate of heat generated by friction to the maximum power is given by:

\[ Q' = \left( 4 \frac{F'}{P} + \frac{F'_b}{6} \right) \left( \frac{t}{D} \right) s' \mu (J-1) \]

Where \( F'_p \) is the load on blade/maximum allowable load on blade,
\( F'_b \) is the band pretension/maximum load on blade,
\( s' \) is the stroke/stroke at maximum load,
\( t/D \) is the band thickness/roller diameter,
\( \mu \) is the coefficient of friction,
\( J \) is the number of bands.
When the bands are prestressed so that at maximum load the return band carries no load, then:

\[
\frac{F'}{p} = \frac{3}{8}
\]

The amount of heat generated varies from .1% to .2% of rated power.

Up to now only the forces acting on the blade were considered. The question of what forces are present when the cylinder assembly mass does not equal the piston assembly mass was answered after the acceleration of the piston was equated to the accelerations of the cylinder. This yielded the following equation for the force in the band:

\[
F = \frac{(A_p)P}{2} P \Delta m
\]

Where \(A_p\) is the piston area,
\(P\) is the cylinder pressure,
\(\Delta m\) is the ratio of weight difference to sum of piston and cylinder weights \((\Delta m = \frac{M_p - M_c}{M_p + M_c})\)

The equation showed that a 3% difference in weight between the piston and cylinder assemblies would cause a 36 lb force to act in the band when the cylinder pressure was 2,000 psi. This difference in weight can occur as the result of sharpening the teeth (shortening them by 1/4 in). The resulting force will be carried by the single return band because the piston is lighter than the cylinder.
The bands can be fastened to the cylinder by forming hooks on each end of the band. The minimum radius for a cold worked hook is about 1/32 in so the maximum force the hook can carry is 6.7 lbs. To carry a larger force, the band can be heated and a smaller hook formed, or the hook can be clamped. Alternatives to the hook are drilled holes or welded brackets.

Among the methods of adjusting band tension considered, one required attaching the steel band to a flexible material such as nylon, another required adding some flexible material on the rollers or under the hooks, and a third required adding a setscrew-adjusted link on the end of the band. Because of its positive fastening and adjustment possibilities even under high temperature operation, the adjustable link was chosen.

3.3.4 Arresting Mechanism

In designing a suitable device to stop and lock the piston and cylinder, it was necessary to consider the magnitude of the forces involved in stopping the assemblies. For this purpose the energy absorbed by the device was equated to the energy released by the spring in expanding from the end of the stroke to the stopped position. The force equation became:

\[ F = 2Kx \left[ 1 + \frac{\delta}{\sum z_n} \right] \]
Where $K$ is the spring rate,
$x$ is the spring deflection,
$\delta$ is the displacement from BDC to point latch engages,
$Z_n$ is the deflection of nth member,
$F$ is the load on device.

The force is smallest when the device locks the piston at the end of the stroke ($\delta=0$). But because of impact, the magnitude of the instantaneous force is twice the steady state magnitude.

A possible locking device is a one-way clutch. It allows very little motion before locking but it requires a very large force to unlock and a mechanism that will allow the clutch to engage only at the end of a selected stroke. A latch or ratchet allows some movement before the teeth lock, so the force involved in stopping the assemblies is higher than for a one-way clutch. If the energy absorbing member is flexible ($Z_n \gg 0$), the forces involved are not excessive. By calculating the deflection of each member in terms of stress (assuming a constant stress-strain relationship), the force equation becomes:

$$F = Kx + \sqrt{(Kx)^2 + \frac{2(Kx)\delta}{\sum A_n E_n}}$$
Where \( L_n \) is the length of \( n \)th member,
\( A_n \) is the area of \( n \)th member,
\( E_n \) is Young's modulus for \( n \)th member.

This equation can be used when the size of members is known but the stresses are not. For example, when a 3/16 in diameter steel rod, 2.8 in long stops the piston in .10 in \((Kx=175)\), the rod will deflect .005 in, be loaded to 1,380 lbs, and experience a stress of 50,000 psi. But if the steel rod rests on a 3/16 in diameter nylon rod, .25 in long, the assembly will deflect .054 in and each member will be loaded to only 990 lbs and experience a stress of 36,000 psi.

When the size of all members is not known but all the maximum allowable stresses are, then the following rearranged equation can be used to calculate the unknown dimension:

\[
\sum \frac{S_n L_n}{E_n} = \frac{\delta}{S_i A_i} - 1
\]

For example, when used to calculate the length of a nylon rod so that the stress in the 3/16 in diameter by 2.8 in long steel rod will not exceed 30,000 psi, and the stress in the nylon rod will not exceed 20,000 psi, the equation gave the length as 5/8 in and the diameter as 1/4 in. In this example the rods will transmit a force of 830 lbs.

In the design of the teeth on the latch, the fact
that the coefficient of sliding friction is less than the
coefficient of static friction was used to advantage. The
angles of the teeth were chosen so that the force of fric-
tion holds the latch in the locked position when the latch
is stationary and helps to open the latch when it is
already starting to open. The force of friction will hold
or release the latch as required when tangent of the tooth
angle is between the static and sliding coefficient of
friction:

$$\mu_{\text{sliding}} < \tan \theta < \mu_{\text{static}}$$

But the coefficient of friction depends on the
type and quantity of lubrication used and on the surface
finish. The coefficient of static friction of steel-on-
steel is .78 when dry and .23 when lubricated with a light
mineral oil; the coefficient of sliding friction of steel-
on-steel is .42 when dry and .080 when lubricated with a
castor oil or grease [3.12]. Therefore, the tooth angle
should be between 23° and 38° when the teeth are dry and
between 4 1/2° and 13° when the teeth are lubricated with
oil. Consequently, the latches designed for operation with
lubrication will require a large force to open them when the
teeth are not lubricated, and latches designed for operation
with no lubrication will fly open by themselves when the
teeth receive lubrication.
A force of 10 lbs is required to break an oiled latch with 10° teeth free from the locked position and no force is necessary to keep it moving. A force of 107 lbs is required to break free a dry latch with the same tooth angle and a 44 lb force is needed to keep it moving. As it opens, the dry latch will dissipate 2.2 in-lb of energy.

The large force required to open the latch (e.g. 107 lbs when dry) could make starting the engine difficult. This difficulty was avoided by using a coil spring, pre-tensioned by the motion of the cylinders, to open the latch when the trigger released the extended spring, Figure 3.3. The operator starts the engine by pulling a small flat spring out of a slot; once it is no longer locked by the flat spring, the coil spring contracts, pulls the level, slides through an arc, and pulls the latch away from the piston rod, thereby freeing the piston rod and cylinder. The operator initiates the stopping action by releasing the flat spring; once the flat spring locks in the slot, the motion of the cylinder latch assembly extends the coiled spring, pulls the lever through an arc, forces the latch to slide over the teeth on the piston rod, and engages the teeth when the motion reverses. To allow the latch teeth to follow the contour of the rod teeth and to impart an impact load to the slider, the lever was hinged.

The coil spring attached to the hinge must supply the force required to overcome the locking force and keep
the latch in contact with the rod teeth. The force on the latch, the inertia of the latch-slider assembly, and the geometry will determine the distance the piston and cylinder travel from the time the teeth tops separate until the teeth flats touch. For the size of latch used, the distance $(\ell)$ depends on the spring force $(F)$ according to the equation:

$$\ell = \frac{0.022}{F} \text{ [in]}$$

A 1 lb force will cause the teeth to engage fully when the cylinder-piston assembly travels at least .022 in after the teeth tops have separated. If the cylinder-piston reaches bottom dead center before it has travelled .022 in, the teeth will not be fully engaged. A coil spring with a rate of 6 ppi, will release 2.2 in-lb of energy as it shortens from 1 in to 1/2 in, thus giving the required force and releasing the required energy.

The calculated stresses in the slider and piston rod during impact showed the necessity of specifying steel as the required material. The maximum stress in the slider is 31,000 psi and in the piston rod it is 50,700 psi. (The rod stress in the same location during full load is 29,300 psi). The teflon bearing inside the slider must be 3/16 in long to safely transmit the 109 lb shock load.
3.3.5 Cooling System

In designing efficient cooling fins, no mechanism is as important as the heat exchange between a solid body and air. The heat transfer, according to Mackerle [3.13], varies directly with the tangential surface friction force and inversely with the velocity. If the surface friction force is expressed in terms of air velocity, the heat transfer varies directly with the air velocity when flow is turbulent, and with the square root of the velocity when the flow is laminar:

\[ Q = kV^{n-1} \]

Where  
\[ n = 1.5 \] for laminar flow,
\[ n = 2.0 \] for turbulent flow,
\[ n = 1.73 \] for part turbulent and part laminar,
\( k \) is a constant,
\( V \) is the mean velocity of flow.

Since the heat transfer is most intense through a turbulent boundary layer, the sooner the transition from laminar to turbulent flow takes place, the better the average heat transfer per unit heat will be.

Twenty to twenty five percent of the heat released in the cylinder in a high-speed two-stroke engine is removed by the cooling air. The purpose of cooling is to secure a sliding surface for the piston which is moving at an average speed of 40 fps. This surface is flushed by hot gases whose
maximum temperature at the onset of expansion may reach 4,000°F. The conditions for the formation of a lubricating film are adverse, so that the temperature of the walls must be maintained below 400°F for normal operation. At elevated temperatures lubrication becomes irregular and the piston rings suffer from excessive wear. It is a common practice to supply power saws with a rich fuel-air mixture to cool the engine internally and to provide adequate lubrication.

Some of Mackerle's conclusions from the results of his tests on the heat transmission from air cooled engines concern the surface coefficient of heat transfer (h) and are listed below:

1. The height of fins normally used has no influence on the surface heat transfer coefficient.
2. The fin thickness has no significant influence.
3. The fin spacing (Z) influences the coefficient, according to the relation:
   \[ h \propto Z^{0.32} \]
   but the coefficient deteriorates considerably at spacings below .060 in.
4. The air velocity (V) influences the coefficient according to the approximate relation:
   \[ h \propto V^{0.73} \]
5. A matte surface has definite advantages over a glossy one whereas a punch marked surface (.020 in deep,
.070 in apart) has no advantage over an unpunched one. Although the surface quality influences the coefficient, it is not true that heat is always better transmitted by a roughly cast surface than by a machined one. In the case of large projections on the surface, increased resistance to air flow is not caused by surface friction but by pressure head differences. This type of resistance impedes heat transfer.

6. The cylinder diameter influences the coefficient because the ratio of surface areas exposed to laminar and turbulent flow varies with the diameter.

7. The air stream direction influences the coefficient because if the air flow between the fins is disrupted and intensive turbulence in the fin interstice results, heat transfer is considerably augmented.

8. Rapid vibration of fins has little influence on the coefficient.

The extent to which Mackerle's observations on steady air flows over stationary cylinders also apply to reciprocating cylinders moving in stationary air is yet to be determined.

Even though the theoretically ideal fin has sides in a parabolic shape terminating in a sharp edge, the best practical compromise, according to Judge [3.14] is a truncated conical fin with rounded edges. In practice, the fin
thickness is chosen as small as production processes permit. There should be as many fins as possible but the interstices must be large enough to ensure a satisfactory flow of cooling air. For small air velocities Mackerle recommends a spacing of .31 to .47 in for a freely exposed cylinder having no cowling. If a stronger air flow is available for cooling, fins are more densely spaced. But if the interstices are smaller than twice the laminar boundary layer thickness, a rapid deterioration of flow conditions and fin efficiencies occurs. At an air speed of 127 fps, Mackerle found that efficiency drops off rapidly at interstices below .10 in.

The simplest method of cooling the cylinder uses the inherent reciprocating motion of the cylinder. A literature search revealed very little valuable information on this type of heat transfer. In a survey article on the effects of vibration and sound on heat transfer, Richardson [3.15] surmised that under some circumstances heat transfer by oscillations would exceed that by forced convection when compared on a mean velocity basis. But the required amplitudes of oscillations are so large that they may not be obtained in practice and on a cost and weight basis, forced convection is an outright winner.

To generate enough information for a valid evaluation of this type of heat transfer and to determine the amount of heat transferred from a reciprocating cylinder
head, an experiment was set up and tests were performed. The apparatus consisted of a heater element, thermocouples, standard and special cylinder heads, and a mechanism to reciprocate the head at varying speeds and through several amplitudes. Photographs of the test apparatus are shown on Figure 3.2 and the data is given in Appendix VI.

Calculations for the heat transfer coefficient \( h \) were based on the measured values of the power supplied to the heater, the fin temperature, the air temperature, and the fin area. The data is plotted on Graph 3.13. Of all the co-ordinates tried, the ones used on the graph (Nusselt number \( \text{Nu} \) versus the product of the Reynolds number \( \text{Re} \) and the square root of the fin height-to-stroke ratio \( \ell/s \)), best correlated all the available data. The equation obtained from the plot is given by:

\[
\text{Nu} = 10.8 + 0.0862 \left( \text{Re} \sqrt{\frac{\ell}{s}} \right)^{0.986},
\]

Where \( \text{Nu} = \frac{h\ell}{K} \),
\( \text{Re} = \frac{N\ell}{\nu} \)
\( K = \text{conductivity of air} \),
\( \nu = \text{kinematic viscosity of cooling air} \).

The graph was then used to calculate the expected temperature of the designed cylinder. For an ambient temperature 70°F, (frequency \( N \)=6,000 cpm, stroke \( s \)=.75 in, fin height \( \ell \)=
Figure 3.2 Photograph of heat transfer test apparatus:
(A) Model 210 head with heating element and spacer, (B) Model 275 head, (C) 275 head and block with heating element (D) 210 head assembled on scotch yolk, (E) test set-up (F) special cylinder head and mating stationary fins assembled on scotch yolk
1.30 in, fin area 100 in\(^2\) the fin temperature may go up to 430°F. This means that the fin temperature in the free-piston engine will be approximately the same as in the conventional power saw.

Graph 3.13 Heat transfer data for reciprocating cylinder heads
3.3.6 Combustion System

The need for a lightweight cylinder assembly to keep overall weight low and operating speed high, limited the choice of the cylinder and fin material to the aluminum alloys shown on Table VI. The final choice was Alcan 135 as it was available locally and gives satisfactory service in conventional power saws.

Table VI

Materials Suitable for Cylinder Heads and Blocks

<table>
<thead>
<tr>
<th>Form</th>
<th>Alcan 135 ASTM S470n</th>
<th>Alcan 225 ASTM C44</th>
<th>Alcan 250 ASTM C4010A</th>
<th>Alcan 385 ASTM 4032</th>
<th>Alcan 125 ASTM 5651</th>
<th>Alcan 318 ASTM C516</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat treatment</td>
<td>Sand &amp; P.Mold</td>
<td>Sand</td>
<td>Sand &amp; P.Mold</td>
<td>Forged</td>
<td>Sand &amp; P.Mold</td>
<td>Sand</td>
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<td>35</td>
<td>39</td>
<td>35</td>
<td>37</td>
<td>36</td>
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<td>22</td>
<td>25</td>
<td>29</td>
<td>40</td>
<td>25</td>
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<tr>
<td>Yield strength ($\times 10^{-3}$)</td>
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<td>27</td>
<td>28</td>
<td>24</td>
<td>35</td>
<td>27</td>
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<td>75</td>
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<td>110</td>
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<tr>
<td>Shear strength ($\times 10^{-3}$)</td>
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<td>27</td>
<td>30</td>
<td>27</td>
<td>30</td>
<td>89</td>
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<td>Fatigue limit ($\times 10^{-3}$)</td>
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<td>8</td>
<td>6.5</td>
<td>9.5</td>
<td>8.5</td>
<td>16.5</td>
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<td>Yield at 400°F ($\times 10^{-3}$)</td>
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<td>(3.18, 3.19)</td>
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<tr>
<td>Uses</td>
<td>Cylinder head and block, reciprocating parts in engines, crank case</td>
<td>Cylinder head pistons, bushings</td>
<td>Pistons</td>
<td>Cylinder head heavy duty timing gears, piston and cylinder heads</td>
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</tbody>
</table>

Because the rings require a special sliding surface, the cylinder bore had to be anodized, chrome plated, metal sprayed or lined with a cast iron sleeve. To minimize
weight and maximize reliability, chrome plating to a depth of .001 in was originally specified but when, during fabrication, a machining error destroyed the surface to be plated, a .040 in wall cast iron sleeve was specified. For a room temperature interference fit of .003 in on the outside diameter (specified to prevent the liner and cylinder block from separating even when the cylinder temperature reaches 500°F) the contact pressure is 1,150 psi and the liner bore is .0017 in smaller. To allow for the removal of .0010-.0015 in thick material in the final honing process, the machined bore diameter was specified .001 in undersize.

To estimate the maximum stress in the cylinder walls, the cylinder head was assumed to act as a flat circular plate, subjected to a uniform 2,000 psi pressure and simply supported around the edge. The calculated stress, neglecting the stiffening effect of the fins and walls, was 12,000 psi. The shear stress around the cylinder head was 2,500 psi and the tensile stress in the walls was 5,000 psi. The tensile stress was higher at points of stress concentration, and because the accelerated mass decreased, the stress was lower farther away from the head. The stress due to the load on the blade (140 lbs) was 560 psi and due to the spring load it was 800 psi.

To achieve a very high compression ratio before the piston reaches the top mechanical limit, the combustion chamber and piston face must remain flat. In an attempt to
determine how adversely this requirement would affect the heat transmission, the product of the exposed area and the calculated temperature difference between the wall and the gas was plotted as a function of time for a conventional power saw with a conical combustion chamber and a free-piston saw with a flat combustion chamber. Although combustion was assumed to occur instantaneously at a compression ratio of 7.4 in both engines, the maximum compression ratio in the free-piston engine is a function of load. Temperatures were taken from Taylor's [3.1] graphs of thermodynamic properties of gasoline-air mixtures or products of combustion at realistic air-fuel ratios and proportions of unburned fuel.

The result, plotted on Graph 3.14, indicates that because of higher piston speeds in the free-piston engine, less heat transfer will take place in the free-piston engine as the compression ratio goes up. For example, if the heat transfer coefficient remains constant, then at a compression ratio of 7.4 the heat transmission from the free-piston engine will be about the same as from the conventional engine, but at a compression ratio of 22.4 the transmission in the free-piston engine will be halved because the power stroke will be completed in much less time. So at part load, the heat transfer from the free-piston engine should be considerably less than from the conventional engine.
Tests performed by the author [3.16] on the conventional engine yielded the following observations when the conical combustion chamber was replaced with a flat one:

1. power went down 5-12%,
2. exhaust temperature dropped,
3. cylinder head temperature went up (led to detonation at high speeds),
4. scavenging efficiency went down.

From the results it was concluded that on standard engines the heat transmission from a flat head is higher than from a conical head.

Since the piston assembly was to weigh the same as the cylinder assembly (so that the piston stroke would equal the cylinder stroke), the choice of a material for the piston was limited to heavy metals such as nodular cast iron (ASTM 100-75-04), malleable cast iron (ASTM 80-002), nitriding steels (class N) or high strength alloy steels. The characteristics of these metals are described on Table VII. When the decision was made to fabricate the piston and rod in one piece, the strength requirement of the rod limited the choice of metals to high strength alloys. The final choice was AISI 6150, a tough material commonly used for shafts, gears, liners and pistons.

In existing free piston engines the extremely high accelerations and rapid pressure changes near top dead point leads to insufficient gas pressure under the rings. This results in erratic "flutter" and eventual breakage of the rings [3.21]. High thermal loading and marginal lubrication also shorten ring life. The Ford Motor Company
### Table VII

**Materials Suitable for Pistons**

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<td><strong>Density (lb/in³)</strong></td>
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<td><strong>Uses</strong></td>
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<td>pistons &amp; cylinders</td>
<td>pistons con rods cylinders</td>
<td>gears, pistons</td>
<td>gears, diesel pistons</td>
<td>cylinder liners, bushings</td>
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</table>

[3.22] overcame this problem of ring fracture and burning by reducing the pressure (originally 4,000 psi), changing fuel injection timing, employing porous-chrome plated cylinder bores, and using steel instead of cast iron rings.

Rings can be made of cast iron, steel alloys and compounded forms of TFE fluorocarbon. Cast iron rings, used extensively in conventional piston engines, are inexpensive and if properly lubricated, give long trouble-free service. In the manufacturing process the ring can be covered with an oil absorptive coating that facilitates rapid seating and
retards scoring and scuffing. Alternately, the cylinder can be plated with tin or cadmium to induce seating and retard scoring and scuffing. For added protection against abrasion, the ring faces can be plated with chromium and for protection against scuffing they can be sprayed with molybdenum.

Rings made of a TFE fluorocarbon (teflon) compounded with special wear resistant materials for long life, are expensive, have a low coefficient of friction (0.4 dry) and require no lubrication. Because it conforms to the eccentricities of the bore and is forced against it by a metal expander, teflon makes a good seal up to 500°F and reduces gas leakage to a negligible amount. If the ring can be made without a gap (necessary in cast iron rings to allow for thermal expansion), then the continuous seal will reduce blowby. This will reduce the discharge of pollutants into the air and increase the compression ratio before metal-to-metal contact occurs. A number of 100 hour tests at 1,800 to 4,200 rpm performed at the Du Pont Laboratory [3.23] showed that teflon rings in gasoline engines were practical.

Sealing with cast iron rings can be improved by making the headland (the area between the piston head and the conventional top ring), part of the top piston ring. With this change the Sealed Power Corporation [3.24] claims to have decreased blowby by 50%. Extending the ring up to the piston head seals the annulus between the piston and cylinder wall. Although the L-shape permits the compression-
gas pressure to effect a better seal, the larger area over which the pressure acts increases the friction force, and the direct contact with high temperature gases raises the ring temperature.

The gas pressure under the ring and the internal pre-tension combine to effect a good seal in the conventional design. To keep the ring temperature low so that set does not remove the pretension, the conventional ring is placed some distance away from the hot gas and to get an equal pressure drop across each ring, the land diameter is made smaller than the skirt diameter. These conventional practices were followed in designing the free-piston power saw.

For satisfactory combustion to occur, the air-fuel charge must be well mixed, in the correct ratio (14.6 for a stoichiometric charge), and brought up to ignition temperature by a flame, a spark, a hot spot, or a high compression pressure. A lean mixture requires a higher ignition temperature.

When the intake is throttled, the air-fuel charge entering the cylinder mixes with the residual exhaust gases and is diluted. For rapid and complete combustion, the charge entering must be rich or the ignition energy must be high. In the conventional engine the overall mixture is enriched, in the stratified charge engine the portion of the charge in the vicinity of the spark plug is enriched [3.25], and in the free-piston engine the compression ratio is automatically increased.
The carburetor specified was a conventional diaphragm type with a 3/8 in diameter venturi. The carburetor receives its pressure pulses from the scavenging chamber through a tube in the mount, and its fuel from the tank in the handle through a tube in the frame. To limit the magnitude of the pulses during part-load operation, the hole opening to the chamber was located so that it would be closed by the cylinder cap when the stroke increased to about 3/4 in. Consideration was given to designing a special carburetor fitting inside the engine. In this carburetor a fluidic device could meter the fuel [3.26]. But for reliability during the initial tests a standard carburetor was thought to be best.

The felt pick-up in the tank draws fuel into the lines from where it is pumped into the carburetor by the diaphragm. An insulator block placed between the carburetor and frame reduces heat transfer to carburetor and retards the vapour-lock. An air filter made of wire screen covered with a rayon flocking restricts the entrance of dirt and foreign material; air entering the filter cap must flow horizontally, so material cannot fall onto the filter.

Although designed for an AISI 1010/1015 steel sheet, the muffler was made of aluminum, to facilitate machining in the shop. The reciprocating motion moves cooling air over the exposed parts and prevents twigs and leaves from settling on the hot surface. As a purpose of the muffler is to break up the large carbon particles, the muffler was constructed
so that the escape outlet was at right angles to the exhaust port opening. Carbon sparks impinge directly against the side and top wall of the muffler and then work through the perforations in the screen to the second chamber where they again impinge on the wall of the muffler before working through the second group of perforations. These impingements break the carbon particles into sizes that will not start a fire. Thirty-six holes (3/32 in diameter) gave a perforation area of .25 in$^2$.

This completed the design of the important components. It was of course necessary to design the small elements but their delineation was straightforward and will not be described. Figure 3.3 shows the final assembly layout.

3.4 Fabrication of Parts

In the development of a new concept as complex as a two-stroke engine, modifications are not limited to the theoretical design stages. Even during the fabrication process many valid modifications suggest themselves, some to be incorporated immediately and others to be incorporated in future models. The modifications can be grouped as follows:

R - revisions that simplify fabrication or improve performance,

C - corrections to design or drawings,

T - temporary changes to make revised parts fit existing components, to accept machining and
Figure 3.3 Free-piston power saw assembly sketch
casting errors, and to allow for the instrumentation of the unit.

1. (T) To give the rings a good bearing surface while keeping the weight low, the initial design specified chrome plating on the cylinder bore. When an undercut was accidentally made in the bore so that plating was no longer possible, the cylinder was modified to accept a cast iron sleeve.

2. (T) The distances from the tops of the exhaust holes to the tops of the transfer holes (blowdown) were machined .050 and .085 in instead of .100 in. Because the transfer ports opened too soon, the exhaust gases flowed into the scavenging volume and prevented efficient scavenging. To correct this machining error, the low transfer port heights and the exhaust ports were raised .080 to give a .100 blowdown. The machining operation is shown on Figure 3.4.

3. (T) Because the piston skirt was the same width as the exhaust port width, a small misalignment in the angular location of the skirt resulted in a hole between the scavenging chamber and muffler. Plastic steel was used to fill the hole.

4. (T) Because they allowed the inspection of the transfer ports, temporary removable plates were substituted for expansion plugs in the transfer port holes. In the process of drilling the transfer holes the
drill came through the material so the area was built up with weld material. A comparison of the wooden pattern with the drawing revealed an error in the pattern.

Figure 3.4 Photograph of transfer port machining operation

5. (C) When the mount was lengthened by .050 in during a design change, the corresponding rod length was inadvertently left unchanged. This error required machining clearance slots for the blade.

6. (T) Because proper surface grinding facilities for correcting heat treatment distortion were not available, the piston rod was accepted unhardened (yield stress 100,000 psi instead of 200,000 psi when hardened to Rc 40). The blade slot was machined
.132-.140 in wide instead of the specified .125 in, the screw location was off by about .020 in, and the blade spigot was .120 in diameter and tapered instead of .125 in and parallel to the blade. Consequently, the saw blade was not parallel to the piston rod (varied 1/2° to 3°). The blade was made from an existing saw blade and screwed to a large backing plate (made large to balance the weight of the cylinder).

7. (R) The original design for attaching the band to the piston rod called for a milled slot on the rod. An inspection of the completed rod revealed that the slots had not been machined perpendicular to the rod, the inside edge of the slot had been machined convex and irregular, and the slots were not cut directly opposite each other as specified on the drawing. An attempt was made to manually square the slots but an installed band failed after a few cycles. To rectify the problem a new slot was milled across the rod but again the edge was not smooth, perpendicular nor round. Even an undercut did not prevent another band from failing. For the next attempt, spot faces for dowels were end-milled at the center of the slot. Even though the drawings had specified close tolerances for the important surfaces to prevent the previous problem from re-occurring, when completed, one spot
face was machined inaccurately. The location was off by .014 in, the diameter of the same spot face was larger by about .005 in and the edge of the spotface was 1° from the perpendicular. To salvage the piston, the diameter of the offending spotface was enlarged to take a shim. With this modification, the attachment as shown on Figure 3.5, performed satisfactorily but the piston rod had a number of sharp edges and thin sections, so it failed when driven by a fixed-throw crankshaft.

8. (R) Although the ends of the first bands were bent into hooks which clamped over the attachment plate, most of the load was to be carried by friction when the band was squeezed between the attachment plate and the cylinder. The rivets locating the bands were replaced with screws to allow dismantling and to increase clearance under the attachment plate.

9. (R) When the nylon band was replaced with a steel band, the roller diameter was increased to give a small clearance between the piston rod and roller (to stop a piston from twisting) and to keep band tension constant throughout the stroke.

10. (T) When the first springs were tested, the spring rates, instead of being 175 ppi as specified, were only 147 to 150 ppi and oversize. When the springs were ground down to the specified size, the rate had
Figure 3.5 Photographs of the synchronizing and arresting mechanism assemblies
decreased to 140 ppi. Consequently two back-up springs were designed to fit inside the original ones and a new set of springs were ordered. The sharp edges of the spring ends were rounded so that the bands would not be cut and the flow path would be smoother. The final test engine used a double spring under the piston (K=158 and K=32 ppi) and a single spring in the cylinder (K=190 ppi).

11. (R) The new arresting mechanism, devised when the rack and pinion arrangement was replaced by the band and roller arrangement after the cover had already been cast, required a new trigger assembly which pushed instead of pulled, and some minor modifications to the existing cover so the mechanism would fit.

12. (T) Not until the arresting mechanism had been assembled and tried out was it established that the lever arms to the slider were .100 in shorter than the drawing called for, so new ones were made. When it became obvious that the teflon tape could not be bonded to the slider with the available epoxy, a bronze bushing was brazed into the slider. The springs which pulled the slider were doubled one inside the other to conserve space. A guide for the levers was added so that the latch remained parallel to the piston rod while the engine was running.
13. (T) Because the cover as received from the foundry did not have sufficient material on the bar for the mount, the required material was added by welding and a hole was drilled through this welded material to connect the fuel line with the carburetor.

After the fabrication and modifications had been completed, the parts were assembled into the complete free-piston power saw. Figures 3.6 and 3.7 show the finished parts and their relationships to each other. The completed saw was instrumented and put on a test bench for concept evaluation, as described in the next chapter.
Figure 3.7 Photographs of FPS components
4. EVALUATION

4.1 Performance Characteristics of the Free-Piston Power Saw

While the engine was being fabricated and the tests were being performed, the computer program was being refined to allow more accurate representation of engine thermodynamics and to permit more conditions to be investigated. The refinements led to the following assumptions:

1. The pressure in the first ring groove is equal to the cylinder pressure and in the second ring groove the pressure is equal to half of the cylinder pressure, [3.1]. In addition to the pressure forces, a constant ring pre-tension force acts on the rings.

2. The combustion efficiency below a fuel/air ratio of .068 is constant at 95% and above .068 it decreases linearly with the fuel/air ratio to 54% at a ratio of .100. No combustion takes place when the mixture ratio is below .05 and above .143 [4.1].

3. The coefficient of friction doubles when the velocity drops below 10 ips.

4. The minimum volume at the top end of the stroke when the piston touches the cylinder head includes the volume under the first ring, half of the volume under the second ring, the annular volume between the piston
head-land and cylinder, half of the volume between
the rings, and the volume inside the glow plug.

5. The damping coefficient varies inversely with the
temperature according to the following relationship:

\[ \frac{r}{r_{ref}} = e^{(T_{ref} - T_{fin}) \cdot 0.00356} \]

\[ \frac{r}{r_{ref}} \text{ changed from } 2. \text{ at } 100^\circ F \text{ to } .5 \text{ at } 500^\circ F \]

Where \( T_{ref} \) = reference temperature,
\( T_{fin} \) = fin temperature.

6. The overall effective ring gap area varies in direct
proportion to the thermal expansion:

\[ A_1 = (\text{GAP} + 0.28 \times 10^{-4} (T_{fin} - T_{ref})) \times \]
\[ (\text{DD1} + 0.89 \times 10^{-5} (T_{fin} - T_{ref})) \]

\[ A_{\text{effective}} = \frac{A_1 A_2}{A_1 + A_2} \text{ for the two rings.} \]

(If an aluminum piston and an aluminum cylinder is
used, the clearance between the piston and cylinder
(DD1) does not change.)

7. Combustion starts when the gas temperature in the
cylinder reaches a pre-set value or when the piston
comes within a pre-set distance from the head, and
continues at a pre-set rate (based on the speed and
the amount of combustible mixture present) until all available fuel has ignited or until the pre-set detonation temperature is reached. As soon as the detonation temperature is exceeded, the temperature increases very rapidly. The rate of temperature increase during detonation was chosen to give complete combustion in the conventional engine during a 3° crank rotation [4.1].

8. Jaklish's heat transfer equations represent the amount of heat transferred from the gas to the cylinder, [4.2, 3.13],

\[
\alpha = \frac{(1 + 0.0063 \, C_{\text{mean}})}{0.001685} \sqrt[3]{\frac{P_{\text{cy}}^2}{T_{\text{cy}}}}
\]

\[
\dot{Q} = \alpha \left(2\pi \left(\frac{\text{BORE}}{2}\right)^2 + \pi \text{BORE} \, x \right) \left(T_{\text{cy}} - T_{\text{fin}}\right)/144 \, \text{[Btu/hr]}
\]

Where

- \(C_{\text{mean}}\) = mean piston speed (fpm),
- \(\text{BORE}\) = piston diameter (inch),
- \(x\) = piston position (inch),
- \(P_{\text{cy}}\) = cylinder pressure (psi),
- \(T_{\text{cy}}\) = cylinder gas temperature (°F).

9. The amount of heat transferred from the cylinder to the cooling air is given by the following formula which had been determined experimentally:

\[
\dot{Q} = A_{\text{fin}} \left(1.48 + 0.00257 \, \text{Speed} \cdot 985 \, (X_{\text{max}})^{1.477}\right) \left(T_{\text{fin}} - T_{\text{atm}}\right) \, \text{[Btu/hr]}
\]
Where \( A_f \) = fin area (\( \text{ft}^2 \)),

\[
\text{Speed} = \text{engine speed (cpm)},
\]

\[
X_{\text{max}} = \text{piston position at BDC (inch)},
\]

\[
T_{\text{atm}} = \text{ambient temperature (°F)},
\]

\[
T_{\text{fin}} = \text{fin temperature (°F)}.
\]

10. The fin temperature is based on an energy balance between the heat transfer from the gases (including the heat produced by friction) and the heat transferred to the cooling air.

11. Mackerle's [3.13] experimentally derived flow coefficients are valid for the flow through the ports, \((C=0.5)\). Perfect mixing occurs and the flow is given by the following equations:

(a) for sub-critical flow

\[
\dot{W}_f = 0.923 \text{ C Area} \left( \frac{P_{\text{up}}}{T_{\text{up}}} \right) 5.72 \sqrt{\left( \frac{P_{\text{dn}}}{P_{\text{up}}} \right)^{1.482} - \left( \frac{P_{\text{dn}}}{P_{\text{up}}} \right)^{1.744}}
\]

(b) for critical flow

\[
\dot{W}_f = 0.923 \text{ C Area} \left( \frac{P_{\text{up}}}{T_{\text{up}}} \right)^{0.58}
\]

Where subscripts "up" and "dn" refer to upstream and downstream.

12. The following forces act on the piston and cylinder:

(a) Gas forces due to pressures in the cylinder and in the precompression chamber (= \( A_p (P_{\text{cy}} - P_{\text{pc}}) \)).
(b) Ring friction force due to ring pretension and
gas pressure under the rings (= \( \mu \left[ P_{cy} A_r + \left( \frac{P_{cy}}{2} \right) A_r + F_r \right] \)).

(c) Load on the blade (= \( F_\ell \)).

(d) Friction force due to loading force, (= \( \frac{F_\ell}{2} \)).

(e) Spring force (= kx).

The force equation is:

\[
F = (P_{cy} - P_{pc}) A_p - \mu \left( \frac{3}{2} P_{cy} A_r + F_r + \frac{3}{2} F_\ell \right) - kx
\]

Where
- \( A_p \) = piston area (in\(^2\)),
- \( P_{cy} \) = cylinder gas pressure (psi),
- \( P_{pc} \) = precompression gas pressure (psi),
- \( \mu \) = coefficient of friction,
- \( A_r \) = piston ring area (in\(^2\)),
- \( F_r \) = ring pretension force (lb),
- \( F_\ell \) = load on blade (lb),
- \( k \) = spring rate (ppi),
- \( x \) = piston stroke (in).

Performance characteristics during preliminary tests were obtained from the prototype which was instrumented with a crankcase pressure transducer, a cylinder head accelerometer, a cylinder head thermocouple, and a photo-electric position indicating mechanism. This mechanism consisted of a light source and a photo-electric cell and was mounted so that the blade position determined the amount of light reaching the
cell. When the start-stop trigger was pressed, a contact switch triggered an oscilloscope which displayed the piston position as a function of time. Photographs of the instrumented assembly are shown on Figure 4.1.

Tests without fuel but with oil on the cylinder wall indicated that the coefficient of friction and ring leakage was high. Experimental piston positions and crankcase pressure traces shown on Graph 4.1 were compared with the computer calculated results shown on Graph 4.2. Computed results were obtained by using measured engine parameters whenever possible. Parameters not easily measured were determined by repeatedly comparing the computed result with the test results. Some of the parameter variations are shown on Graph 4.3. The procedure led to the following important observations:

1. The amount of leakage from the cylinder influenced the final stopped piston position. (An effective gap area of .0005 in$^2$ correlated the computed piston positions with the first test data. Since the calculated ring gap area was .0001 in$^2$ the amount of leakage over the rings was four times higher than the amount of leakage through the ring gaps. As the rings wore in, the effective gap area decreased).

2. The coefficient of friction influenced the number of cycles before the piston motion stopped. (A friction coefficient of .55 correlated the computed values with the test observation.)
Figure 4.1 Photographs of the instrumented FPS

Figure 4.2 Photograph of the FPS with a fixed-throw crankshaft
Note: Reference lines for position traces are in 1/4 in increments. Time = 20 millsec/div

Graph 4.1 Prototype engine test traces
Graph 4.2 Prototype engine traces (computed) without combustion
Graph 4.3 Effect of leakage, friction and damping on traces (computed)
Graph 4.4 Prototype engine position traces (computed) with combustion
3. The amount of leakage from the precompression chamber influenced the chamber pressure and the rate of decay. (The deduced leakage area was .006 in\(^2\) which was equivalent to a clearance of .001 in on all sliding surfaces.)

4. The amount of damping influenced the shape of the stroke-time curve. (The deduced damping coefficient was .05 sec\(^2\)-lb/in\(^2\)).

Originally only a few combustion cycles could be obtained from the test engine, as shown earlier on Graph 4.1, and then only when starting fluid was injected into the cylinder prior to a start. Nevertheless, important deductions were made by comparing the engine traces with the computed results, as shown on Graph 4.4. Some of the deductions are as follows:

1. When the ignited air-fuel ratio was approximately stoichiometric or when too much energy was put into the spring, the piston hit the top of the cylinder head, Graph 4.1 E.

2. When the cylinder mixture was rich, two combustion cycles occurred; when the mixture was very rich, the second cycle released more energy than the first, Graph 4.1 H.

3. When the crankcase contained a lean mixture and the cylinder contained a rich mixture, three combustion cycles occurred.
4. When the crankcase contained a rich mixture and the cylinder also contained a rich mixture, four or five combustion cycles occurred.

When it became obvious that in order to find the correct air-fuel ratio the engine had to be reciprocated continually, a three-throw crankshaft was designed and fabricated. The crankshaft allowed the engine to be driven with an electric motor at a fixed stroke so that the carburetor could be adjusted and the rings could be worn in. The mechanism is shown on Figure 4.2.

After the carburetor had been adjusted to produce a combustible mixture, combustion took place continually but not until the engine was cooled with an external blower did the engine become self-sustaining indefinitely. The use of the external blower to keep the fin temperature low came about because the computer results had shown that the uncooled engine would attain a fin temperature of 567°F if driven externally. The high fin temperature caused a large ring gap, a low charge density and early ignition. Above a fin temperature of 250°F the indicated horsepower was less than the friction horsepower so that the engine would not run by itself. The distribution of energy as the fin temperature changes is shown on Graph 4.5.

The computed charge flow was verified experimentally by measuring the air flows and estimating cylinder tempera-
tures. The data is given in Appendix VIII and the results are included on the graph.

After the carburetor had been adjusted to produce a combustible mixture, the special crankshaft was moved and replaced with an oscillating crankshaft. The new crankshaft synchronized the piston to the cylinder and later allowed
Figure 4.3 Photographs of FPS with an oscillating crankshaft (A) with a fixed-throw drive crankshaft (B) details of connecting rod (C) Prony brake free from crankshaft flywheel (D) Prony brake loaded (E) general view in free-piston configuration (F) manual starting with wrench
the spring to be pre-compressed manually. A side effect of the crankshaft and connecting rod masses was to reduce the operating frequency from 4,200 cpm to 2,500 cpm.

The new crankshaft was oscillated with the original crankshaft-motor mechanism or precompressed manually with a wrench (Figure 4.3). When the wrench was suddenly removed the spring was free to drive the piston through its compression strokes. Because the strokes were no longer controlled by a crankshaft, the engine oscillated on the free-piston principle as originally designed, although at a lower frequency because the crankshaft and connecting rods increased the oscillating masses.

A Prony brake with a leather friction surface was added to the frame so that the oscillating crankshaft could be loaded for power test, as shown on Figure 4.3. Although the blade in an actual saw would be loaded only during its outstroke, the Prony brake applied a load to the crankshaft flywheel (diameter = 2.5 in, crank throw = .75 in) during the instroke as well as during the outstroke.

Experimental position traces during load and no-load conditions are shown on Graph 4.6. Several typical piston positions at the end of the stroke and the computed traces are shown on Graph 4.7. The lower curve on the graph shows what effect air-fuel ratio variation has on the piston position.

The test engine traces of Graph 4.6 indicate that the engine started and ran when the equivalent of a 7.4 lb
Graph 4.6  Experimental engine test position traces
Graph 4.7  Experimental engine position traces (computed)
load (.05 bhp) was acting on the saw blade but stalled when a 13.7 lb load (.10 bhp) was acting. These results were verified by the computed results. What the computed results do not show is the sporadic type of combustion taking place during the slow speed oscillation. The computer program assumed ideal carburetion. The sporadic engine operation can be explained as follows:

1. The operating speed was just above idling so that, with the existing large-throat carburetor, carburetion was poor: air-fuel fluctuated and vaporization was incomplete. Typical power saw engines at this speed have similar problems (A 4.2 in\(^3\) engine produced only .1 bhp at 2,300 rpm, [2.47]).

2. A good combustion cycle released enough energy to force the piston against the cylinder head. The impact often absorbed enough energy to stop the engine.

3. The friction and damping coefficients were high.

4. The amount of leakage past the rings was very high; (the gap area at room temperature was .0001 in\(^2\) whereas the designed area was .00001 in\(^2\)).

5. Ignition did not start at the optimum piston position. The point at which ignition started depended on the cylinder head temperature, on the composition of the fuel (mainly on the amount of starting fluid
present and what fuel was used—engine ran on diesel fuel, white gas, regular gas and glow-plug engine fuel), and on the amount of energy supplied to the glow-plug. Consequently the indicated horsepower varied from cycle to cycle and from test to test.

Some of the aforementioned drawbacks to good engine performance will also be present in the redesigned engine. It will be difficult to know what mixture the carburetor is producing unless the engine is already running. Unlike the conventional engine where quite a few revolutions occur when the engine is started, the present free-piston engine has a maximum of seven charging cycles per starting attempt (from an initial stroke of 1.9 in). For example, with an initial air/fuel ratio of 100:1 and with the carburetor correctly adjusted, at least two starting attempts will be required before the mixture in the cylinder is ignitable. But because the free-piston engine always runs at full speed and the bleed port charge recirculates through the carburetor, the air velocity in the carburetor throat will always be high. Consequently mixing and vaporization in a properly matched carburetor will be good.

Initial ignition of a combustible charge should present no problem because the initial compression ratio is very high. For example, if the engine is started from an initial stroke of 1.90 in, the first compression ratio will be 55:1 and the gas temperature will be 1,800°F. If the air
temperature is 40° below zero instead of 60°F, the compression temperature will be 1200°F.

Another drawback of the prototype saw will be the vibration caused by the unidirectional cutting action. A 3 lb unit oscillating at 6,400 cpm will have a peak-to-peak hand-held vibration amplitude of .04 in at full load (57 lbs). When held against a bucking spike the amplitude will be considerably less. If two counter-oscillating saw blades are used, the amplitude can be reduced to a negligible amount.

The lack of overload protection will be a deterrent to running the engine at maximum power. The drawback will be less critical if the engine has a flat power-versus-stroke curve near maximum power, or if the frame or blade is designed to transmit no more than the maximum permissible load. If the blade hits a solid object and the load applied through the handle is less than the permissible load, the frame will accelerate and store more energy. The acceleration will prevent the engine from stalling unless the friction on the saw blade is excessive. More development work will be required before these drawbacks can be fully evaluated, although a number of conclusions can be drawn about the engine concept. These conclusions are given in the next section.
4.2 Conclusions

The performance data of the test engine, the predicted values of the prototype, and the predicted values of a redesigned engine are listed in Table VIII. The increased

Table VIII
Performance Characteristics of FPS

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<th>Data Test Engine</th>
<th>Computed Values</th>
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<td>Redesigned</td>
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<td>Noise levels</td>
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<td>Starting</td>
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</table>
power of the redesigned engine is due to higher operating speeds (smaller piston mass), lower cylinder temperatures, and a larger area for the intake port.

The simplicity of the free-piston power saw is shown by the photographs on Figure 4.4. When compared with a conventional reciprocating blade power saw, the fewer components of the FPS (no ignition system, starter assembly, fan, throttle and reed valves) show the advantage of the free piston principle in a power saw.

The new saw will be safer to operate because of the following characteristics:

1. No time-lag occurs when starting and stopping the saw. Once the trigger is released, the blade will be motionless in less than .01 sec. A motionless blade is obviously safer than a moving blade and much safer than a moving chain with treacherous teeth on the top and bottom of the blade. The instant, effortless starting feature enables the operator to start the saw after he has maneuvered it into an awkward position and so frees him from the necessity of handling a dangerous weapon in a hazardous situation.

2. Throttling is automatic. This means that the operator will not be concerned about the engine speed during part-load operations.
conventional blade saw

free-piston power saw

**SUB ASSEMBLIES NOT REQUIRED**

reed valve
starter
throttle
magneto
spark plug
fan

(crackshaft and connecting rods not shown)

**REQUIRED**

cover
carburetor
blade, piston and cylinder assembly

fuel tank and pickup
muffler

Figure 4.4 Photographs of conventional power saw and the FPS
3. If the engine stalls suddenly during an overload, some accidents will be prevented or reduced.

Although the exhaust noise levels from the free-piston engine will be about the same as from a conventional engine with a similar muffler, the noise caused by structural vibrations will be lower for the following reasons:

1. Since the engine is balanced, the vibration amplitude of the frame will be lower.
2. Since no crankshaft is used, the amount of piston slap and bearing noise is reduced.
3. Since the cylinder is short and stiff and the transmitted forces are mainly longitudinal, little lateral vibration will occur.
4. Since no fan is required, aerodynamic noise is reduced.
5. Since a blade and not a chain is used, the chain and sprocket noises are eliminated.

Although the prototype engine is self-sustaining and has produced a positive output, more development work is required before it can produce the specified power. The following design steps could be followed:

1. Redesign piston head to reduce the clearance volume at the top of the stroke. This reduction can be achieved by using rings with smaller gaps and by
reducing the volume under the rings, between the piston headland and cylinder liner, and in the glow plug.

2. Design an ignition system for a proximity-type spark plug.

3. Redesign the synchronizing and arresting mechanism to simplify the design, to reduce the number of components, and to permit the use of a light aluminum piston.

4. Design a fuel metering system for a small propane tank and so reduce the specific fuel consumption and the amount of pollutants in the exhaust gases.

5. Redesign engine to incorporate spark ignition, to incorporate a gaseous fuel metering system, to reduce the frame weight, and to allow for economical manufacture.

4.3 Summary

The design envelope for the power saw was based on experiments, published information, questionnaires, and practical experience. Experiments were performed on existing saws to determine optimum cutting speeds and typical vibration and noise levels. These levels were compared with allowable noise and vibration levels published in medical and engineering papers. The specifications for a spark
arrestor design was based on government regulations. The optimum size and desired operating characteristics were based on a questionnaire distributed to power saw users and on personal experience gained while employed as a research engineer with a chain saw company and as a chain saw operator with a logging firm.

The cutting speed tests produced data on the effect of chain pitch, wood type, sprocket size, "bite" depth, bar length, gear reduction and speed variation on the specific energy required. Calculations based on the experimental results showed that for hemlock the minimum specific energy required was $1,600 \text{ in-lb/in}^2$, and for maple the energy required was $3,700 \text{ in-lb/in}^2$.

Vibration level tests showed that the vibration amplitudes of existing saws varied from .013 to .030 in, so that for continuous saw operation the vibration could cause damage to the vasculature of the hands. The noise level tests also showed that for long term exposure the noise could cause hearing impairment (typical levels were about 105 dB). The second design stage consisted of an evaluation of available energy sources and an analysis of wood cutting devices. It was concluded that of the engines considered, the internal combustion engine burning hydrocarbon fuels had one of the lowest weight/power ratios, and that shears had one of the lowest specific power requirements. Although quite efficient, shears are nevertheless heavy so that the
conventional saw blade was chosen as the cutting device.

The synthesis of new engine arrangements started with an analysis of problem areas in existing engines and produced three new engine configurations, one of which was to produce rotary motion and the other two were to produce reciprocating motion. The rotary configuration consisted of a piston bouncing in a rotating cylinder so that the weight of the piston caused the cylinder to turn; this engine was not dynamically stable. The reciprocating configurations consisted of a piston bouncing in an oscillating cylinder; these engines were stable if the piston and cylinder were synchronized. It was necessary to represent the dynamics of all three configurations by two coupled non-linear differential equations before their stability could be checked with the aid of a computer.

In the configuration finally chosen, the piston oscillated between gases in one end of the cylinder and a mechanical spring in the other. Because the cylinder was also free to oscillate, the engine was balanced and the cylinder was self-cooling. To determine the amount of self-cooling and the size of fins required to prevent the cylinder from overheating, it was necessary to obtain heat transfer coefficients experimentally. The data indicated that the heat transfer from a cylinder head increased by a factor of 6 over free convection when a cylinder was reciprocated with a stroke of 1.375 in at a speed of 3,000 cpm ($Nu = 10.8 + .082 (Re \sqrt{\frac{L}{S}})^{.985}$).
The information for the piston size and port dimensions was obtained from an analysis of existing engine performances and dimensions. The available data was assembled, correlated in terms of useful dimensionless parameters, and used in conjunction with derived scaling factors to provide the necessary dimensions for the port sizes. The information for the scavenging and bleed ports (for automatic throttling) was obtained from an analysis of the dynamic and thermo-dynamic behaviour of the proposed engine in conjunction with a computer program. The design of the machine components followed conventional techniques and delineated a prototype that was evaluated theoretically as well as experimentally.

The first attempts at starting the engine, after the parts had been made and assembled, produced only a maximum of 5 successive combustion cycles. When the engine was driven with a fixed-throw crankshaft it was possible to break-in the engine and adjust the air-fuel ratio so that it would become self-sustaining. Following this adjustment it was a simple matter to disconnect the connecting rod and run the engine in the free-piston configuration.

The prototype of the free piston power saw incorporated the following novel features:

1. The engine can be accurately balanced. When correctly synchronized, the acceleration of the piston is balanced by the acceleration of the cylinder.
2. The cylinder head is self-cooling. The reciprocating motion of the cylinder moves the cooling air across the fins.

3. The piston stroke is controlled by gas and spring forces and depends on an energy balance between the amount released and the amount taken out during the cycle.

4. Throttling is automatic. Because the stroke changes when the load is changed, the length of time the transfer and bleed ports are open depends on the load.

5. The compression ratio increases when throttling increases. As a result, more energy is available for ignition when the fresh charge is more diluted with residual gases.

6. The carbureted air-fuel mixture is ignited with the energy of compression.

7. The spring is automatically locked in a compressed state when the engine is stopped. This feature results in instant, effortless stop-start characteristics.

8. The reciprocating motion of the piston is used directly since the blade of the saw is part of the piston.

Once the experimental test data and computed results were correlated, it was possible to predict engine
performance and to isolate and evaluate problem areas. At this stage the project objectives had been achieved: a new principle had been conceived, components had been delineated and a working prototype had been evaluated.
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1.51 Power Saw Manufacturers Association, Recommended Practice for Spark Arresters Used on Multiposition Engines (proposed draft), Chicago, May 5, 1967.

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### APPENDIX I

**CUTTING SPEED TESTS WITH CONVENTIONAL POWER SAW**

performed in U.B.C. Endowment Lands Forest

<table>
<thead>
<tr>
<th>Date &amp; Type of Wood</th>
<th>Saw 3/4 &amp; Spec.</th>
<th>Speed (rpm)</th>
<th>Cutting Log Teeth</th>
<th>Diameter (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>24/4/67 hemlock</td>
<td>7 teeth</td>
<td>6000</td>
<td>8</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>14 x 16</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(rot 4/67)</td>
<td></td>
</tr>
<tr>
<td>24/4/67 maple</td>
<td>8 teeth (fresh)</td>
<td>4500</td>
<td>8</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>14 x 16</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(rot 4/67)</td>
<td></td>
</tr>
<tr>
<td>24/4/67 maple</td>
<td>7 teeth (fresh)</td>
<td>5000</td>
<td>8</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>14 x 16</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(rot 4/67)</td>
<td></td>
</tr>
<tr>
<td>24/4/67 hemlock</td>
<td>7 teeth (semi-dry)</td>
<td>6000</td>
<td>8</td>
<td>14 x 16</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(rot 4/67)</td>
<td></td>
</tr>
<tr>
<td>1/5/67 hemlock</td>
<td>7 teeth</td>
<td>6000</td>
<td>8</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>14 x 16</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(rot 4/67)</td>
<td></td>
</tr>
<tr>
<td>1/5/67 cedar</td>
<td>8 teeth (small rot)</td>
<td>6000</td>
<td>8</td>
<td>14 x 16</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(4 1/2 dia., semi-dry)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(rot 4/67)</td>
<td></td>
</tr>
</tbody>
</table>

**Power check 3/8 1770120**

<table>
<thead>
<tr>
<th>Date &amp; Type of Power</th>
<th>Saw 3/8 &amp; Spec.</th>
<th>Speed (rpm)</th>
<th>Cutting Log</th>
<th>Diameter (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>11/4/67</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>10/5/67</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>scale</td>
<td>bhp</td>
<td>bhp</td>
</tr>
<tr>
<td>4500</td>
<td>6.9</td>
<td>3.06</td>
</tr>
<tr>
<td>5000</td>
<td>6.5</td>
<td>3.45</td>
</tr>
<tr>
<td>5500</td>
<td>6.1</td>
<td>3.85</td>
</tr>
<tr>
<td>6000</td>
<td>5.8</td>
<td>3.36</td>
</tr>
<tr>
<td>6500</td>
<td>5.6</td>
<td>3.57</td>
</tr>
<tr>
<td>7000</td>
<td>3.6</td>
<td>3.77</td>
</tr>
<tr>
<td>8000</td>
<td>2.8</td>
<td>3.30</td>
</tr>
<tr>
<td>9000</td>
<td>2.5</td>
<td>3.55</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>scale</td>
<td>bhp</td>
<td>bhp</td>
</tr>
<tr>
<td>6000</td>
<td>7.8</td>
<td>4.68</td>
</tr>
<tr>
<td>7000</td>
<td>6.9</td>
<td>4.49</td>
</tr>
<tr>
<td>MACHINE &amp; S/N</td>
<td>DETAILS TEST DATE</td>
<td>SPEED</td>
</tr>
<tr>
<td>-------------------</td>
<td>-------------------</td>
<td>-------</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A 28100008</td>
<td>no bar</td>
<td>2400</td>
</tr>
<tr>
<td>(248 gm c'wt)</td>
<td>29/5/68</td>
<td>6000</td>
</tr>
<tr>
<td>76% balanced</td>
<td>7000</td>
<td>7000</td>
</tr>
<tr>
<td></td>
<td>36&quot; bar, cutting</td>
<td>7000</td>
</tr>
<tr>
<td></td>
<td>30/5/68</td>
<td>7000</td>
</tr>
<tr>
<td>H 28100005</td>
<td>no bar</td>
<td>2400</td>
</tr>
<tr>
<td>(196 gm c'wt)</td>
<td>29/5/68</td>
<td>6000</td>
</tr>
<tr>
<td>53% balanced</td>
<td>6000</td>
<td>7200</td>
</tr>
<tr>
<td></td>
<td>36&quot; bar, cut, 29/5</td>
<td>7000</td>
</tr>
<tr>
<td></td>
<td>36&quot; bar, cutting</td>
<td>7000</td>
</tr>
<tr>
<td></td>
<td>30/5/68</td>
<td>7000</td>
</tr>
<tr>
<td></td>
<td></td>
<td>7000</td>
</tr>
<tr>
<td>G 28100007</td>
<td>no bar</td>
<td>2400</td>
</tr>
<tr>
<td>(207 gm c'wt)</td>
<td>29/5/68</td>
<td>6000</td>
</tr>
<tr>
<td>58% balanced</td>
<td>7500</td>
<td>7500</td>
</tr>
<tr>
<td></td>
<td>36&quot; bar, cutting</td>
<td>7000</td>
</tr>
<tr>
<td></td>
<td>30/5/68</td>
<td>7000</td>
</tr>
<tr>
<td>F 28100006</td>
<td>no bar</td>
<td>2400</td>
</tr>
<tr>
<td>(222 gm c'wt)</td>
<td>29/5/68</td>
<td>6000</td>
</tr>
<tr>
<td>63% balanced</td>
<td>7000</td>
<td>7000</td>
</tr>
<tr>
<td></td>
<td>36&quot; bar, cutting</td>
<td>7000</td>
</tr>
<tr>
<td></td>
<td>30/5/68</td>
<td>7000</td>
</tr>
<tr>
<td>E /rubber 1128940</td>
<td>no bar, 30/5/68</td>
<td>7000</td>
</tr>
<tr>
<td></td>
<td>36&quot; bar, cut, 30/5</td>
<td>7000</td>
</tr>
<tr>
<td>C 34001031</td>
<td>15&quot; bar, cutting</td>
<td>7000</td>
</tr>
<tr>
<td>31/5/68</td>
<td>15&quot; bar, not cutting</td>
<td>7000</td>
</tr>
<tr>
<td>(23% balanced)</td>
<td>15&quot; bar, cutting</td>
<td>7000</td>
</tr>
<tr>
<td></td>
<td>15&quot; bar, not cutting</td>
<td>7000</td>
</tr>
<tr>
<td>B 18500002</td>
<td>15&quot; bar, cutting</td>
<td>7000</td>
</tr>
<tr>
<td>(20% balanced)</td>
<td>31/5/68</td>
<td>7000</td>
</tr>
<tr>
<td>D 34007139</td>
<td>18&quot; bar, cutting</td>
<td>7000</td>
</tr>
<tr>
<td>(27% balanced)</td>
<td>31/5/68</td>
<td>7000</td>
</tr>
</tbody>
</table>
For typical power saws with wide open throttle taken on May 17, 1967, at a distance of 2.5 feet from microphone to sprocket centerline using a Bruel & Kjaer Sound Meter Type 2203 with octave filter type 1613 and microphone type 4131. Readings taken on flat grassy terrain in New Orleans, La.

<table>
<thead>
<tr>
<th>Saw S/N and Speed</th>
<th>dBA</th>
<th>dB</th>
<th>OCTAVE BAND NOISE LEVEL IN DECIBELS</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>Linear</td>
<td>CPS (Hz)</td>
<td>K CPS (kHz)</td>
</tr>
<tr>
<td></td>
<td>63</td>
<td>125</td>
<td>250</td>
</tr>
<tr>
<td></td>
<td>45</td>
<td>89</td>
<td>178</td>
</tr>
<tr>
<td></td>
<td>89</td>
<td>178</td>
<td>355</td>
</tr>
<tr>
<td>1760103 at 7000 rpm</td>
<td>109</td>
<td>110</td>
<td>82</td>
</tr>
<tr>
<td>A</td>
<td>109</td>
<td>110</td>
<td>73</td>
</tr>
<tr>
<td>36001171 at cutting rpm</td>
<td>110</td>
<td>111</td>
<td>85</td>
</tr>
<tr>
<td>B</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3100-2-1948 at cutting rpm</td>
<td>104</td>
<td>105</td>
<td>75</td>
</tr>
<tr>
<td>C</td>
<td>102</td>
<td>102</td>
<td>78</td>
</tr>
<tr>
<td>1760103 at 7000 rpm</td>
<td>108</td>
<td>110</td>
<td>80</td>
</tr>
<tr>
<td>on electric dynamometer</td>
<td>110</td>
<td>110</td>
<td>80</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Operator absent
### APPENDIX IV
### RESULTS OF QUESTIONNAIRE DISTRIBUTED TO
### CHAIN SAW USERS IN 1967

<table>
<thead>
<tr>
<th>Extent machine characteristics bother operator (code: V—very much, Q—quite a bit, S—somewhat, N—not at all)</th>
<th>Replies Received</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. vibration</td>
<td>Professional Loggers</td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td>2. smell</td>
<td></td>
</tr>
<tr>
<td>3. noise</td>
<td></td>
</tr>
<tr>
<td>4. weight</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Importance of specific items (code: E—extremely important, Q—quite important, S—only slightly important, N—unimportant)</th>
<th>Replies Received</th>
</tr>
</thead>
<tbody>
<tr>
<td>5. easy starting</td>
<td>E Q E E E Q E E E E</td>
</tr>
<tr>
<td>6. reliability</td>
<td>E E E E Q E E E E</td>
</tr>
<tr>
<td>7. easy maintenance</td>
<td>N Q Q N E Q Q Q Q E</td>
</tr>
<tr>
<td>8. low fuel consumption</td>
<td>Q Q S S E E Q S S</td>
</tr>
<tr>
<td>9. neat appearance</td>
<td>N S S E S S S S N</td>
</tr>
<tr>
<td>10. low upkeep cost</td>
<td>N Q N E Q E Q Q</td>
</tr>
<tr>
<td>11. low oil consumption</td>
<td>Q S S E Q Q S Q</td>
</tr>
<tr>
<td>12. low first cost price</td>
<td>Q E S E E Q Q</td>
</tr>
<tr>
<td>13. low weight and small size</td>
<td>Q Q E E E E E E S Q E</td>
</tr>
<tr>
<td>14. long, troublefree life</td>
<td>Q Q E E E E Q Q Q E E</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Area saw is used</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Average tree size cut (inch diameter)</td>
<td>7 8 6 6 8 7 40 18 18</td>
</tr>
<tr>
<td>Maximum tree size cut (inch diameter)</td>
<td>16 15 15 12 26 24 120 36 36</td>
</tr>
<tr>
<td>Saw used in snow</td>
<td>y y y y y y y y y no</td>
</tr>
<tr>
<td>Saw used in rain</td>
<td>y y y y y y y y y y y</td>
</tr>
<tr>
<td>Saw used where temperatures drop below 0°F</td>
<td>y y y y y y y y y y</td>
</tr>
<tr>
<td>20°F below</td>
<td>y y y y y y y y y y</td>
</tr>
<tr>
<td>40°F below</td>
<td>y y y y y y y no</td>
</tr>
<tr>
<td>Saw used where temperature goes above 110°F</td>
<td>y y y y y y no no no</td>
</tr>
<tr>
<td>Number of starts per day</td>
<td>20 20 20 20 50 50 25 6 15</td>
</tr>
<tr>
<td>Amount of wood cut—cords per day</td>
<td>14 50 14 12 12 6 3 30th 1</td>
</tr>
<tr>
<td>cords per year</td>
<td>2800 2800 3000 1188 6M 2 2</td>
</tr>
<tr>
<td>cords per saw</td>
<td>4000 200 2000 1000 10M 6</td>
</tr>
<tr>
<td>Desired running time on tankful of gas (minutes)</td>
<td>60 60 60 50 120 90 90 40 45</td>
</tr>
</tbody>
</table>
### APPENDIX V

**TYPICAL POWER SAW PERFORMANCE DATA**

<table>
<thead>
<tr>
<th>SAW SYMBOL</th>
<th>V D</th>
<th>S/b</th>
<th>MAX BMEP</th>
<th>bhp In 2 &amp; rpm</th>
<th>EXHAUST</th>
<th>TRANSFER</th>
<th>INTAKE</th>
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*Note: V indicates the rated speed of the engine in revolutions per minute (rpm).*

*Reed valve* indicates the type of valve used in the engine.
### APPENDIX VI

#### HEAT TRANSFERS FROM RECIPROCATING CYLINDER HEADS

<table>
<thead>
<tr>
<th>Description of cylinder head</th>
<th>Test No.</th>
<th>Time since change (min.)</th>
<th>Amplitude pr-to-pr (in.)</th>
<th>Frequency (cycles per min)</th>
<th>Heat power (watts)</th>
<th>Plug Temp. (°F)</th>
<th>Fin Temp. (°F)</th>
<th>Room Temp. (°F)</th>
<th>Head Temp. (°F)</th>
<th>Block Temp. (°F)</th>
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<td>13/11/67</td>
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**NOTES:**
- Clearance at TDC between spacer & head = .05".
- Clearance at TDC between head and block = .002.
- Screw holes on spacer open.
- Test 73 - oil added
- Test 75 - oil added
- Test 76 - screw holes plugged.
- Test 79 - buffer grams.

<table>
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<tr>
<th>Test No.</th>
<th>Time since change (min.)</th>
<th>Diameter</th>
<th>Length</th>
<th>Width</th>
<th>Thickness</th>
<th>Pitch</th>
<th>Rotation</th>
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<td>4</td>
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<td>50</td>
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<td>106</td>
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<td>0.02</td>
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<th>Width</th>
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<td>106</td>
<td>0.26</td>
<td>0.02</td>
<td>70</td>
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**NOTES:**
- Clearance at TDC between spacer & head = .05".
- Clearance at TDC between head and block = .002.
- Screw holes on spacer open.
- Test 73 - oil added
- Test 75 - oil added
- Test 76 - screw holes plugged.
- Test 79 - buffer grams.
APPENDIX VII

FPS PROTOTYPE TEST DATA

A. Test Data for engine with fixed throw crankshaft-driven externally

<table>
<thead>
<tr>
<th>Date of Test</th>
<th>No.</th>
<th>Speed (rpm)</th>
<th>Pres drop (in H₂O)</th>
<th>Fin temp (°F)</th>
<th>Combustion</th>
<th>Fuel Flow</th>
<th>Scavenging ratio</th>
<th>Air/Fuel</th>
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<td>3200</td>
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<td>1</td>
<td>2750</td>
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<td>2</td>
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<td>2.8cc/.56min</td>
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</tr>
<tr>
<td>10/1/70</td>
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<td>3600</td>
<td>with external cooling supplied by blower, fin temp remained below 200°F and engine ran continuously</td>
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*estimated temperature of piston

B. Test Data for free-piston test engine.

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<th>Date</th>
<th>No.</th>
<th>Speed (rpm)</th>
<th>Length (cycles)</th>
<th>Loading (out &amp; in)</th>
<th>Stroke</th>
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<td>19/1/70</td>
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<td>2500</td>
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<td>reg. gas, oil</td>
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<td>2480</td>
<td>20</td>
<td>1/2 @ 19&quot;</td>
<td>1.15</td>
<td>reg. gas, oil, ether</td>
</tr>
<tr>
<td>5/2/70</td>
<td>4</td>
<td>2460</td>
<td>6</td>
<td>1/2 @ 19&quot;</td>
<td>1.05</td>
<td>reg. gas, oil, ether</td>
</tr>
</tbody>
</table>

\[
\text{Load} = \left( \frac{\text{Load moment} + \text{lever arm moment}}{1.5} \right) \left( \frac{2.5}{1.5} \right)^{\mu}
\]

where lever arm moment = .26 x 9.7

\[
\mu = \text{coefficient of friction, leather on C.I. = .56}
\]