DEVELOPMENT OF A STEAM POWERED

SPORTS CAR

by

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TECHNOLOGY

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Dear Mr. Hamilton:

This thesis, entitled, "Development of a Steam Powered Sports Car", is hereby submitted in partial fulfillment of the requirements for the degree of Bachelor of Science in Mechanical Engineering.

Respectfully submitted,

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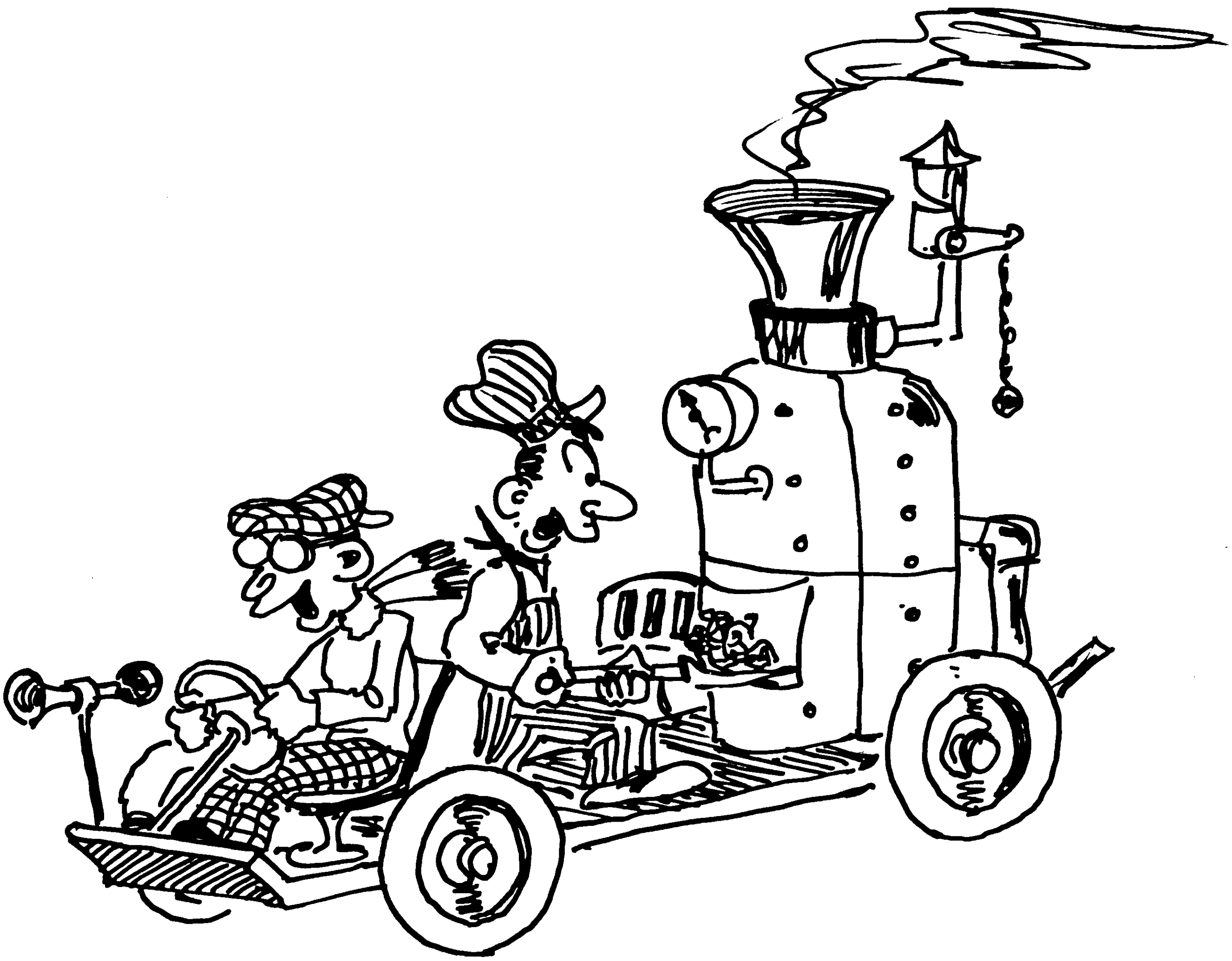

# FOREWORD

The recent appearance of a quite modern steam car built by Charles F. Keen of Madison, Wisconsin, gave us the idea of attempting to build a steam sports car. Although this particular application may be novel, the idea of a steam powered car is not new. Extensive work was done on it at the turn of the century and many working models were sold. These steam cars competed with internal combustion engine designs for a few decades, but suddenly disappeared from the scene. By 1930 the market for steam cars had vanished.

Since that time little work has been done on improving older models or building new ones. As a consequence the internal combustion engine car now has tremendous advantages in the field of automobile design. A great wealth of technological knowledge has been poured into the internal combustion engine, thereby putting it in the position where it is now used almost exclusively as the prime mover in automotive applications.

Recently, consumer demand for small high performance cars has increased significantly. These sports cars are unique in that they can double both as a personal automobile and as competitive machines. They must give their owners above average performance in normal daily driving, while they should also be able to show up well on the racing circuit. A sports car is expected to have high acceleration and top speed, low fuel consumption, and a good looking, sturdy body. The consumer in this field is willing to pay well for a design which shows up nicely at home, and at the races.

Such a design may be developed around a steam engine. A two-cylinder double-acting steam engine with less than forty moving parts provides as many power impulses per crankshaft revolution as an eightcylinder internal combustion engine with its increasingly large number of complex accessories; the steam engine requires neither clutch nor torque converter of any kind! Geared directly to the axle of the car the engine can be conveniently mounted between the frame giving the car a desirable low center of gravity, a necessity for stability when cornering. In addition, this combination features the availability of large torques at all speeds to insure fast acceleration, which, indeed, is important for racing the car.

However, many problems are met in steam car design. A good portion of these are mentioned in the introduction along with a brief description of the basic features of steam cars. Perhaps the section of most interest to the reader is the one which compares the steam car to a conventional sports car. A short historical sketch also shows the advances made in steam car design to date.

Encouraged by the favorable characteristics of steam engine propulsion for sports car application and Mr. Keen's success with a larger unit, the primary purpose of this paper is to design or set up specifications for the major components of a steam sports car. Prospects for mass producing this car may not be unrealistic in light of the sports car market which would welcome such a vehicle.

Therefore, the rest of this report is concerned with the design of our particular sports car. The components of this car have been taken from various sources, and a number of new features have been incorporated. A layout drawing showing the location of all the parts is also included.

## ACKNOWLEDGMENTS

Many people have been instrumental in assisting us with our particular design. We wish to thank G. L. Lindsay and H. E. Goetz of the Skinner Engine Company for information pertaining to uniflow steam engines, L. C. Hoagland at M.I.T. for explaining the mechanics of compact heat exchangers, Professor C. F. Taylor of M.I.T. for his poignant remarks which directed our energies along the proper channels, Professor J. Holt of M.I.T. for his help with the boiler and engine problems, R. B. Purdy of the Socony Mobil Oil Company for information concerning our lubrication problems, Professor M. A. Santalo of M.I.T. for his patience in assisting us with some of the basic theoretical problems, Brian O'Kane for his ideas on automotive body design.

In particular we wish to express our gratitude to Professor George Arthur Brown of the Mechanical Engineering Department at M.I.T. for his guidance as our thesis advisor on this project. He has taken the time and effort to keep up with our work throughout all the stages of its development. In addition, his recommendations were used as a basis for the studies we carried out over the past year.

## SUMMARY

The problems involved in building a steam powered sports car are investigated and presented in conjunction with their design applications. The engine and condenser are treated in detail, while other components are placed in their proper perspective.

One particular sports car is offered as a modern design solution. Powered by a 100 hp two-cylinder compound double-acting uniflow steam engine, speeds well over 100 mph should be attainable. Entrance steam is delivered by a flash boiler at 1100 psia and 660°F, this temperature being limited by lubrication difficulties in the engine. A small turbine to power auxiliaries further expands the steam. The latter exhausts to a plate-fin type compact heat exchanger which has been adopted for use as a condenser. It is capable of handling a minimum of 2,000 pounds of steam an hour, requiring only 8 square feet of frontal area.

Other features of this car include the turbine powered auxiliary equipment (pumps, fans, generator) and a regenerative feedwater heating cycle for overload operations. An original body design and component layout drawing is shown in Figure 13 while Figure 14 demonstrates the basic steam cycle employed.

Future work on this car should be directed at testing an actual engine and devising a means for improving the lubrication situation. Not only is it desired to lubricate at higher temperatures, but a system of separators must be designed to separate the oil from the exhaust steam before it reaches the boiler. Though many types of separators are available, their performance for this application must be determined.

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# INTRODUCTION

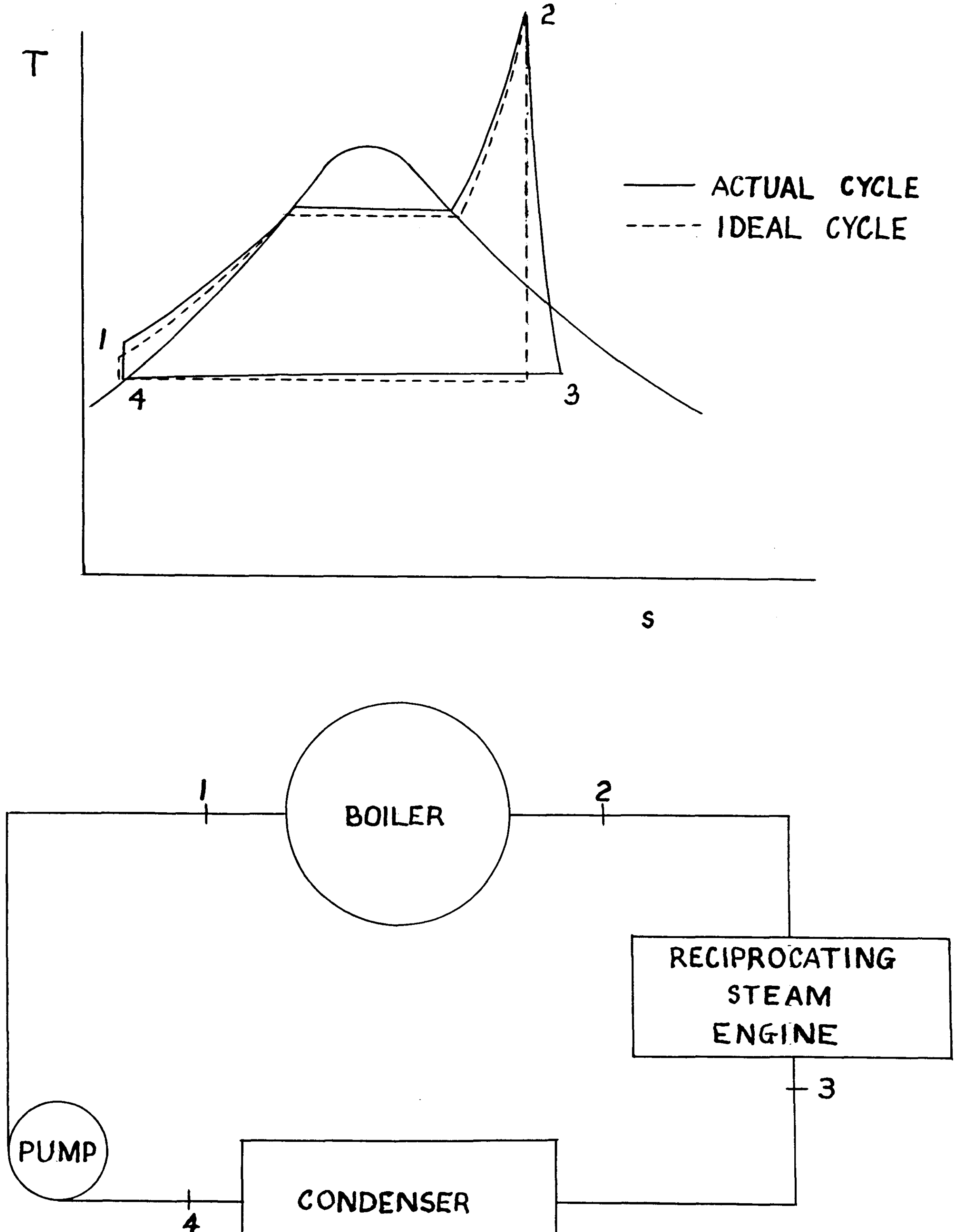
## A. PURPOSE OF INVESTIGATION

This study of a steam power plant for use in a sports car is undertaken with two distinct goals in mind. The first aim is to study the features of steam cars in general, while the second is to design one particular model which can compete with conventional internal combustion engine sports cars.

Fortunately, this investigation has a comparative model from which to work. The internal combustion engine has been used in automobiles for over half a century, providing a good standard of performance from which to judge any design. A comparison of the relative merits of steam power versus the internal combustion is presented later in this paper.

## B. BASIC STEAM CYCLE

Essential to any analysis of a steam power plant is a working knowledge of the basic thermodynamic steam cycle. This cycle is best illustrated with the aid of a temperature-entropy (T-S) diagram. On such a diagram (See Figure 1) steam is in its liquid phase at (1), which is the entrance to the boiler. The liquid is heated at essentially constant pressure to a superheated vapor state at (2). This steam is expanded through a reciprocating steam engine to some lower temperature and pressure at (4) where it is exhausted to a condenser. The steam is condensed (essentially at constant pressure and temperature) to the saturated liquid state where




# FIGURE 1
BASIC STEAM CYCLE

it is then raised in pressure with a negligible temperature change by the pump which restores the initial state (1).

The idealized cycle is shown by the dash lines on the T-S diagram. It consists of two constant pressure and two isentropic processes. This ideal cycle acts as a useful base for comparing actual steam cycles.

The efficiency of a steam cycle is defined as the ratio of the work delivered to the heat received. For a reversible or ideal cycle, this efficiency depends upon the temperature difference as well as the temperature maxima and minima. If the engine exhaust temperature is maintained constant, it is seen that improvement in efficiency may be achieved by increasing the average temperature at which heat is received. Similarly, for constant inlet temperatures, lowering engine exhaust temperatures will improve cycle efficiencies.

From this fact a few general conclusions can be drawn. In order to increase the average temperature at which heat is received, the following approaches may be used.

1. Increase the temperature of vaporization.
2. Superheat
3. Reheat

The average temperature at which heat is received may be most readily increased by increasing the temperature, and, of course, pressure of vaporization. For a maximum superheat temperature, such an increase causes a marked rise in the moisture content of the steam at the end of the expansion, as shown in Figure 2. This effect is sometimes more important than the immediate gain in efficiency. Higher superheat also increases the average temperature at which heat is received. Along with this improvement in cycle efficiency, Figure 3 shows the consequent reduction in the amount of moisture in the vapor exhaust. The degree of superheat is limited by metallurgical considerations.

The reheat cycle depicted in Figure 4 demonstrates a means of raising the temperature of vaporization and efficiency without exceeding a possible limiting value in moisture content. Steam can be withdrawn after partial expansion, superheated, and re-introduced into a second cylinder. Thus, the vapor quality after complete expansion can be controlled satisfactorily. However, the reheat cycle is costly because of its added complexity. Since the gain in work area obtained by this method can usually be achieved by other means, this method has not been thought useful for automobile applications.

An alternate means for increasing cycle efficiency consists of decreasing the flow of water which the working fluid receives at the lowest temperatures. This process involves bleeding steam between cylinders of a compound engine and using it for feed heating. Such a regenerative process appears more complex than advisable, although it is worthy of consideration.

The other temperature level of interest is that associated with the condenser. It is advisable to maintain as low a level as possible from the standpoint of cycle efficiency. Since many factors enter into the determination of this level, discussion of it will be deferred to the section of this report concerned with cycle components.

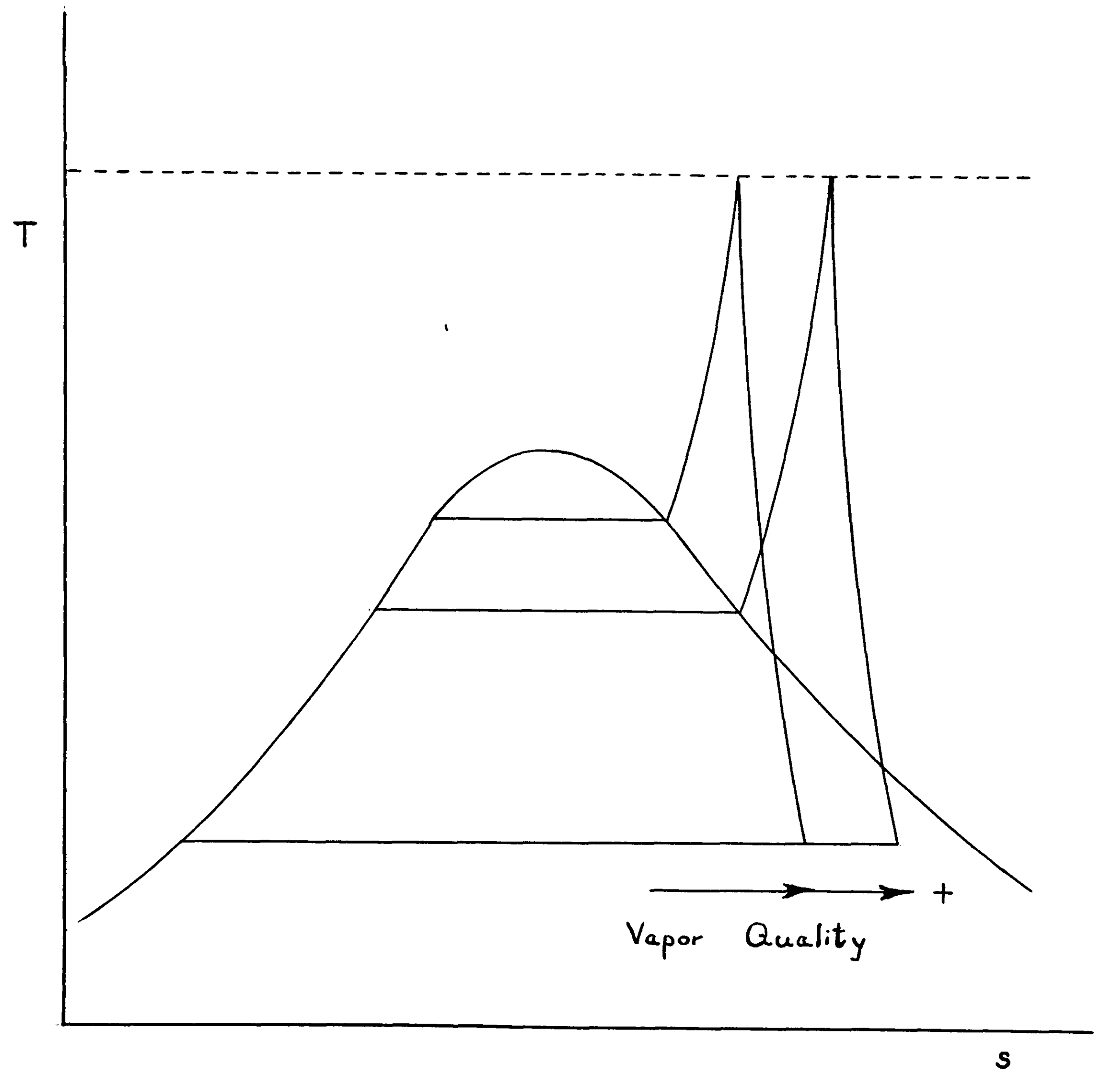



FIGURE 2
EFFECT OF TEMPERATURE OF VAPORIZATION ON FINAL MOISTURE

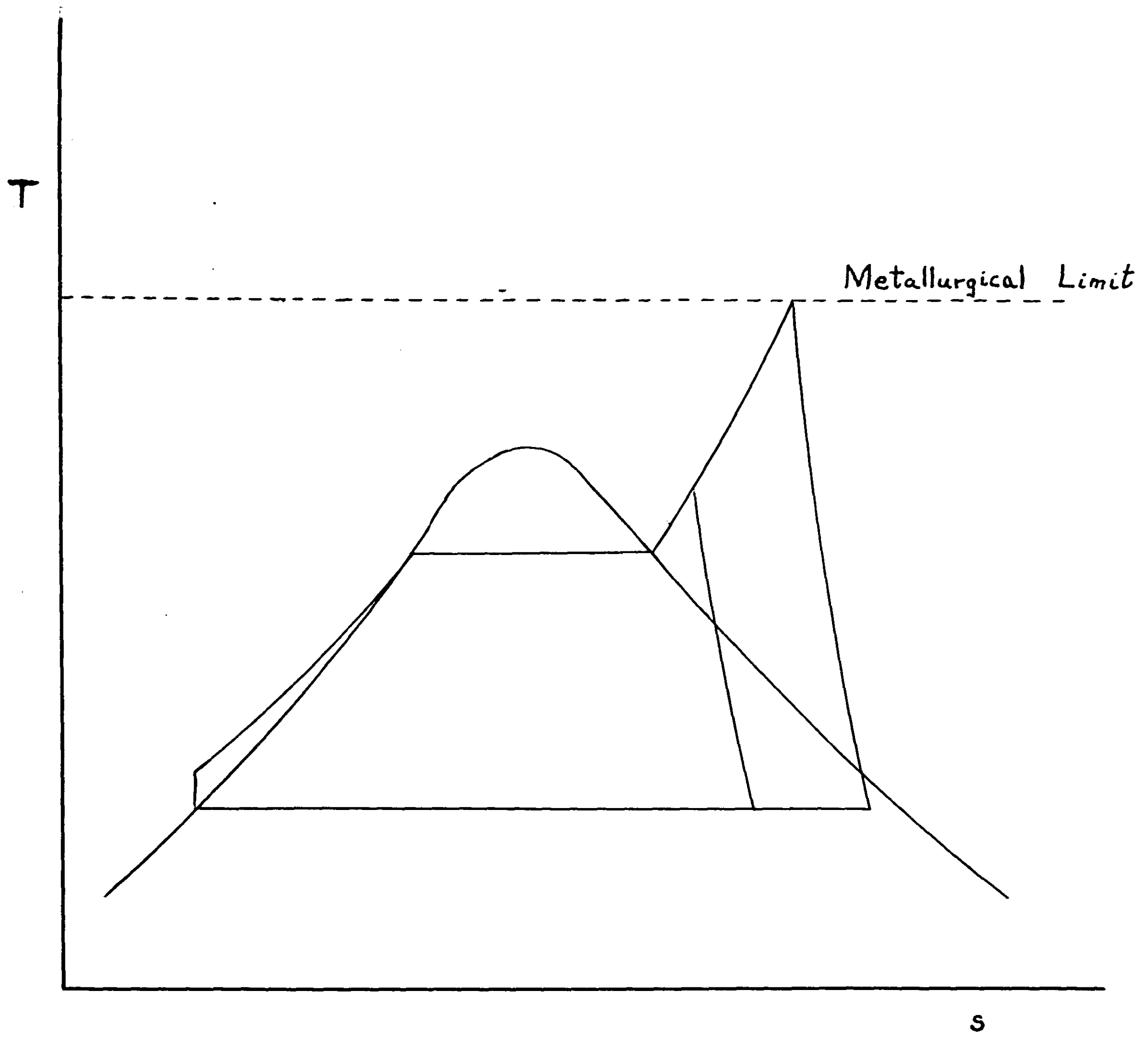



FIGURE 3

EFFECT OF SUPERHEAT ON FINAL MOISTURE

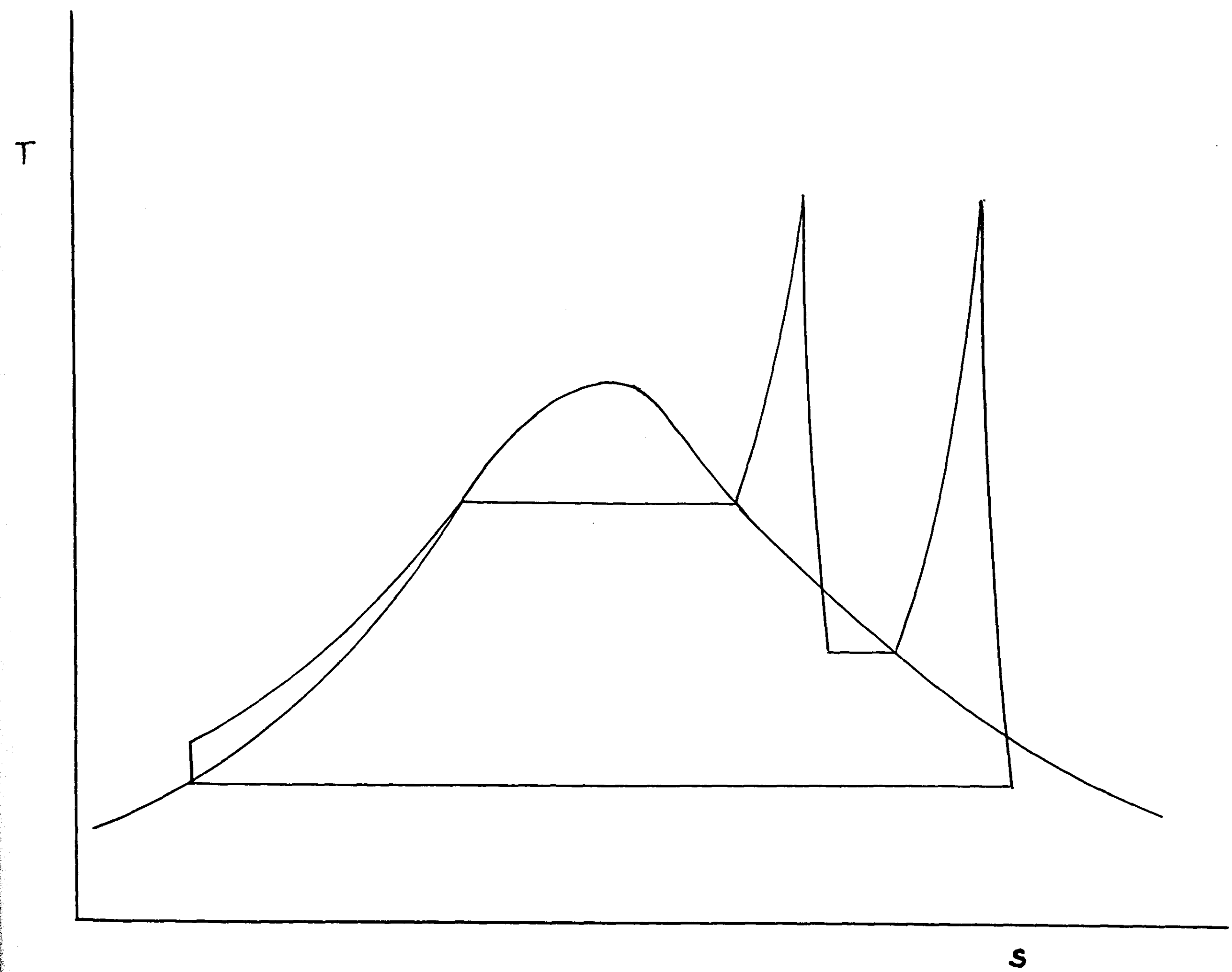



FIGURE 4
THE REHEAT CYCLE

## C. COMPONENT DESIGN PROBLEMS

### Steam Engine

The reciprocating engine is a device which delivers work when provided with a supply of fluid at high pressure and a region of low pressure into which the fluid may be exhausted. Such an engine is depicted schematically in Figure 5. For purposes of thermodynamic analysis, a plot is made of steam pressure versus displacement of the piston as in Figure 6. This plot is also known as an indicator diagram.

Referring to this indicator card, steam is admitted from the boiler at A, the point where the inlet valve opens. Cut-off occurs at CO, the point where the inlet valve closes. The steam is then allowed to expand to R where the exhaust valve opens. This is the start of the exhaust phase which lasts until K, when the exhaust valve closes. The cycle is then completed as the remaining steam is compressed back to point A.

Since the volume of steam enclosed in the cylinder by the piston is directly proportional to the piston displacement, the area under the curve on the indicator diagram represents the work done by the steam on the piston. This concept can be looked at in another light. A mean effective pressure on a piston is defined as that pressure which when multiplied by the area of the piston and the stroke gives the net work done on the piston in one cycle of events. Thus, the mean effective pressure (mep) multiplied by the change in displacement equals the area under the curve shown on the indicator card. An increase in boiler pressure or cut-off, or a decrease in exhaust pressure, tend to raise the mep.

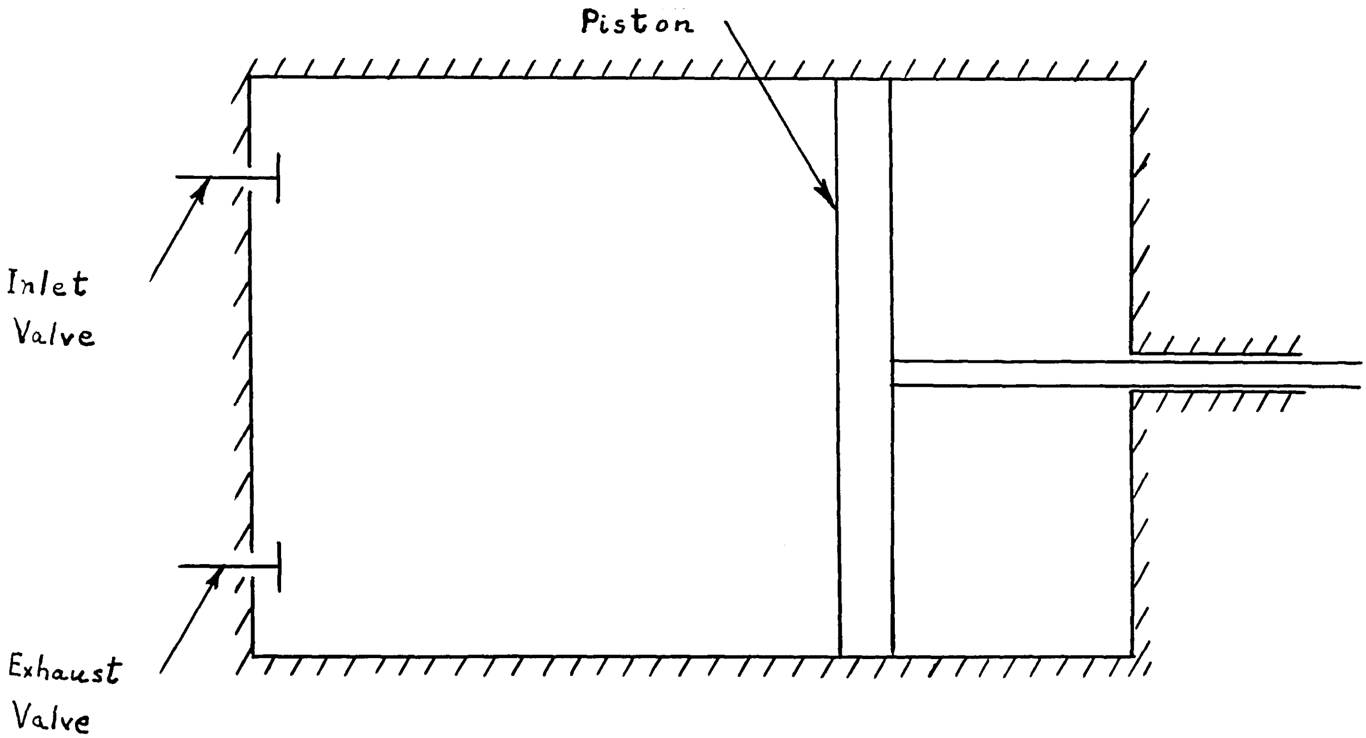



FIGURE 5
SCHEMATIC DRAWING OF
STEAM ENGINE CYLINDER

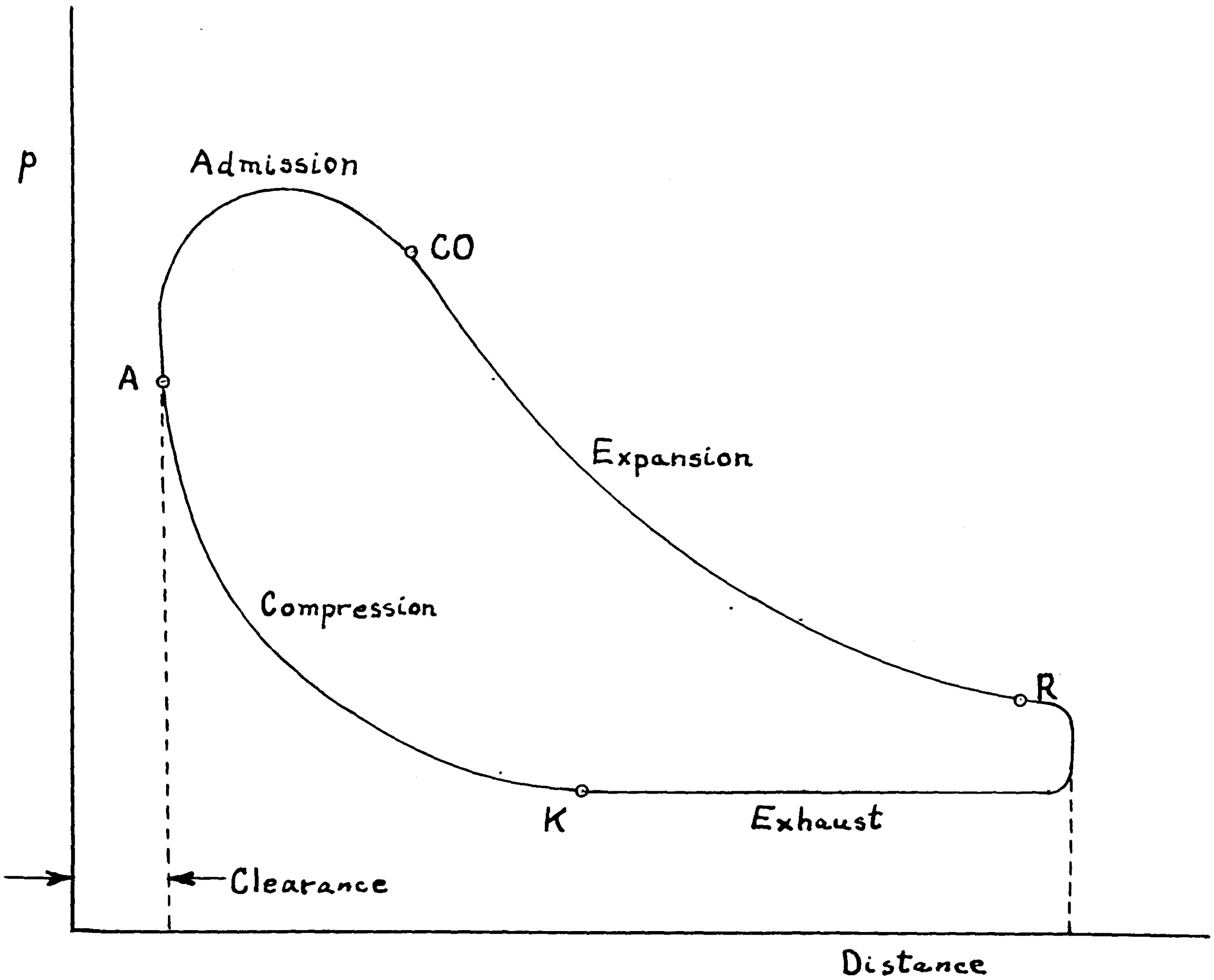



FIGURE 6

INDICATOR DIAGRAM

The mep can readily be varied in the steam engine by changing the point of cut-off. Also, initial and final pressures may vary widely, depending on boiler capacity and cut-off respectively.

The most important cause of reduced engine efficiency, as compared with the ideal engine, is not directly evident on an indicator diagram. When hot inlet steam flows by walls which have been cooled a moment before by exhaust steam, a large transfer of heat occurs from steam to walls. This hot steam is then cooled and sometimes a large portion is condensed (initial cylinder condensation). Similarly, on the exhaust stroke heat is transferred from the walls to the colder steam. Both these processes are inherently irreversible. Hence, a loss of work results, as is shown in Figure 7, where the dash and solid lines indicate, respectively, operation with and without transfer of heat. These effects are large enough to justify various modifications of the design.

Superheating the steam before it is admitted to the cylinder will decrease the moisture present during the exhaust stroke. The resistance to heat transfer at this end of the cycle is much greater and the fall in wall temperature less. Incoming steam, exposed to hotter walls, gives up less heat to them. By superheating sufficiently to insure no condensation before cut-off, the efficiency of a small engine has been increased as much as 40 per cent. There are numerous cases that show efficiency gains of 10 to 20 per cent due to superheating.

If the expansion phase of the engine cycle is carried out in two successive cylinders, the temperature difference between inlet and exhaust for each cylinder may be reduced to half the overall difference. Thus the temperature difference between steam and walls are in turn reduced to about

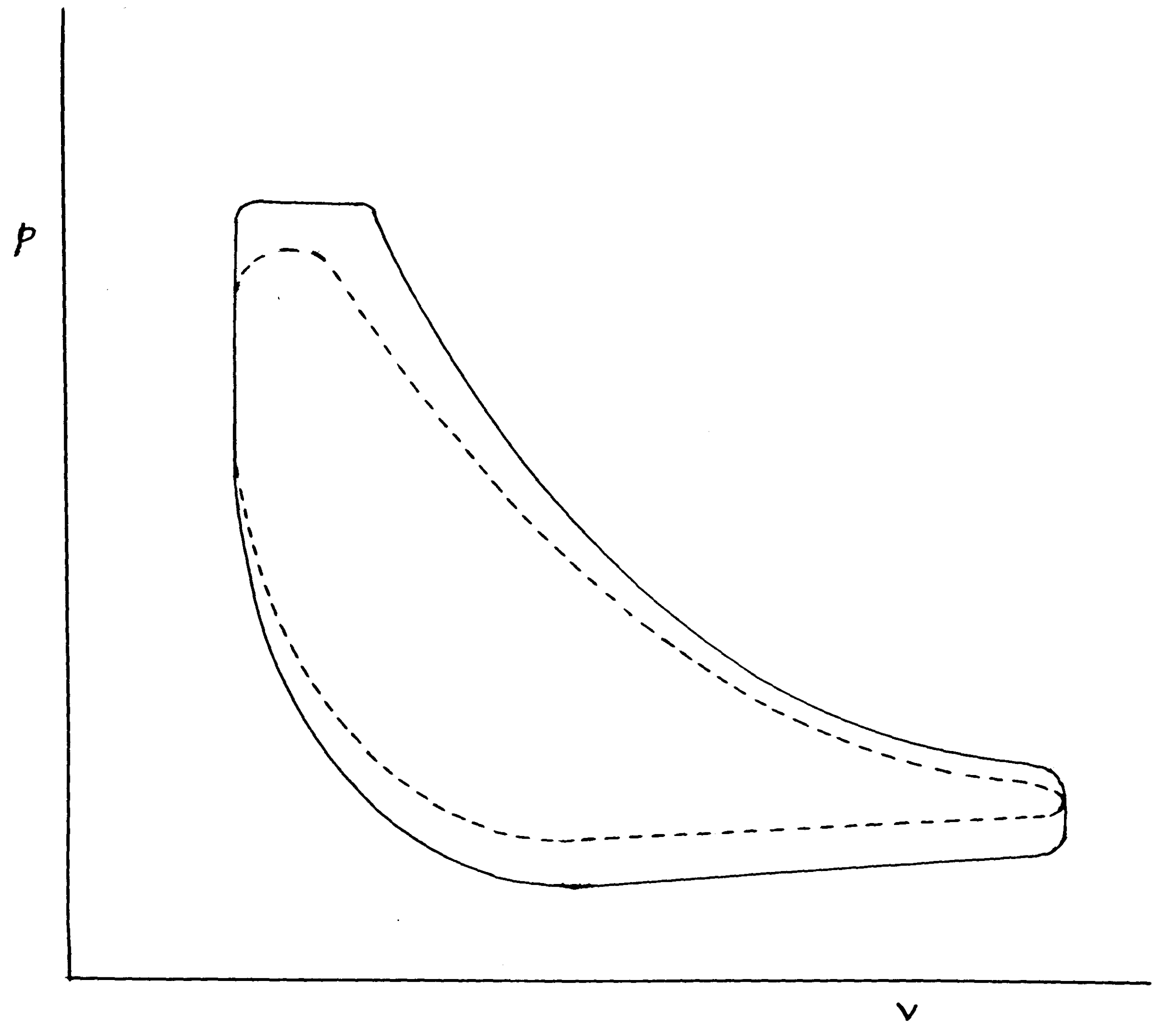
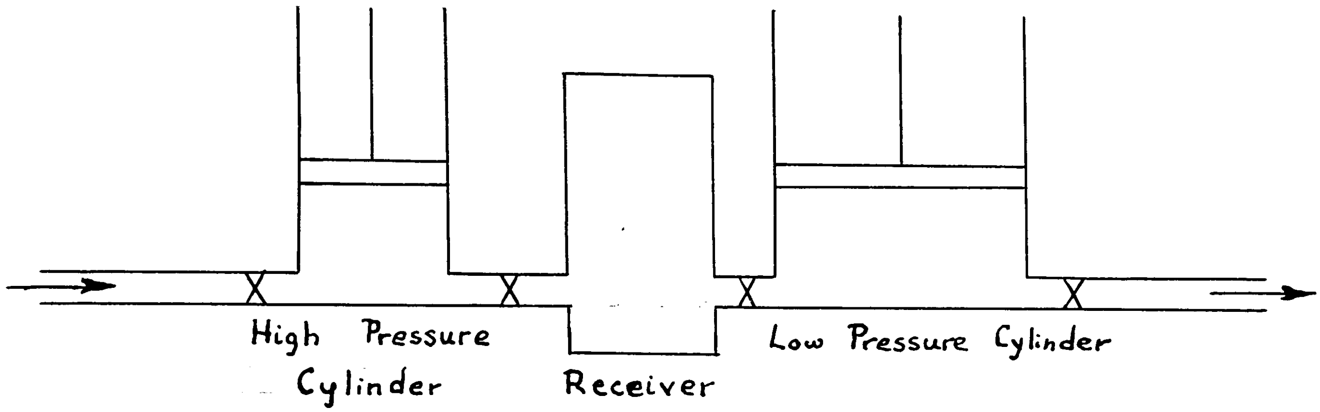



FIGURE 7
CYLINDER CONDENSATION

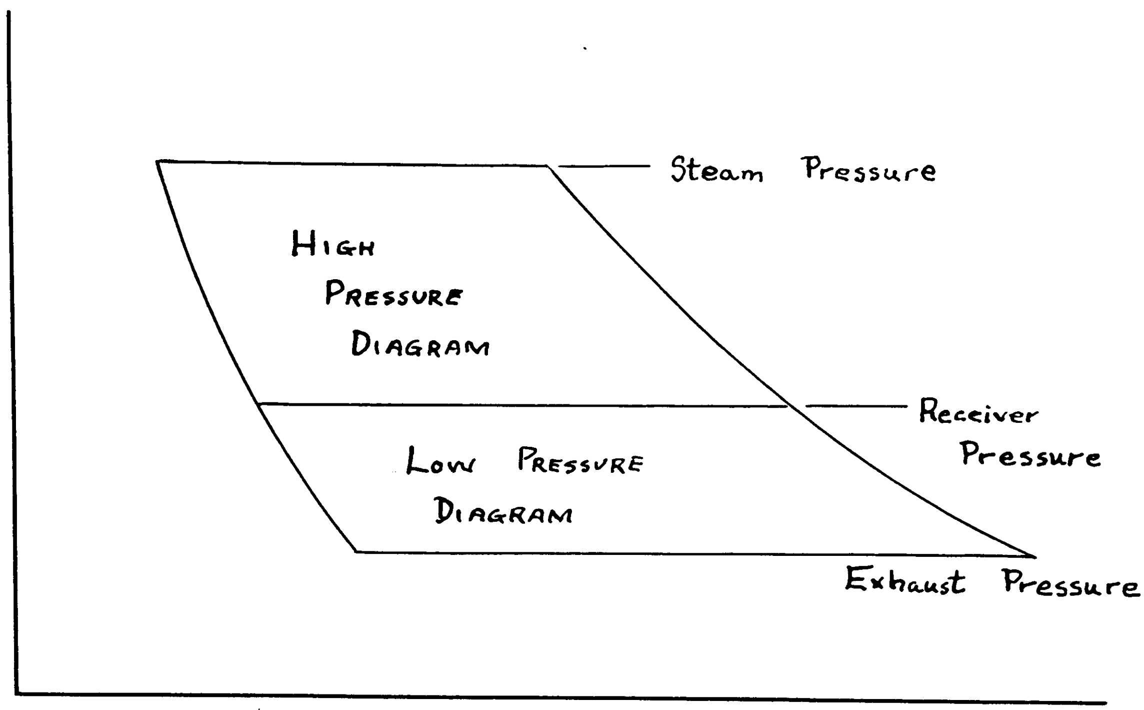
half. Such a compound engine is shown schematically in Figure 8. The low pressure cylinder is of the same dimensions as the simple engine, while the high pressure cylinder is much smaller. The wall surface is <u>not</u> increased to the same proportion that the temperature difference between the steam and the walls decreased. Therefore, the heat transfer between steam and walls is reduced.

An appreciable amount of cylinder condensation can take place in the engine ports especially in a slide-valve engine where these ports are subjected both to admission and exhaust temperatures. A logical modification here is to separate the steam and exhaust ports. Another is to use valves which can be placed close against the end of the cylinder (for example Corliss valves) so as to reduce the area of the port walls to a minimum.

The separation of steam and exhaust ports is carried to its logical conclusion in the uniflow engine (Figure 9). The uniflow principle has for its object the elimination of one of the greatest losses in reciprocating steam engines, namely -- initial condensation. In this arrangement cylinder condensation is reduced by eliminating the usual reversal in the direction of the flow of steam between admission and exhaust. Hot steam is admitted at the ends of the cylinder, and cool steam is exhausted through the ports arranged around the center of the cylinder which are uncovered by the piston at the end of its expansion stroke. By virtue of this design, the basic uniflow advantage is seen in that hot steam never comes in contact with the middle ports of the cylinder and cold steam never flows past the hot ends of the cylinder. Consequently, condensation is minimized.




(a) SCHEMATIC OF COMPOUND ENGINE




(b) INDICATOR DIAGRAM FOR IDEAL COMPOUND ENGINE

FIGURE 8

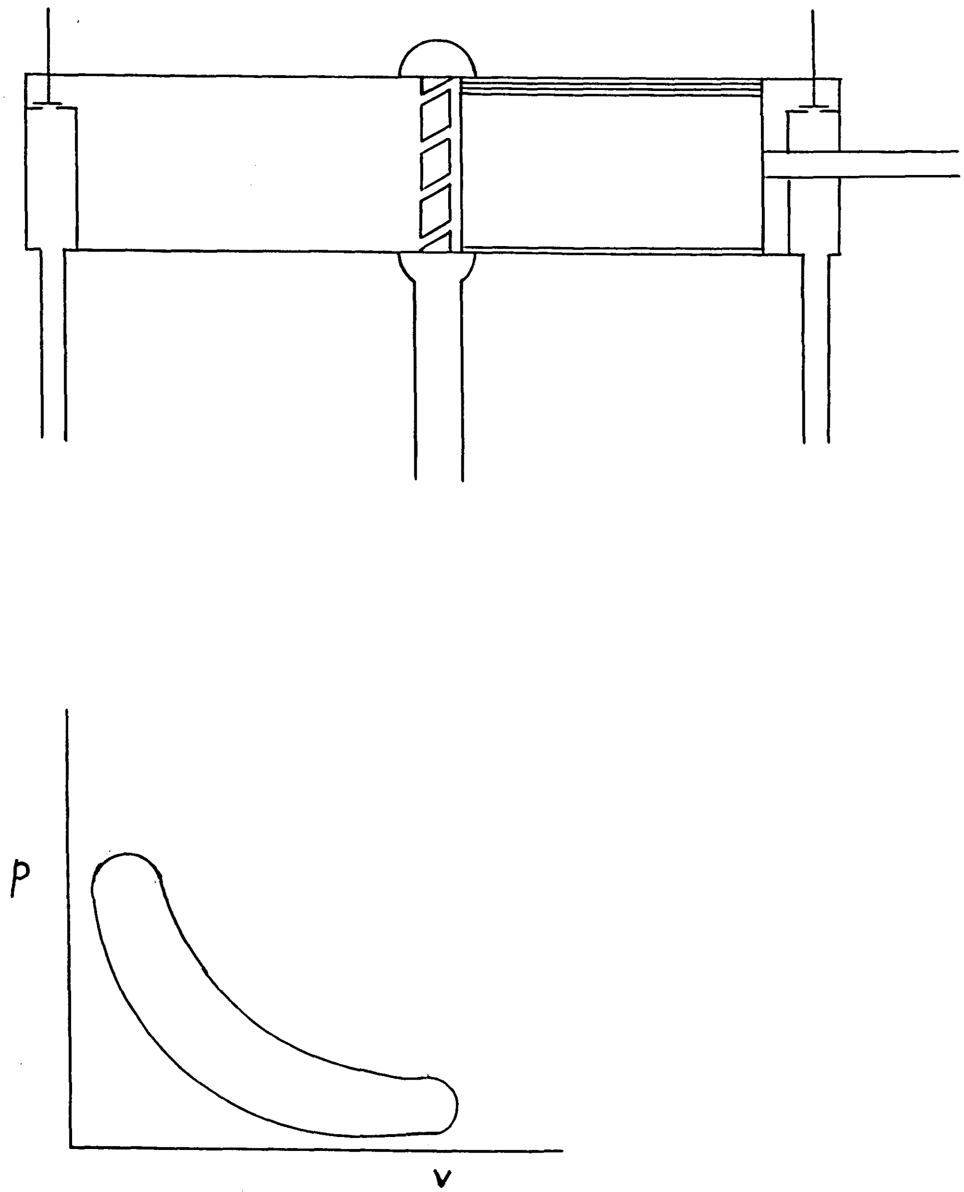



FIGURE 9
*THE UNIFLOW STEAM ENGINE*

It is noted that compression occurs early in the return stroke of the piston under the uniflow design. Hence, the uniflow arrangement is not well adopted to atmospheric exhaust pressure because the early closing of the exhaust port causes excessive compression pressure. On the other hand, the uniflow engine is well suited to the use of high vacuum (low absolute exhaust pressure) due to the exhaust port configuration which provides the great flow area necessary to pass large volumes of exhaust steam without excessive pressure drop.

Figure 10 shows the indicator card for this original European uniflow cylinder operating condensing. Compression begins when the piston covers the central exhaust ports on the return stroke, and continues for the remaining 90 per cent of the stroke. With the small clearances usually used (2 to 4 per cent), compression will not rise above the initial pressure only if the engine is operating condensing with a good vacuum.

Figure 11 shows the original European noncondensing indicator card. Again, compression begins when the central exhaust ports are closed by the piston, and continues for the remaining 90 per cent of the return stroke. However, large clearances (12 to 20 per cent) are required to prevent the compression at the end of the return stroke from exceeding the initial (boiler) pressure. This large clearance is, however, highly detrimental to steam economy.

In America, auxiliary exhaust valves are generally used when operating the uniflow cycle noncondensing. These valves remain closed on the expansion stroke and are opened on the compression stroke, delaying the beginning of compression for 60 to 70 per cent of the return stroke as shown in Figure 12.

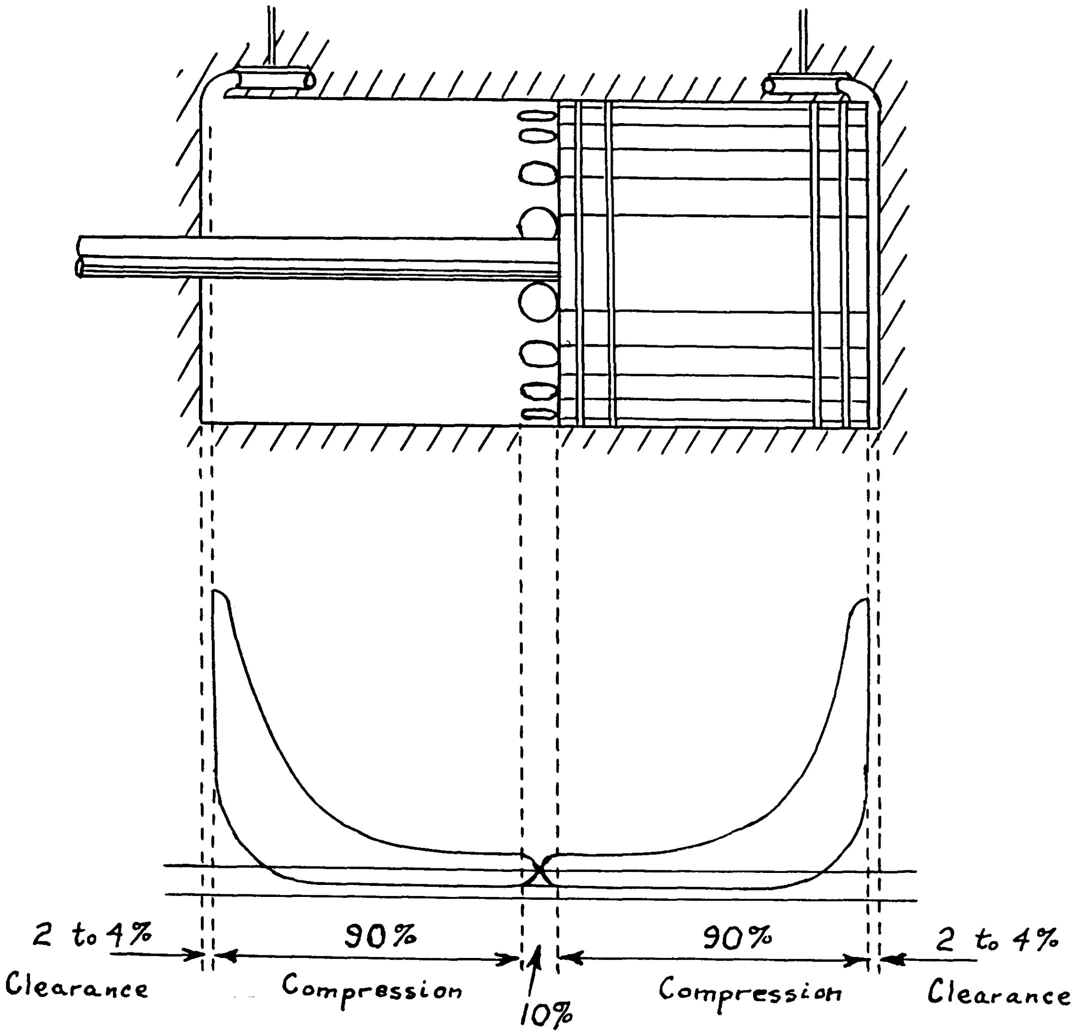



FIGURE 10
EUROPEAN UNIFLOW CYLINDER
OPERATING CONDENSING

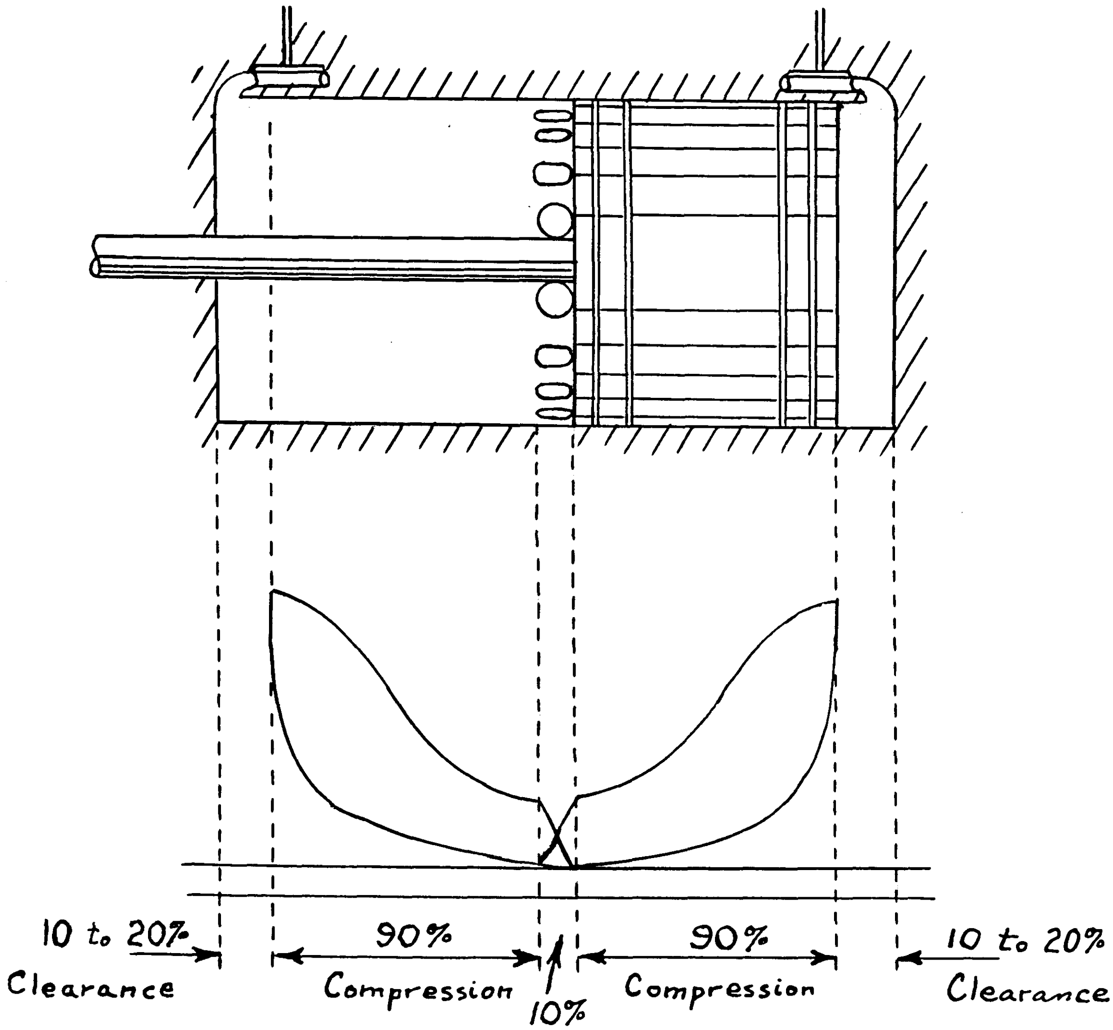



FIGURE 11
EUROPEAN UNIFLOW CYLINDER
OPERATING NON - CONDENSING

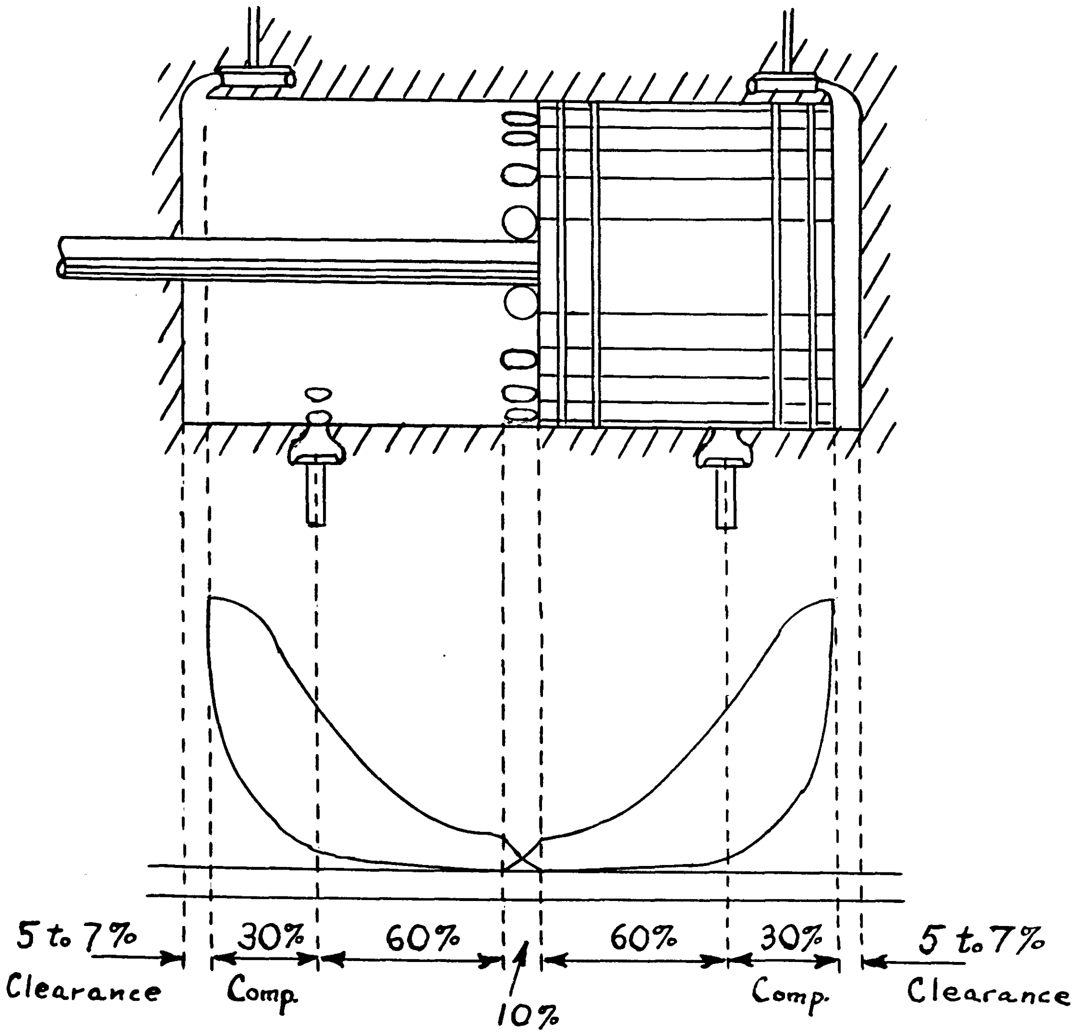



FIGURE 12

NON-CONDENSING UNIFLOW CYLINDER WITH AUXILIARY EXHAUST VALVES

"It is the use of early cut-off, absence of initial condensation, reduction of volumetric clearance and the employment of steam-tight valves, that is responsible for the flat economy curve of the Skinner uniflow engine, from very light loads to overloads".

Poppet valves, similar to those of the internal combustion engine, are widely recommended for use with the uniflow cylinder. This is due to many factors, including symmetrical construction, small size, large port opening with small valve lift, and absence of friction (occurring with valves that slide on their seats). In addition, they are well adapted for use with high temperature superheated steam. In designing for high speed engines, however, valves must be larger to prevent "wire drawing" or excessive pressure gradients before cutoff.

## The Condenser

The condenser has long been a major thorn in the side of steam car manufacturers. In order to operate on a closed cycle the system must fully condense all the engine exhaust steam to the saturated liquid state. This liquid can then be pumped to boiler pressure and started over again on another cycle of events. If the system is not fully condensing, as was the case on all early model cars, it is necessary to replenish the water supply at frequent intervals.

When the uniflow engine is used, it is desirable to operate the condenser under a partial vacuum of about 1 to 5 psia (see engine analysis). Under these conditions, maximum use is being made of the uniflow principle. Cycle efficiency is also increased by using this lower pressure level (as compared with atmospheric exhaust at the same boiler pressure).

A condenser large enough to handle all the exhaust steam is necessary on closed cycle operation. Unfortunately, a conventional automobile radiator is not adequate. When the engine is developing high torque and steam expansion ratio is low, the required rate of heat dissipation necessitates large amounts of usable condenser surface area.

There are several alternative appraoches to the condenser problem (Reference 12)

1. No attempt is made to condense all the steam when the heat loss to the condenser becomes higher than some predetermined level. This has been the case in most steam cars in the past.
2. The condenser is made large enough to cope with the worst operating conditions.
3. Means are provided to obtain added cooling capacity
4. Exhaust heat is used for power
5. The vapor cycle is altered so that some heat is put back into the system.

Solutions 1 and 2 are obvious. One way - 3 - has been applied in the past to have the exhaust steam operate a cooling fan by means of a small turbine, so that added cooling capacity was obtained at lower speeds which are likely to exist when added condenser capacity is required. Solution 4 could be applied by using the above mentioned turbine as a supplementary power source. Solution 5 can be effected by raising condenser temperature and pressure, resulting in efficiency and power losses.

However, the possibility of high fluid loss is eliminated.

## The Boiler

Boiler specifications are generally determined by cycle requirements. High pressures are desirable from an efficiency viewpoint as was pointed out previously. However, this pressure may be limited by a number of factors, one of which will be dominant for a given application. The first is a metallurgical limit, which depends on the cost of the material and the maximum temperature it can withstand. Another factor to be considered is the condensation that can be tolerated at the engine exhaust. This consideration determines the amount of superheat required above the saturation temperature at boiler pressure. Finally, boiler pressure must be sufficient to produce the necessary number of expansions that take place in the engine over the range of its operation.

Due to size limitations imposed by automobile design, the boiler and its accessories should be as compact as is practicable. A variety of fuels can be used for the burner. The choice of fuel can be made on the basis of relative cost, higher heating values, rate of combustinn, and the rate of heat loss by radiation and convection. As for the burner, an atomizing type with electrical ignition seems to be suitable for motor vehicle use.

The greatest problem in developing an efficient steam generator is that of obtaining high heat-transfer rates from the combustion flame to the fluid.

The ability to achieve these high rates will not only reduce the size and weight of boiler tubing, but will also decrease the need for carrying large quantities of high temperature and pressure steam. Highest heat-transfer rates are obtained with the greatest temperature difference between flame and fluid. However, combustion gases may be hot enough to effect the desired heat transfer within reasonable size limitations.

Boiler maintenance may be a problem, especially when a lubricant remains mixed with the steam on a closed cycle operation. The lubricant would tend to coat the boiler tube walls, thereby decreasing heat transfer and resulting in the eventual blistering of the boiler tubes. A means of maximizing lubricant separation before the boiler has to be found in this system.

The boiler is responsible for slow starts in cold weather. A definite undesirable time lag is exhibited between the firing up of the boiler and the availability of steam at proper temperature and pressure before take-off. One possible solution to this problem involves the use of a flash boiler which heats only a small volume of feedwater at any given time.

Along the same lines, boiler feedwater freezes when left overnight in wintertime. An additive may be introduced into the fluid to prevent this. However, steam properties might change appreciably and could occur inside the boiler tubes. Another approach would be to build the water tank with sloping walls and allow the water to freeze. On start up hot flue gases could be passed over the tank to aid in melting the frozen feedwater.

## Auxiliary Equipment

The boiler, engine, and condenser comprise the three principal components that go into a steam car. In addition, there are numerous other parts, all of which perform a specific function. In many cases these functions are necessary for the proper operation of the car. The rest, although not essential, are useful in many instances. This group include those devices which recover "waste" heat for use elsewhere in the vehicle, safety installations, and luxury items.

Each of the above mentioned basic components requires auxiliary equipment to assist in the cyclic performance of events. For instance, the boiler needs burners, a feedwater pump, and a fuel pump and tank. The engine must be lubricated, and in turn the lubricant must be separated from the exhaust steam. The condenser requires a fan to help speed the cooling air flow, and a vacuum pump to maintain a partial vacuum on the steam side.

In addition, the piping system has to be air-tight. Control and safety valves have to be put at key points, with important measuring instruments supplied to the dashboard. A battery and generator are used to provide the necessary electrical power.

It may be desirable to derive the power for much of this auxiliary equipment from a small turbine rather than directly from the engine. Such a turbine could receive exhaust engine steam and empty into the condenser. It could be used to power many of the pumps, the condenser fan, and the generator.

Below is a partial list of auxiliary equipment that could be used on a steam car. Each of them is designed for a particular purpose, which, depending on circumstances, may be better achieved in other ways or eliminated entirely.

1. Pumps

    Feedwater pump

    Fuel pump

    Oil pumps

2. Fans

    Condenser fan

    Combustion air fan

3. Electric Generator

4. Battery

5. Storage Wells

    Water tank

    Fuel tank

    Oil tank

6. Separators

    Oil separator

    Air filter

7. Exhaust Piping For The Combustion Products

8. Instrumentation

    Ammeter, Water level

    Boiler pressure

    Oil pressure (bearing and cylinder feed)

    Fuel level

    Speedometer -

9. Controls

   Steering wheel

   Reverse lever

   Brakes

   Throttle

   Cut-off valve

   Ignition

10. Safety Devices

   Boiler pop safety valve

   Atmospheric exhaust system for engine

   Horn

   Lights

11. Luxury Items

   Heater - Air Conditioner

   Radio

## D. HISTORICAL BACKGROUND

Steam car development took place in two fairly well defined periods. The first was from the earliest days of steam cars up to about 1913, and the second was from shortly after World War I to the late 1920's. Some famous names in steam cars included the Stanley, Doble, Stelling, Scott-Newcomb, and White. By 1930 the market for these cars had all but vanished.

Edmund B. Neil* has presented what he feels are the reasons for the historic downfall of the steam automobile. Some of the technical difficulties were:

* Ref. 12

1. Long cold weather starting time. This was largely overcome in later designs (to be discussed).
2. Complexity of controls. Proper operation of the car depended on the driver's ability to use these effectively.
3. Relatively high fuel consumption for developing high torques at low speeds or for handling large loads.
4. Lack of sufficient boiler capacity for sustained high speeds and loads.
5. Lack of suitable materials to reduce car weight, efficiency, cost, etc.
6. Need for frequent replenishment of water supply.

A fairly complete description of a late model car made by the American Steam Automobile Company can be found in reference 2 . This car used a water-tube boiler made of seamless steel tubing capable of holding pressures up to 1000 psi. The boiler was provided with a spring - loaded safety valve, a water level guage, and a water level regulator. Any sediment that collected in the mud drum (located outside the burner) was blown out through bottom blow-off valves.

The engine was a two-cylinder double-acting locomotive type with plain slide valves and a link motion for reverse. It was placed horizontally with the steel gear on the crank shaft engaging the main gear of the differential, thereby effecting direct power transmission from the engine to axle. The driving gear and differential were enclosed in an oil tight case and run in an oil bath. The cylinders were lubricated independent of the crank case by means of a positive pressure cylinder lubricator driven from the water pump cross head. Oil consumption in the cylinders amounted to one gallon every thousand miles. The lubrication problem is simplified due to the fact that there are only twenty moving parts, exclusive of the roller bearings. Since the engine was geared directly to the rear axle with a 1-1/2 to 1 reduction in general, these parts also moved at a comparatively slow pace, thereby decreasing frictional effects. Another striking feature of the engine was its almost complete lack of vibration at any speed. Reverse motion was actuated by a foot pedal which reversed the valve motion. The throttle valve was a combination of a poppet and piston valve. Efforts at using uniflow engines had not proved effective at this date.

The electrical system was primarily used for lights and the horn. A six volt battery was used in conjunction with a generator which was mounted on and driven from the rear axle.

A radiator was located at the front of the car to condense exhaust steam. Air was used as a coolant, but no fan was present as it was found to be unnecessary. The radiator could not possibly freeze in cold weather because any condensate would drain out as soon as it was formed.

Vaporizing type burners were most widely used with the boilers. These consisted of two parts, a main burner and the pilot. When the boiler pressure fell below some predetermined point, on automatic valve in the fuel line opened and supplied fuel to the main burner until the pressure reached a fixed maximum. Then the valve closed. This limiting pressure could be adjusted up to 750 psi. The fuel for the main and pilot burners was supplied from two small pressure tanks which in turn were supplied from a main tank. Gasoline was ordinarily used for the pilot light, while gasoline, kerosene, or range oil was used in the main burner.

The latest type burners were of the atomizing type. A special 6 volt motor drove a standard fan which supplied air, and an oil pump which supplied oil to an atomizing nozzle of standard design. Ignition was effected by a 6 volt spark coil and a standard spark plug. The oil pressure was regulated by an automatic valve. A cut-off valve was used between the pump and the nozzle to stop the oil flow when the motor was off. This burner was controlled by an automatic switch which opened on rising pressure, thus holding boiler pressure constant. The burner lit instantly when cold, generating steam in less than three minutes. No pilot light was necessary.

The life expectancy of an average steam car was 10 to 15 years. The reason for this is:

1. the total number of moving parts was about forty,
2. the engine ran at low speeds, and consequently the parts moved slowly,
3. the engine oil was not contaminated.

The 1923 Stanley, as described in reference 11 , was made in one chassis model which had an 130 inch wheelbase and 32 x 4-1/2 inch tires. The boiler was of the fire-tube type, drum shaped, and stood on end. It contained 640 half-inch tubes in a case 23 inches in diameter by 18 inches high. Its maximum capacity was 23 gallons, but the normal water level content was 16 gallons. The necessary wall strength of this boiler was achieved by winding three layers of piano wire around a thin shell. Hence, a lighter boiler was made possible.

The water supply was carried in a 24 gallon copper tank, which was located at a low point in the car, under the front seat. To avoid injury due to freezing, the tank had flat surfaces, with the front and rear ones slanted. (By passing combustion waste products over the front slanted surface, more heat could be transferred to the bottom surface of the tank, and the feedwater was kept warmer). Since the tank sat low, condensate water naturally drained back to it. Although all the exhaust steam was fed to the condenser, some was being continuously lost, with the result that the tank had to be refilled every 150 - 200 miles. Boiler feed was accomplished by a double-headed plunger pump, and the boiler water level was maintained by a feed-water regulator. Bottom blow off valves were provided to keep the boiler free from lime deposits, etc. A water level blow off valve (skimming valve) was also supplied to blow off any lubricating oil that might get into the boiler and collect on top of the water.

Fuel was carried at the rear of the car in a drawn steel fuel tank. The tank was made up of two compartments, a 20 gallon kerosene and a 7 gallon gasoline section. Fuel was drawn out by means of a plunger pump which was actuated by the same rod as the water pump. The main pressure regulator controlled the flow of fluid into the pressure tank, as well as the main burner. This car used a vaporizing system similar to the one described above. The pilot light served three functions:

1. to maintain steam pressure in the boiler;
2. to keep the fuel system hot (ensuring quick starts after standing overnight or for long intervals);
3. to ignite the main burner as it was turned on.

A safety type throttle valve was used with the boiler, i.e., the valve opens up against boiler pressure. This valve was placed between the boiler and superheater.

A two-cylinder double-acting engine with a 4 inch bore and 5 inch stroke was arranged horizontally at the rear of the frame and geared directly to the rear axle with spur gears. It used slide valves of the D type made of cast iron. The valves were operated by the usual Stephenson link motion which permits varying cut off and reversing direction of rotation of the engine for backing the car. Cut off was arranged so as to give either 28 or 60 per cent of the stroke. The long cut off provided smooth starts and increased power at low speeds, while the more economical short cut off was used at high speeds and under normal conditions. Reversal of the engine and change in cut off was effected by a foot pedal. The maximum temperatures in the engine cylinders rarely exceeded 700$^{o}$F. Six quarts of lubricating oil poured into the housing provided sufficient cylinder lubrication (again, independent of bearing lubrication).

Flexible metal hose carried exhaust steam to the condenser which was 23-1/2 inches wide, 24-5/16 inches high, and 4-3/16 inches deep. This fin and tube type condenser was located at the front of the car and drained all its condensate water immediately to the water tank.

The electrical system was similar to the one described before.

Another car that sold well after World War I was the Doble. Some features used on these cars included a flash boiler, a turbine to drive the fan used to cool the condenser, and a four-cylinder double-acting compound engine. A 1926 model using such an engine put out about 120 h.p. at 900 rpm. A flash boiler supplied this engine with superheated steam at 210 psi working pressure from a 30 gallon tank, enough for 750 miles. Full steam could be raised in less than a minute and the car was capable of 60 mph at 900 rpm. Top speed was better than 95 mph; gas was consumed at 8 mpg in town and 11 mpg on the open road; oil consumption amounted up to 4,000 mpg.

More recently a steam car was built in Madison, Wisconsin, by Charles F. Keen. His steamliner uses a 24 inch diameter boiler that stands 28 inches high and which is capable of a maximum steam pressure of 1500 psi. This single tube flash boiler steams up from a cold start in 30 seconds to one minute, and will burn gasoline, kerosene, or furnace oil without any adjustments. The burner is of the atomizing type, is spark ignited, and its working pressure is maintained by automatic control of the fuel supply. He also uses a turbine powered fan to draw air through a radiator-like condenser.

Another recent steam car design was attempted by the Paxton Division of the McCullock Motors Corporation. Economic considerations caused the project to be dropped. However, a few design features of the proposed cars are of interest. The engine was a 6 cylinder compound uniflow design. Poppet valves were used to admit steam to the high pressure cylinder and transfer it to the low pressure cylinder. An automatic cut-off control continuously adjusted cut-off to give maximum economy at any operating condition and shifted to long cut-off when the throttle was opened for a burst of power. The engine was designed to accept inlet steam as high as 1200°F and 1800 psia, while the exhaust was to be into as high a vacuum as could be maintained by the condenser and vacuum pumps. The boiler was a flash type capable of a twenty second start up. The fan used with the condenser was operated by the engine at low speeds and at high speeds by a turbine operating off exhaust engine steam.

The Paxton car has advisedly attempted to combine the better features of older steam cars. In any future design it would be wise to draw on past experience as outlined above for a working guide to innovation and improvement.

## E. ADVANTAGES AND DISADVANTAGES OF A STEAM CAR

The advantages derived by using a steam cycle as the basis of an automotive design are:

1. The power producing processes are separately performed and can readily be observed. Control of various phases of the cycle can increase efficiency, work output, or economy as desired.
2. The vapor generator can be adopted to handle a variety of fuels, including solids or pulverized fuels. The fact that low-cost fuels can be used may be very advantageous.
3. Combustion products are separated from the working fluid. Therefore, contamination of the fluid, engine, or lubricant cannot result.
4. The wide variations in mean effective pressures obtainable by a steam engine give high torque and power whenever it is needed.
5. Fewer moving parts simplify maintenance problems and increase the lifetime of an automobile.

6. Miscellaneous advantages include:

   a. virtual silence of operation,

   b. reduced vibrational problems,

   c. ease of installing an all weather air-conditioning unit,

   d. a flexible design for arranging parts in a vehicle,

   e. infallible cold weather starts,

   f. smooth acceleration characteristics.

Some disadvantages of steam cycles for this application are:

1. Large heat losses to the condenser.
2. Frequent replenishment of the water supply may be required, especially on open-cycle operations.
3. Inherently long starting up time.
4. Freezing of water in cold weather.
5. A larger battery and generator may be needed. This is due to the elimination of engine idling.

The historical pitfalls of steam cars have been presented in the previous section. Most of these can be met successfully at the present time. However, there now exists a more formidable stumbling block in the path of building a steam car suitable for mass production. This barrier takes the form of an economic consideration. It does not seem profitable to invest large sums of money to develop a steam car to the point where it could compete with the internal combustion engine design. The latter has had over fifty years of intense technological activity dedicated to improving any poor features that might have existed. At the present time it can do almost anything a steam car is capable of. But, there are differences which do exist between both these opposing automobiles. Some of these are pointed out in the chart below.

| | Internal Combustion Engine Car | Steam Car |
|---|---|---|
| Fuel | Gasoline | Variety of Boiler Fuels |
| Maintenance | Many moving parts - frequent repairs; replenishment of gas at frequent intervals, oil at regular intervals. | Few moving parts - infrequent repairs; replenishment of boiler fuel at frequent intervals, oil regularly, water either as needed or at frequent intervals. |
| Cost | Varying initial costs, high upkeep | High initial cost; low upkeep |
| Durability | Life expectancy of about 10 years | Life expectancy of about 25 years |
| Start-up Time | Instantaneous (though not advisably so) | Warm start about 30 seconds Cold start about 3 minutes |
| Braking power of Engine | Low | More effective than hydraulic rim-shoe brakes. |

## F. SCOPE OF INVESTIGATION

Using the background information presented up to this point as a basis for further endeavor, we shall now pursue the problem of designing one particular steam sports car. As previously stated, this sports car is chosen with one eye on the economic aspects of the steam car problem. We feel that a steam-powered vehicle has its best opportunities for breaking into the automotive industry if it uses the sports car market as a stepping stone. This is due to the fact that a sports car enthusiast is willing to pay a higher premium for improved performance and durability.

The design of a steam car involves picking one particular cycle from which to work and designing suitable components to follow this cycle. Along these lines, we felt it useful to look into the cycle analysis a little more closely. We also undertook a more detailed study of the engine and condenser so as to be able to make better selections.

Incorporated in this car are some parts, advances, and ideas that have been used before. Our findings have also been included where they are applicable. A layout drawing of the car showing the locations of the various components is presented, as well as our reasons for making various selections.

Therefore, this study is concerned with a closer look at the cycle, engine, and condenser in connection with a layout of one particular sports car that could be built at present.

# CAR DESIGN

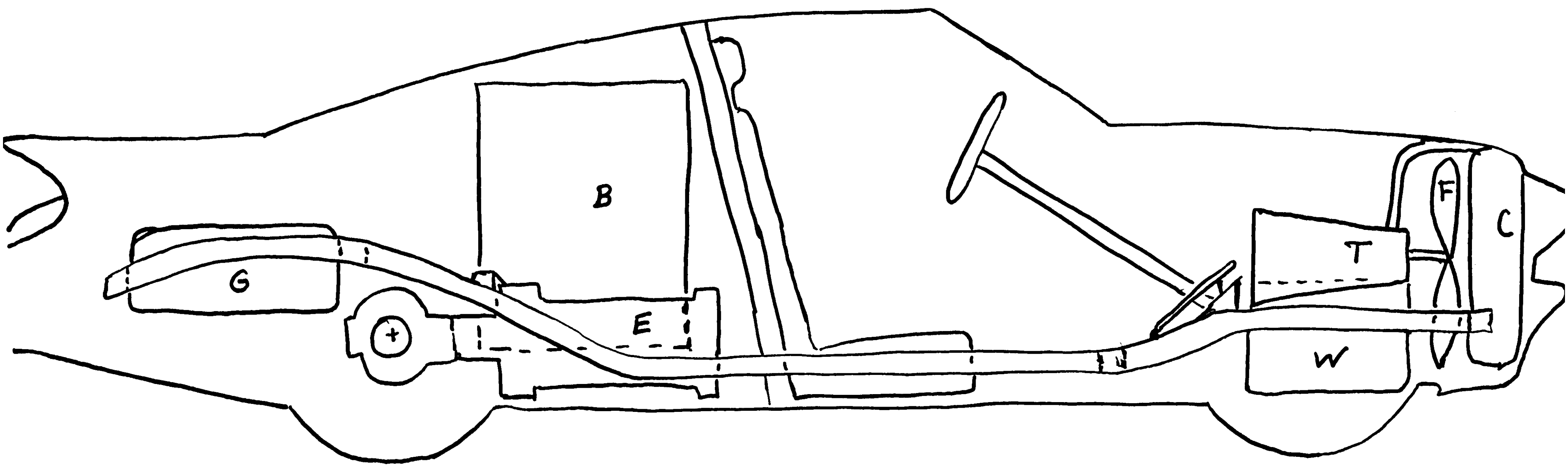
A layout drawing of the particular car under consideration is given in Figures 13a, b, and c. Only the principal components are indicated on this scale drawing which shows the relative sizes of various components. The steam cycle used as a basis for this design is plotted on a T-S diagram in Figure 14. Both of these illustrations provide a starting point for the discussion of the car in general.

A flash boiler is placed in the rear section directly behind the driver. This type of boiler is chosen because it allows only a small amount of feedwater to be heated at any instant. Many such boilers, capable of 1500 pounds pressure, have been produced. A typical one would be 20 inches round by 30 inches high, and would weigh 350 pounds.

Next to the boiler, and geared directly to the differential, is a two-cylinder compound double-acting uniflow steam engine, similar in design to the one shown in Figure 12. The engine is designed so that both cylinders supply approximately equal horsepower under all conditions to crank throws set 90 degrees apart to obtain fairly even torque.

Engine specifications are best presented in tabular form, as follows:

1. Material: alloy iron containing nickel, chromium, and molybdenum.
2. Steam inlet temperature: $T_1 = 660^oF$
3. Steam inlet pressure: $p_1 = 1100$ psia




B - Boiler
C - Condenser
E - Engine
F - Condenser Fan
G - Fuel Tank
T - Turbine
W - Water Tank

Figure 13(a)

Layout of Major Components

Scale : 1/16

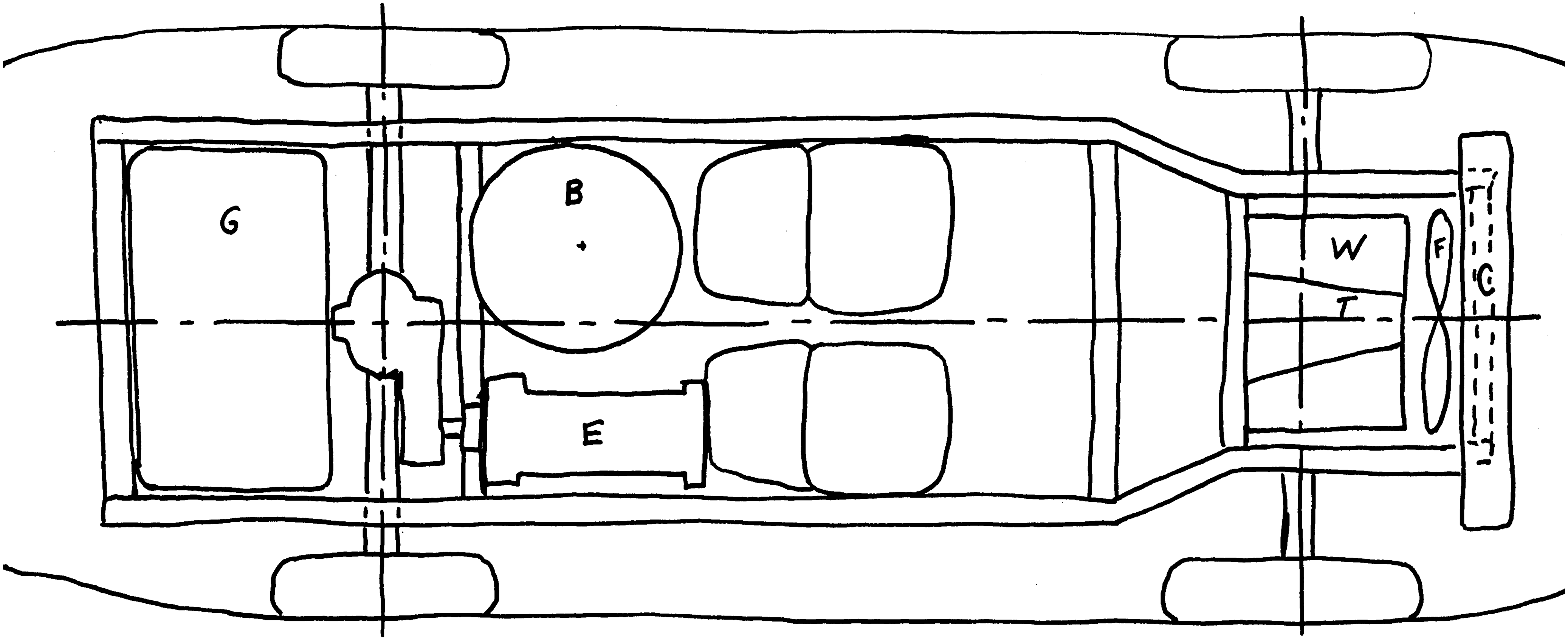



FIGURE 13(b)

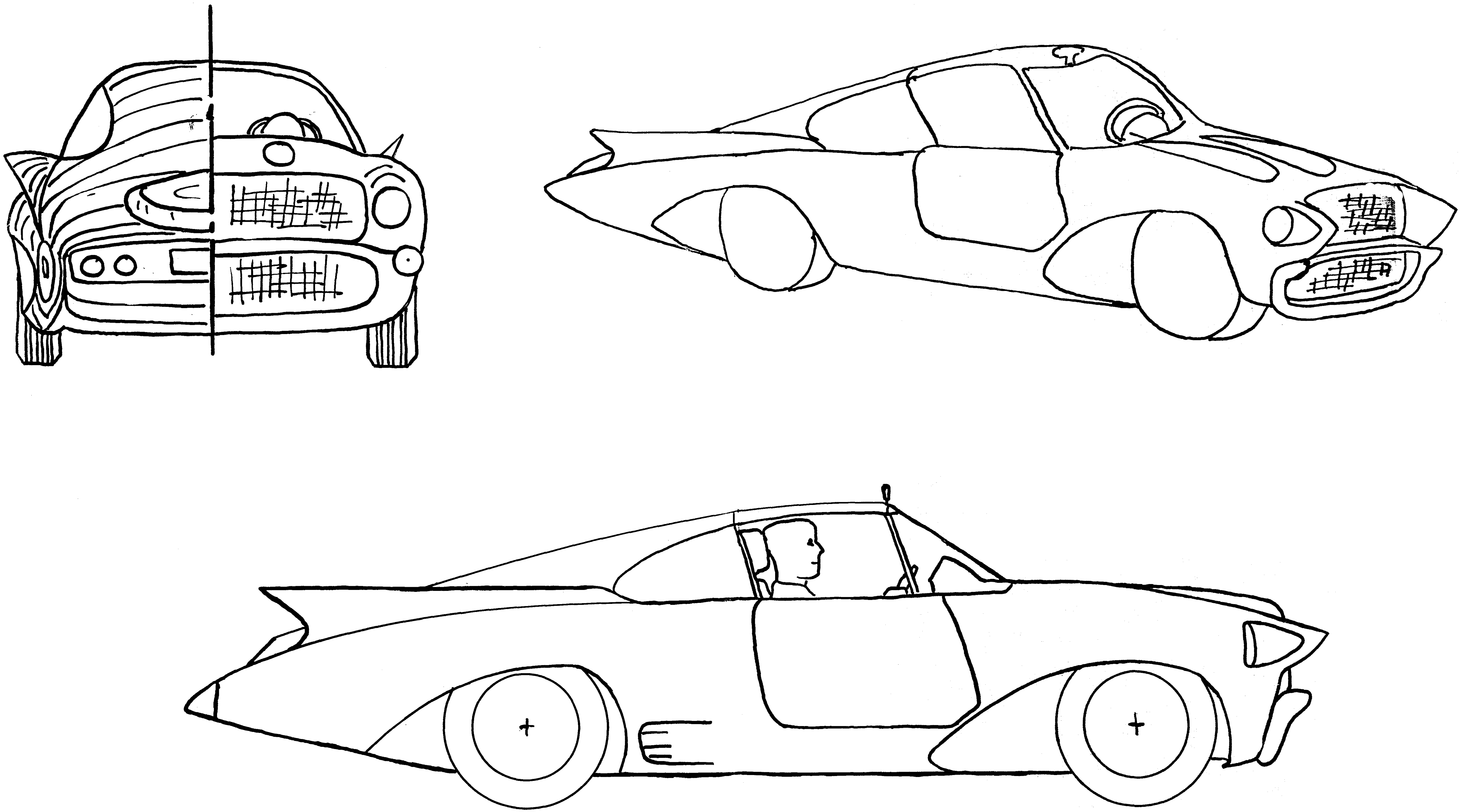


FIGURE 13 (c)

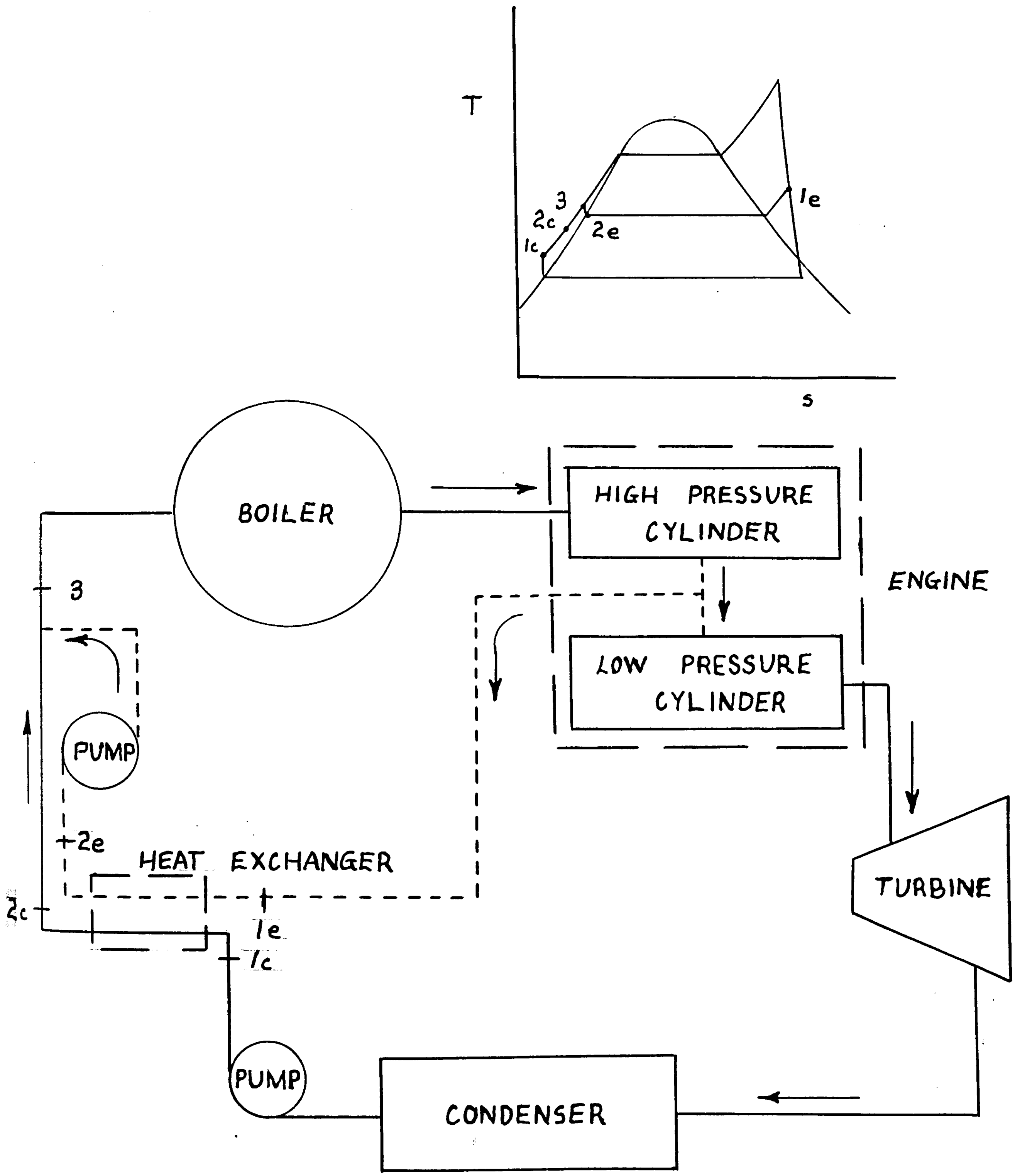



FIGURE 14
STEAM CYCLE

4. High pressure cylinder:

| | |
|---|---|
| Diameter | $D = 4.10$ in. |
| Area | $A = 13.2$ in$^2$. |
| Stroke | $L = 0.488$ ft. |
| Clearance | $C_H = 5\%$ |
| Normal Cut-Off | $= 10\%$ |

5. Low pressure cylinder:

| | |
|---|---|
| Diameter | $D = 8.36$ in. |
| Area | $A = 54.9$ in$^2$. |
| Stroke | $L = 0.82$ ft. |
| Clearance | $C_L = 4\%$ |
| Normal Cut-Off | $= 12.5\%$ |

6. For normal cut-off

Engine horsepower - 100 hp

Steam inlet pressure to second cylinder - $p_2 = 130$ psia

Steam inlet superheat to second cylinder - $20^\circ$F

Steam rate - $w = 12.8$ lbm/hp-hr

Engine exhaust pressure - $p_{2E} = 17.2$ psia

Further expansion of the steam is accomplished through a small turbine which normally exhausts to the condenser at 5 psia, producing approximately 20 hp. This turbine, located under the car hood, is used to power much of the auxiliary equipment, such as pump, fans, and the generator. When the engine is operating at increased flow rates (overload) some steam is diverted from the receiver (as is shown by the dashed line in Figure 14) and is used to preheat boiler feedwater in a simple heat exchanger. This regenerative system limits the flow to the low pressure cylinder, turbine, and condenser.

The condenser is placed at the front of the car directly in front of its fan. It is designed to operate at 5 psia and condense all exhaust steam for slight overloads above the normal 10% cut-off in the high pressure cylinder. A compact heat exchanger of the plate-fin type has been adopted for this purpose. It is capable of condensing 2,000 pounds of steam an hour at the above pressure and an ambient air temperature of $100^{\circ}$F. With a frontal area of about 8 ft$^2$ and a volume of about 10 ft$^2$, this configuration provides 3390 ft$^2$ of heat-transfer surface on the air side.

The engine and condenser were investigated closely in this design. Cycle characteristics were determined primarily by engine requirements, as is shown in the following pages. Boiler specifications are presented, although no attempt was made to design one. It is felt that there are many perfectly good boilers available that would fit well into the car. The condenser approach is original, and is presented in fairly complete detail. Other equipment has not been looked into deeply unless specific application was found pertaining to either the engine or condenser.

# THE STEAM ENGINE

## A. INTRODUCTION

From our discussion of steam engines, the uniflow type will be considered for use in our proposed sports car design. While this feature is aimed at decreasing the percentage rejection losses of a steam engine, only in conjunction with several other methods can this goal be realized. Some of the methods already discussed were increasing boiler pressure, superheating the steam, condensing at lower temperatures, and compounding. Others are increasing rotative speed, decreasing clearance and increasing ratio of expansion (decreasing cut-off). However, since practical restrictions are imposed upon these theoretical proposals, our engine design parameters will be chosen in view of these restrictions.

Increasing boiler pressure and superheating the steam involves higher temperatures. The limit for this temperature in turn depends on both construction materials and the lubricant. Of the two, lubricant, temperature limit is probably the most critical in light of the following discussion![1]

An alloy cylinder iron has been developed, containing nickel, chronium, and molybdenum, showing practically no growth or change up to 800 degrees F.

With a Brinell hardness of 240 and over in sections of 2 inch thickness, the tensile strength averages 50,000 pounds per square inch in these sections. The tensile strength exceeds that of cast steel in smaller sections. Of course, much more expensive alloys might be used to push the metallurgical limit up to about 1100$^{o}$F -- however, material costs would be such that the unit could never be produced in any quantity.

Mr. G. L. Lindsay of the Skinner Engine Company advocates the use of a pure mineral oil refined especially for cylinder lubrication. This oil is usually fed through the steam and lubrication is successful at temperatures as high as 740$^{o}$F. However, some carbon will form on the valves since these lubricants disintegrate quite rapidly above 650$^{o}$F. Along the same lines, Mr. R. B. Purdy of the Socony Mobil Oil Company, Inc., has informed us that they are lubricating many steam-engine cylinders operating with steam at 652 to 700$^{o}$F. Since carbon deposits are detrimental to correct valve sealing, and the products of oil deterioration foul up the boiler feed water, we are designing our engine for a conservative steam inlet temperature of 660$^{o}$F.

We originally planned to use two identical double-acting uniflow cylinders. Steam with pressures varying from 800 psia to 1500 psia and superheated to 660$^{o}$F was considered. Because minimum practical clearance for a small engine is about 5 % due to the ratio of port volume to small displacement, and minimum recommended cut-offs are 10%, the engine would exhaust considerably above atmospheric pressure. Due to this limited expansion, steam flow rates of from 25 to 35 lbm/hp hr would be encountered. Since this is much too high, we decided to compound the engine by using two high pressure cylinders as before and attaching the connecting rod of each to a larger double-acting low pressure cylinder, making four cylinders in all with only two "crank throws".

Though this arrangement would increase the number of expansions, the smaller cylinders would necessitate clearances of about 10%. Cut-offs for the low pressure cylinders are necessarily later than for the high pressure cylinders, depending upon the ratio of cylinder areas. This is due to the fact that the mass of steam exhausted from the high pressure cylinder must be enclosed in the low pressure cylinder at cut-off. Since the maximum practical value for the cylinder area ratio is 8:1, the low pressure cylinder must have a cut-off of at least 12-1/2% with the high pressure cylinders exhausting directly into the low pressure cylinders. Piston strokes are of course equal for both. It is advisable to use receivers of from 1 to 5 times the volume of the high pressure cylinder before introduction of the steam into the low pressure cylinder, and this would necessitate slightly larger low cylinder cut-offs due to expansion before cut-off. As before, the total number of expansions was still unsatisfactory. In view of these last two alternatives, our proposed design seems most feasible at present.

## B. CALCULATION

Starting with an engine inlet temperature $T_1 = 660^oF$, and choosing a boiler pressure $P_B = 1100$ psia, approximately equal to the engine pressure cut-off $P_1$, we get a value of $h_1 = 1288.5$ Btu/lb for initial enthalpy and a corresponding specific volume $v_1 = 0.5110$ ft$^3$/lb. Using a high pressure cylinder clearance of 5% and a normal cut-off of 10%, the ratio of cut-off volume, $v_1$ to 90% expansion stroke specific volume $v_{1R}$ (specific volume at release) is:

$$\frac{V_1}{V_{1R}} = \frac{15}{95}\ ; \quad \therefore \ \frac{95 \times 0.5110}{15} = 3.2350 \ ft^3/lb \ .$$

Using the gas formula $PV^N = K$, and assuming N to be 1.1 for such an expansion, we find $P_{1R}$.

$$P_{1R} = \frac{P_1}{\left(\frac{V_{1R}}{V_1}\right)^{1.1}} = 145 \ psia \ .$$

$h_{1R}$ is then found to be equal to 1208 Btu/lb (Steam Tables). Mass rate of flow through the cylinder $w_1 = 2545/\Delta h = 2545/80.5 = 31.6$ lb/hp hr.

It should be remembered that this same steam flow will do additional work in the low pressure cylinder and turbine, and hence the flow rate for the system will be much less.

Assuming the specific volume of the steam in the receiver, $v_2$, to be that of the cylinder for 100% expansion stroke,

$$V_2 = \frac{105 \times V_1}{15} = 3.580 \ ft^3/lb \ ,$$

$$P_2 = \frac{P_1}{\left(\frac{V_2}{V_1}\right)^{1.1}} = 130 \ psia \ ,$$

and $$h_{2R} = 1123.5 \ Btu/lb \ .$$

This corresponds to about $20^{\circ}$ superheat which is desirable to avoid initial condensation in the low-pressure cylinder!

Assuming a 4% clearance and 12-1/2% cut-off in the low-pressure cylinder,

$$v_{2R} = \frac{94 \times v_2}{16.5} = 20.4 \ ft^3/lb \ ,$$

$$P_{2R} = \frac{P_2}{\left(\frac{V_{2R}}{V_2}\right)^{1.1}} = 19.2 \ psia \ ,$$

and $$h_{2R} = 1123.5 \quad Btu/lb \ .$$

The mass rate of flow through this cylinder $w_2$ = 31 lb/hp hr, which is approximately that of the high pressure cylinder. This is necessary to insure approximately equal horsepower from the two cylinders.

The turbine inlet pressure $P_3$ will be several pounds less than that of the steam leaving the low pressure cylinder at 100% expansion stroke, which is 17.2 psia. Since v at this point is 22.5 ft$^3$/lb, h = 1118.56 Btu/lb. Assuming an isentropic expansion to $P_3$ = 15 psia ($s_3$ = 1.6931), $h_3$ = 1109 Btu/lb. Using a turbine efficiency

$$\eta_t = \frac{h_3 - h_{3R}}{h_3 - h_{3RS}} = 0.70 \ ,$$

where $(h_3 - h_{3RS})$ is the isentropic enthalpy drop and $(h_3 - h_{3R})$ is the real enthalpy drop, the real enthalpy drop to the turbine exhaust pressure $P_{3R}$ can be calculated. Assuming $P_{3R}$ = 7 psia, $h_{3RS}$ = 1058 Btu/lb,

$$h_{3R} = h_3 - \eta_t \left(h_3 - h_{3RS}\right) = 1109 - .7\left(1109 - 1058\right)$$

$$= 1073.8 \ Btu/lb \ ,$$

and $$h_3 - h_{3R} = 35.7 \ Btu/lb \ .$$

The moisture content is about 6-1/2% which is below the 10% maximum allowable to prevent rapid turbine blade erosion. Mass rate of flow for the turbine $w_3$ = 71.4 lb/hp hr.

It now becomes necessary to specify the intended horsepower rating of the high pressure cylinder. Since cut-offs were chosen for "normal operating conditions", we will define this as steady, level, driving at speeds up to 80 mph. For such driving, we will design the engine for a maximum of approximately one hundred indicated horsepower. Distributing this load between the two cylinders, the high pressure cylinder must be capable of supplying 50 horsepower. The total flow, w, can now be calculated for the system.

$$W = w_1 (hp)_{hpc} = 31.6 \, {}^{lb}/_{hp\text{-}hr} \times 50 \, hp = 1580 \, \frac{lb}{hr}$$

The horsepower of the low pressure cylinder is

$$(hp)_{lpc} = \frac{W}{w_2} = 51 \, hp ,$$

and the horsepower of the turbine is

$$(hp)_t = \frac{W}{w_3} = 22.15 \, hp.$$

The mass rate of flow for the system, w, is simply the total flow w divided by the total horsepower

$$[ (hp)_{hpc} + (hp)_{lpc} + (hp)_t ] .$$

$$w = \frac{W}{(hp)_{total}} = 12.8\ \text{lb}/\text{hp-hr} .$$

Besides the indicated horsepower of the cylinder, the mean effective pressure $P_m$ must also be known before the dimensions of the cylinder can be determined from the following equation:

$$ihp = \frac{P_m LAN}{33,000}$$ for a single acting cylinder

$$ihp = \frac{2P_m LAN}{33,000}$$ for a double acting cylinder

such as we are designing.

$P_m$ = mean effective pressure, $lb/in^2$

L = stroke of piston, ft

A = area of piston, $in^2$

N = engine speed, R.P.M.

ihp = indicated horsepower.

If D is the cylinder diameter, then $A = \pi D^2/4$. A good design will have this cylinder diameter, D, .70 to .85 that of the stroke in inches for high pressures. Using .70 for a small cylinder, D = .70L x 12 = 8.4 L in. Therefore,

$$A = \frac{\pi D^2}{4} = \frac{(8.4L)^2\, 3.14}{4} = 55.4\, L^2\, in^2 .$$

Maximum piston speed should range from 600 ft/min to 1000 ft/min, approaching the upper limit for a small engine. This piston speed is simply twice the product of the stroke, L, and engine speed N. Since the larger, low pressure cylinder will have a longer stroke, a conservative 500 ft/min maximum piston speed will be specified for the high pressure cylinder. Therefore, N = 250/L RPM.

For our double-acting engine, the horsepower equation governing the high pressure cylinder becomes:

$$(ihp)_{hpc} = \frac{2 P_m L\, 55.4\, L^2\, 250}{33{,}000\, L} = .894\, P_m\, L^2$$

Once the mean effective pressure has been determined for operating conditions such as those calculated previously, the stroke L can be found. Knowing L, the parameters A and N are easily found.

For the low pressure cylinder, we shall choose

$$D_2 = .85\, L_2\, ft = 10.2\, L_2\, in ,$$

and $$A_2 = \frac{\pi D_2^2}{4} = \frac{(10.2\, L_2)^2\, 3.14}{4} = 81.6\, L_2^2\, in^2 .$$

The number of revolutions, N, must be the same for both cylinders and was found to be 250/L RPM. Hence, for the low pressure cylinder,

$$(ihp)_{lpc} = \frac{2 P_m\, L_2\, 81.6\, L_2^2\, 250}{33{,}000\, L} = \frac{1.23\, P_{m_2}\, L_2^3}{L}$$

To determine the MEP (mean effective pressure) under various operating conditions, it is necessary to plot a series of theoretical indicator cards. The procedure is as follows:

1. <u>Compression</u>

If $P_2$ is condenser or receiver pressure, $P_c$ is compression end pressure equal to about 80 or 85% of the boiler pressure $P_1$, and C is the clearance as a percentage of the volume, then for a 90% uniflow stroke compression, using the gas formula $PV^N = K$,

$P_2 (90 + C)^N = P_c (C)^N$ where $N$

is equal to 1.2 to 1.25.

Intermediate pressures $P_x$ can then be calculated by substituting values in the equation:

$$P_x (x + C)^N = P_c (C)^N$$

A plot of $P_x$ versus stroke then becomes the basic compression curve of the indicator diagram.

2. <u>Expansion</u>

The same procedure above is used to calculate the expansion curve, but, expecting a large amount of re-evaporation in a small engine, N will be assumed between 1.05 and 1.15.

Using the equation $P_R (90 + C_1)^N = P_1 (C_1)^N$, the expansion curve can be calculated from the release pressure to boiler or cut-off pressure by solving for intermediate pressures $P_x$ in

$$P_R\,(90 + C_1)^N = P_x\,(x + C_1)^N \quad \text{where } C_1$$

is clearance plus cut-off.

The upper end should be adjusted for valve wire drawing (resulting in a cut-off pressure somewhat below boiler pressure) and the 90% release point may be connected to the condenser pressure at 100% by a straight line. The MEP can then be measured with a planimeter. The following calculations will illustrate this procedure.

HIGH PRESSURE CYLINDER

$P_1$ = 1100 psia

$T_1$ = 660$^{\circ}$F

Normal Cut-off = 10%

C = 5%

FOR COMPRESSION

$P_c$ = .85 $P_1$ = 935 psia

N = 1.225

Therefore, to obtain the basic compression curve, we plot $P_{x_c}$ vs. x from the equation

$$P_{x_c}\,(x + C)^N = P_c\,(C)^N, \quad \text{or} \quad P_{x_c} = 935\left(\frac{5}{x+5}\right)^{1.225},$$

and obtain the compression curve AK', shown in Figure 15. Since $P_2$ was determined previously from the thermodynamic analysis, $P_2 = 130$ psia, compression in the engine indicator diagram will correspond to AK of Figure 15. Since this corresponds to a value of 20% for x, it is evident that auxiliary exhaust valves will be necessary to delay compression until about 80% of the compression stroke. This design would be similar to that of Figure 12.

FOR EXPANSION

$P_{1R}$ = 145 psia from thermodynamic analysis

$C_1$ = 15%

N = 1.10

and the equation

$$P_R (90 + C_1)^N = P_{x_E} (x + C_1)^N$$

takes the form

$$P_{x_E} = 145 \left( \frac{105}{x + 15} \right)^{1.1} .$$

A plot of $P_{x_E}$ vs. x corresponds to CO-R of the indicator diagram shown in Figure 15. The upper end has been adjusted for wire drawing as compared to the theoretical plot in dotted lines. Straight lines between x = 20, P = 130, and x = 100, P = 130, and x = 90, P = 145, and x = 100, P = 130 complete the normal indicator card for the high pressure

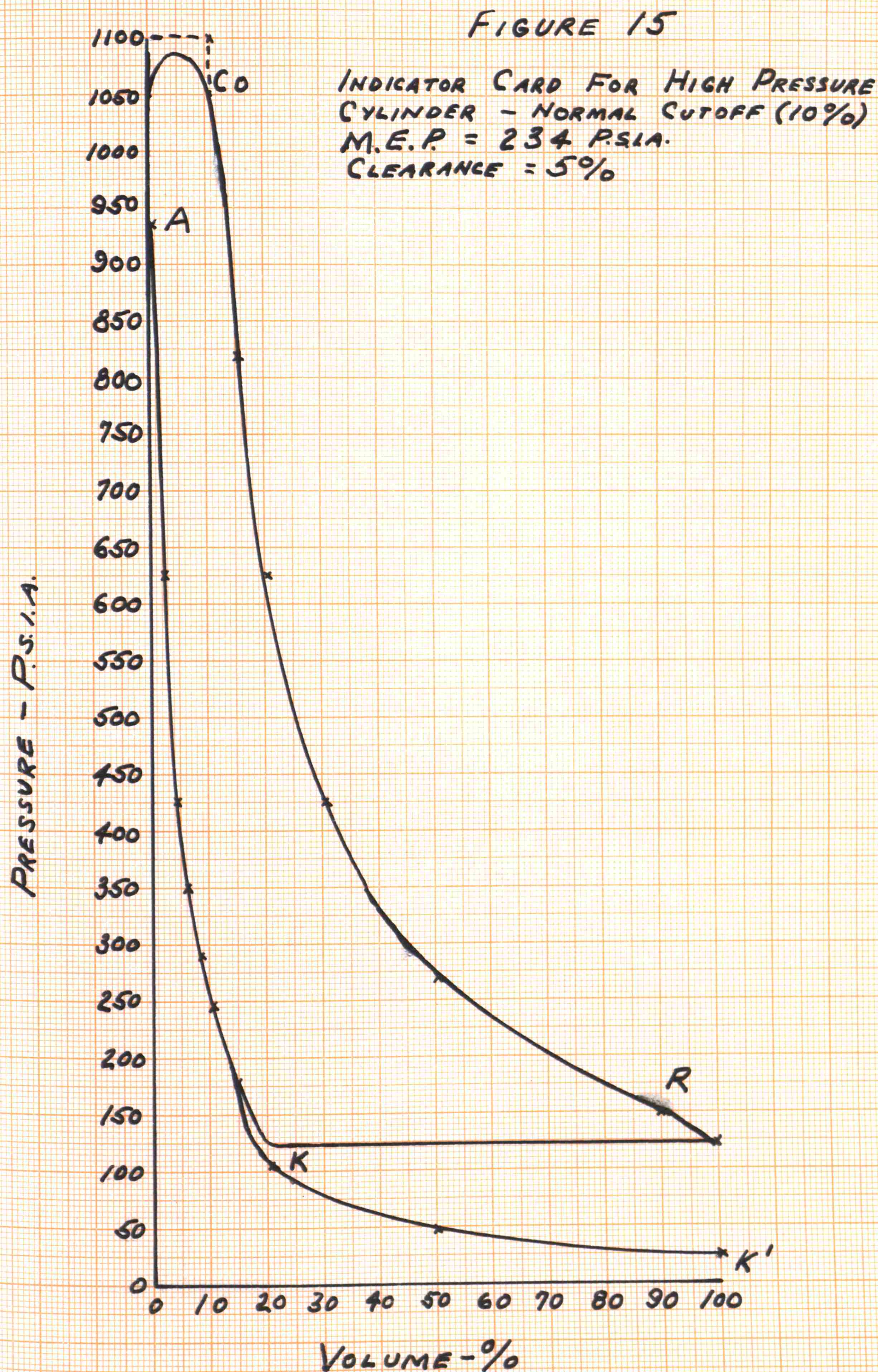


FIGURE 15
INDICATOR CARD FOR HIGH PRESSURE CYLINDER - NORMAL CUTOFF (10%)
M.E.P. = 234 P.S.I.A.
CLEARANCE = 5%
Co
A
R
K
K'
0
50
100
150
200
250
300
350
400
450
500
550
600
650
700
750
800
850
900
950
1000
1050
1100
10
20
30
40
50
60
70
80
90
100
PRESSURE - P.S.I.A.
VOLUME - %

cylinder. The MEP of this plot is equal to 234 psia. ($P_m$ = 234 psia). Remembering that $(ihp)_{hpc} = .894 \, P_m \, L^2$ and, designing for a 50 hp cylinder,

$$L = \sqrt{\frac{50}{.894 \times 234}} = .488 \text{ ft } (5.86 \text{ in})$$

$$D_1 = .70L \times 12 = 4.10 \text{ in.}$$

$$A_1 = 55.4 \, L^2 = 13.20 \text{ in}^2$$

$$N = 250/L = 512 \text{ RPM}$$

The theoretical indicator diagram of the low pressure cylinder, Figure 16, was obtained in the same manner as that for the high pressure cylinder. The MEP was found, $P_{m2}$ = 30.6 psia.

Since $(ihp)_{lpc} = \dfrac{1.48 \, P_{m_2} \, L_2^3}{L}$, and $ihp = 51$,

$$L_2 = \sqrt[3]{\frac{.488 \times 51}{1.48 \times 30.6}} = .820 \text{ ft}$$

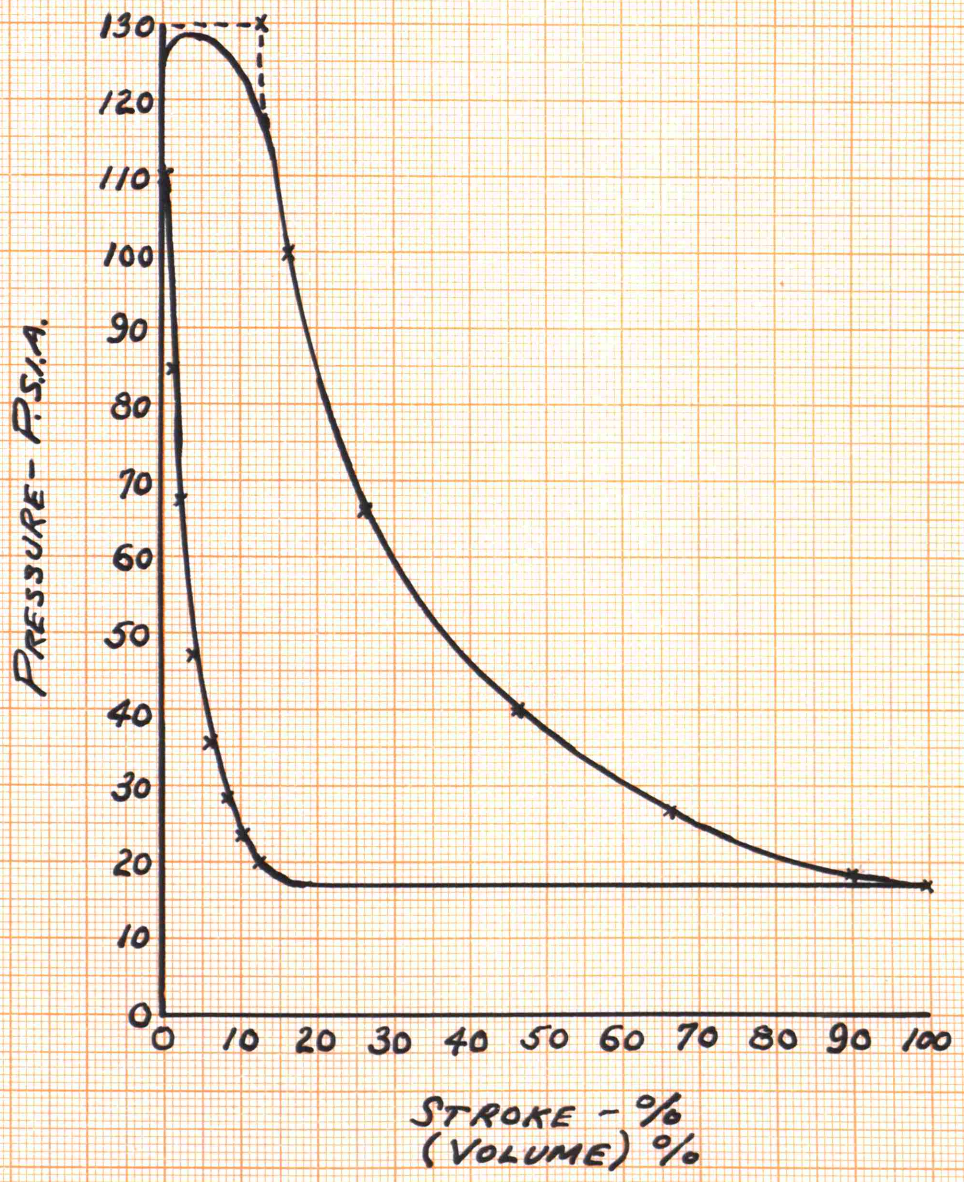
$$D_2 = .85 \, L_2 = 8.36 \text{ in}$$

$$A_2 = 81.6 \, L_2^2 = 54.9 \text{ in}^2$$

$$\text{PISTON SPEED} = 2NL_2 = 840 \text{ ft/min}$$

FIGURE 16

INDICATOR DIAGRAM FOR LOW PRESSURE
CYLINDER — CUTOFF = 12 1/2 %
M.E.P. = 30.6 P.S.I.A
CLEARANCE = 4 %



For accelerating and climbing hills or whenever overload conditions prevail, a longer high pressure cylinder cut-off is necessary. Indicator diagrams have been drawn increasing the cut-off of the high pressure cylinder to 25% and 50% and thereby increasing the MEP to 351 psia and 382 psia. In order to keep the horsepower output of both cylinders equal, the low pressure cylinder cut-off must be shortened. Steam must then be extracted from the receiver for regenerative feedwater heating to avoid excess backpressure to the high pressure cylinder. A cut-off of 50% will increase the output of the engine to about 160 horsepower, which should suffice for a small car! For speeds above 80 mph, a two-speed differential would be necessary to keep the piston speed below its maximum. With a longer cut-off, this differential should make speeds well over 100 mph possible. At this point, it does not at all appear too optimistic that such a steam sports car should outperform present American models, offering at the same time at least as good operation economy.

# THE CONDENSER

It is desired to find a condenser which can handle all exhaust steam under normal operating conditions in the worst possible ambient environment, plus some overload capacity for short periods of acceleration. This problem is first attacked in general for any steam car application. A particular configuration is then chosen to be used as one of the basic components in the sports car cycle.

## A. NOMENCLATURE

Subscripts:

a - air side

s - steam side

w - wall

1 - exchanger entrance

2 - exchanger exit

$A$ Exchanger total heat transfer area on one side, $ft^2$

$A_c$ Exchanger minimum free-flow area, $ft^2$

$A_f$ Exchanger total fin area on one side, $ft^2$

$A_{fr}$ Exchanger total frontal area, $ft^2$

$a$ Plate thickness, ft.

$b$ Plate spacing, ft

$c_p$ Specific heat at constant pressure, $Btu/lbm^oF$

$f$ Mean friction factor, dimensionless

$G$ Exchanger flow stream mass velocity; $G = (w/A_c)$, lbm/sec - ft$^2$

$g_c$ Proportionality factor; $g_c = 32.2$ (lbm/lbf) (ft/sec$^2$)

$h$ Heat transfer coefficient, Btu/hr - ft$^2$ $^oF$

$h_{fg}$ Latent heat of vaporization, Btu/lbm

$j$ A product of dimensionless heat transfer groups; $j = (h/c_p G)(c_p \mu/k)^{2/3}$

$K_c$ Loss coefficient for flow at heat exchanger entrance, dimensionless

$K_e$ Loss coefficient for flow at heat exchanger exit, dimensionless

$k$ Thermal conductivity, Btu/hr ft $^oF$

$L$ Total exchanger length, ft

$l$ Fin length from root to center; $l = b/2$, ft

$m$ $\sqrt{2h/k\delta}$ , ft$^{-1}$

$N_{Re}$ Reynolds number; $N_{Re} = G\,(4r_h/\mu)$, dimensionless

$N_{Pr}$ Prandtl number; $N_{Pr} = C_p \mu/k$, dimensionless

$P$ Pressure, lbf/ft

$Q$ Volume flow rate, ft$^3$/min

$q$ Heat transfer rate, Btu/hr

$r_h$ Hydraulic radius; $r_h = A_c L/A$, ft

$t$ Temperature, degrees Fahrenheit, $^oF$

$\Delta t_{lm}$ Logarithmic-mean temperature difference, $^oF$

$U$ Overall coefficient of heat transfer, Btu/hr ft$^2$ $^oF$

$V_{tot}$ Total volume, ft$^3$

v Specific volume, $ft^3/lbm$

w Mass flow rate, lbm/sec

x Thickness of wall, ft

$\alpha$ Ratio of total transfer area on one side of the exchanger to total volume of the exchanger, $ft^2/ft^3$

$\beta$ Ratio of the total heat transfer area on one side of a plate-fin heat exchanger to the volume between the plates on that side, $ft^2/ft^3$

$\Delta$ Denotes difference

$\delta$ Fin thickness, ft

$\epsilon$ Heat transfer effectiveness of an exchanger, dimensionless

$\eta_f$ Fin temperature efficiency, dimensionless

$\eta_o$ Total surface temperature efficiency, dimensionless

$\sigma$ Ratio of free-flow area to frontal area, $A_c/A_{fr}$, dimensionless

$\mu$ Viscosity, lbm/hr ft

$\rho$ Density, $lbm/ft^3$

$\Omega$ Number of fins per unit length, fins/ft

## B. SELECTION OF A SUITABLE CONDENSER

### Problem

The problem under consideration is to find an air-to-steam condenser which has the following specifications:

1. The steam side pressure, $P_s$, equals 5 psia.
2. The frontal area, $A_{fr}$, is not to exceed 8 $ft^2$.
3. In designing for the worst possible operating conditions, the temperature of the ambient air, $t_a$, is chosen as $100^oF$.
4. The fan to be used in conjunction with the condenser should not exceed 5 hp in size.
5. The condenser is to be capable of cooling approximately 2000 pounds of steam an hour.

In attacking this problem extensive use was made of the data collected by W. M. Kays and A. L. London[1] in their studies conducted at Stanford University on sixty-five test surfaces. This information was used to find a suitable configuration for the condenser.

### Procedure

The steam-to-air compact exchanger is designed to take the form of a "plate-fin" exchanger (See Figure 17). The optimum heat

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[1] Kays, W. M., and A. L. London, COMPACT HEAT EXCHANGERS, Palo Alto, California: The National Press, 1955.

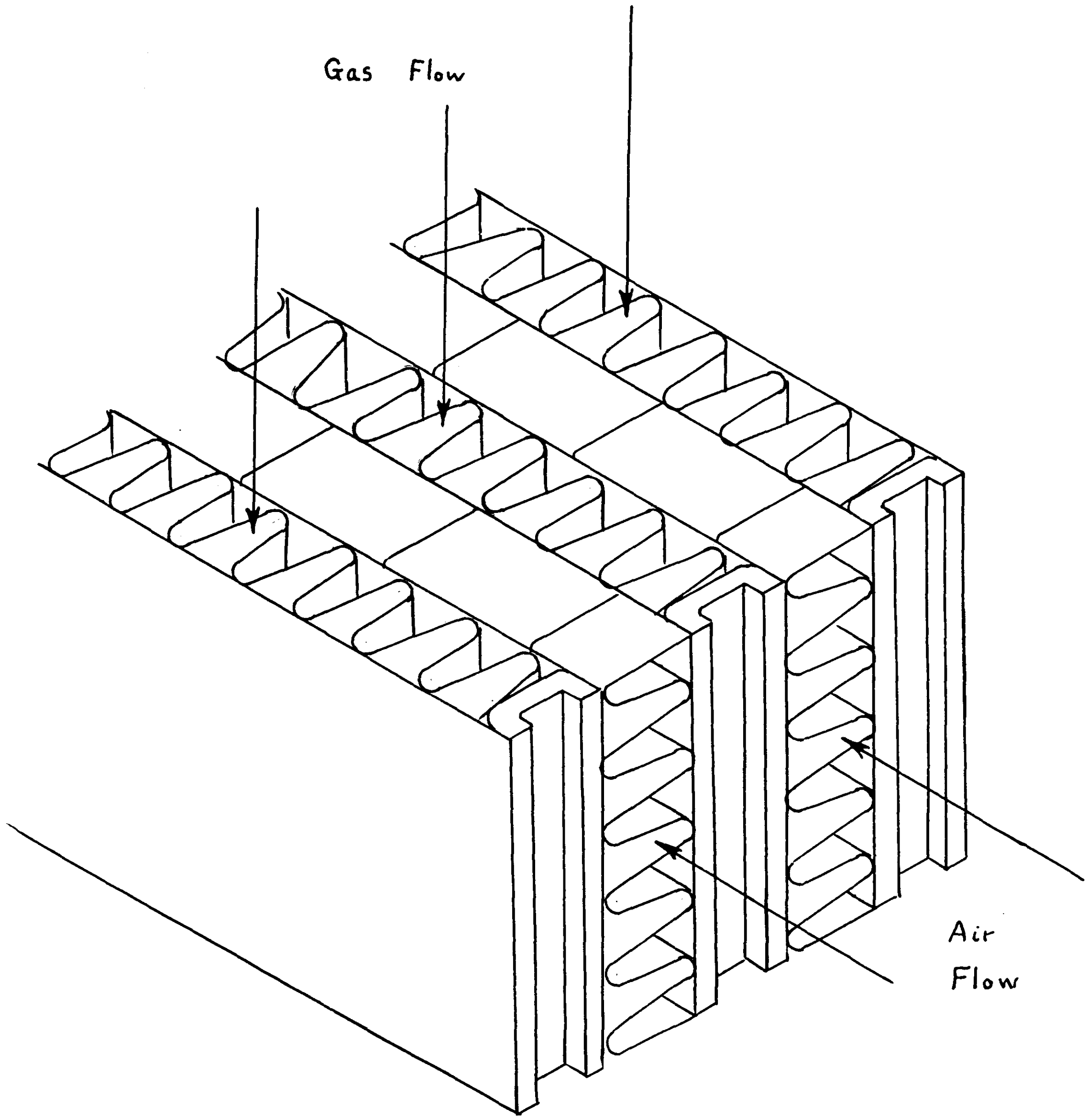



FIGURE 17

COMPACT PLATE - FIN HEAT EXCHANGER

transfer area ratio, as derived by L. C. Hoagland and J. P. Barger at M.I.T.[2], is given by

$$\frac{A_a}{A_s} = \sqrt{\frac{h_s \eta_{os}}{h_a \eta_{oa}}} \qquad (1)$$

Using approximate values for the quantities under the square root sign, the area ratio equals 6.6. This indicates that the air side fins on the exchanger are to be much larger than those on the steam side. Thus a plain fin configuration is chosen for the steam side, with a small plate spacing decided upon (See Figure 18a).

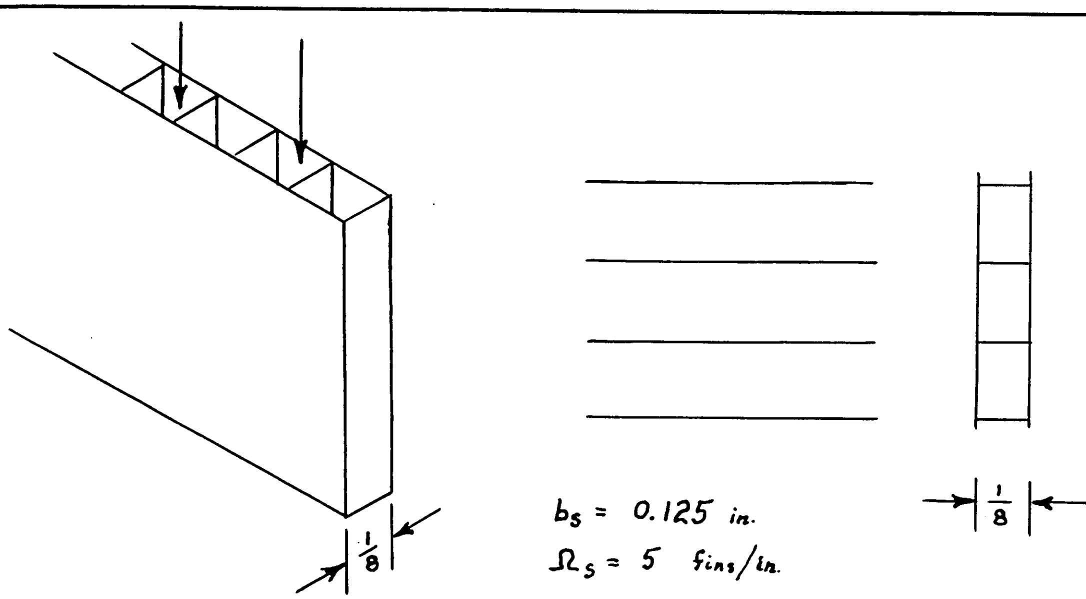
The air side configuration is chosen to be surface 17.8 - 3/8w in Kays and London (See Figure 18b). This choice is made on the basis of a large plate spacing (b = 0.413 in.) and a high heat transfer area-to-volume between plates ratio ( $\beta$ = 514 $ft^2/ft^3$)

From geometrical considerations

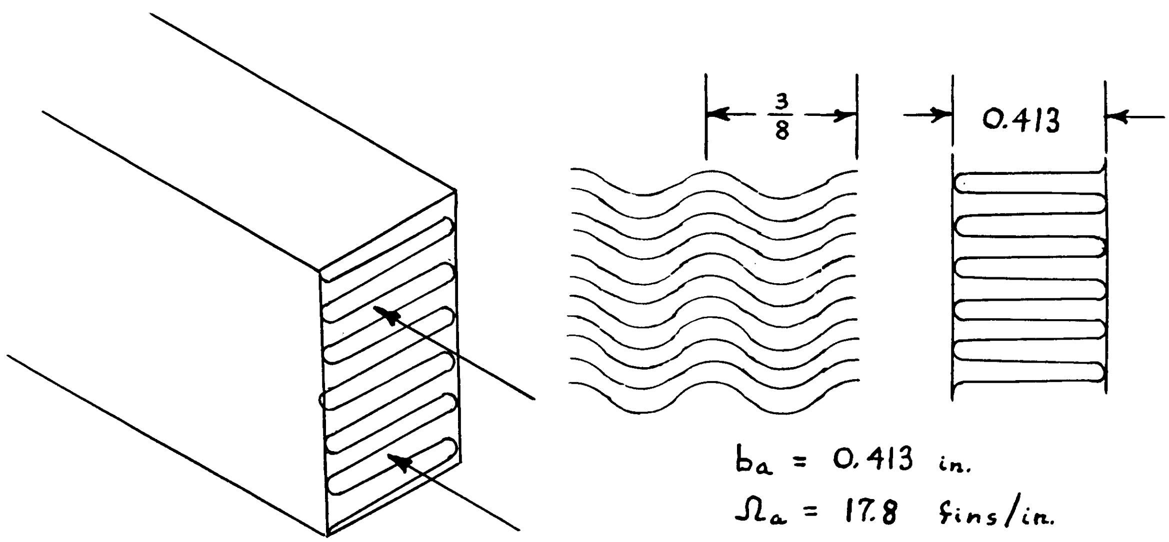
$$\frac{A_a}{A_s} = \frac{1 - \Omega_a \delta + \Omega_a (b_a - \delta)}{1 - \Omega_s \delta + \Omega_s (b_s - \delta)}$$

Choosing $b_s$ = 0.125 inches (using one-eighth of an inch as an arbitrary lower limit), and solving for $\Omega_s$, it is found that approximately 2 fins/inch are required on the steam side to obtain optimum heat transfer. However, strength criteria dictate that more

[2] See Notes in Appendix




(a) STEAM SIDE CONFIGURATION




$b_a = 0.413$ in.
$\Omega_a = 17.8$ fins/in.
$4 r_h = 0.00696$ ft.
$\delta = 0.006$ in.
$\beta = 514$ ft$^2$/ft$^3$
$A_f / A_a = 0.892$

(b) AIR SIDE CONFIGURATION

FIGURE 18

fins are needed. Hence, $\Omega_s$ is taken to be about 5 fins/inch.

The heat-transfer requirements of the exchanger can be calculated from

$$q = w_s h_{fg} \tag{2}$$

where $h_{fg}$ is approximately equal to 1000 Btu/lbm, and $w_s$ is specified at 2000 lbm/hr. Hence, $q = 2 \times 10^6$ BTU/hr.

An exchanger effectivenss is chosen at $\epsilon = 0.75$.

$$\epsilon = \frac{t_{a2} - t_{a1}}{t_s - t_{a2}} \tag{3}$$

where $t_s$ is the saturation temperature at 5 psia, which equals 162.24°F. The ambient air temperature, $t_{a1}$, is given as 100°F. Solving equation (3) for $ta_2$ gives $ta_2$ = 146.7°F.

The heat-transfer rate can also be determined from air side characteristics by means of the equation $q = U_a A_a \Delta t_{lm}$ where $\Delta t_{lm}$, the logarithmic-mean temperature difference, is determined from

$$\Delta t_{lm} = \frac{(t_s - t_{a2}) - (t_s - t_{a1})}{ln\left(\frac{t_s - t_{a2}}{t_s - t_{a1}}\right)} = 33.5\ {}^\circ F$$

Therefore, $U_a A_a$ for this exchanger must equal

$$\frac{2 \times 10^6}{33.5} \approx 6 \times 10^4 \quad \text{BTU/hr-}{}^\circ F \tag{4}$$

$U_a$ is determined by

$$\frac{1}{U_a} = \frac{1}{h_a \eta_{oa}} + \frac{1}{h_s \eta_{os}} \frac{A_a}{A_s} + \frac{x}{k} \frac{A_a}{A_w} \qquad (5)$$

However, x is of the order of 0.01 inch, and k = 100 Btu/hr ft $^oF$ for aluminum (this material is used for its light weight and ability to be molded easily into various fin shapes), making the third term in equation (5) negligible. Hence

$$\frac{1}{V_a} = \frac{1}{h_a \eta_{oa}} + \frac{1}{h_s \eta_{os}} \frac{A_a}{A_s} \qquad (5a)$$

It is also useful to note that the second term is only 5-10% as large as the first term. This means that it can be assumed constant for all practical purposes. Reasonable values for the quantities involved are $h_s$ = 2000 Btu/hr ft$^2$ $^oF$, and $\eta_{os}$ = 0.99. For the geometry under consideration, $A_s/A_a$ = 0.192.

The data for a given surface is presented in Kays and London in the form of a graph. They plot j and f versus $N_{re}$, where j = $\left(\frac{h}{G c_p}\right)\left(N_{Pr}\right)^{2/3}$ and $N_{Re} = \frac{4 r_h G}{\mu}$

Calculations are made by selecting a value of G and then determining the other characteristics of the system. Table 1 shows these characteristics for G = 5,4,3,2, and 1 lbm/sec ft$^2$. A sample calculation for G = 1 lbm/sec ft$^2$ is given in the Appendix.

After these characteristics are determined, the size of the exchanger is found by specifying the air flow rate through the condenser. However, this flow must be supplied by a fan which uses less than 5 hp. The fan specifications for a typical 20 inch diameter fan are given in Table 2.

$A_{fr}$ is given by

$$A_{fr} = \frac{A_c}{\sigma_a} = \frac{w_a}{\sigma_a G} \tag{6}$$

where $$\sigma_a = \frac{b_a \beta_a r_h}{b_a + b_s + 2a} = 0.655$$

for this geometry. Noting that $A_{fr} \leq 8 \ ft^2$ (problem design point), $w_a$ must be $\leq$ (8) (0.655) G = 5.44 G lbm/sec.

Also, the pressure drop in the exchanger must be less than that given in Table 2 at a given $w_a$ if the exchanger is to operate at all. This $\Delta p$ is calculated from the equation

$$\Delta p = \frac{G^2}{2 g_c \rho} \left[ (K_c + 1) + f \frac{A}{A_c} - (1 - K_e) \right]^{***} \tag{7}$$

where $K_c$ and $K_e$ are plotted versus $\sigma$ for various $N_{re}$ in Kays and London.

Table 3 shows $\Delta p$ and $A_{fr}$ for various G. It is interesting to note that varying $w_a$ at constant G has little effect on $\Delta p$, but determines $A_{fr}$ by means of equation (6).

Comparing $\Delta p$ in Table (3) and Table (2), and remembering that $w \lessapprox 5.44\ G$ (lbm/sec), it is easy to see that G = 1 lbm/sec $ft^2$ is the best choice. Therefore, $w \lessapprox 5.44 \times 1 = 5.44$ lbm/sec.

Taking w = 5 lbm/sec, $\Delta p_{fan} = 4.7$ in. $H_2O$

$\Delta p$ condenser = 1.28 in $H_2O$

$$A_{fr} = \frac{5}{0.655 \times 1} = 7.63 \quad ft^2$$

$$L = \frac{V_{tot}}{A_{fr}} = \frac{8.96}{7.63} = 1.175 \quad ft$$

The condenser found to match the engine in this sports car has approximately 8 $ft^2$ frontal area and is about 1-1/4 ft. deep. Calculations for this case are given in Appendix B. As is shown in the layout drawing, the condenser is located in front of the car. It condenses fully only under normal operating cut-off. For overload conditions some receiver steam is "bled" off in a regenerative feedwater heating system. Hence no water is lost outside the cycle.

The fan used with the condenser is rated at about 5 hp and handles a pressure drop of 1.04 in $H_2O$. This fan is capable of circulating 4000 cubic feet of air per minute. It is operated directly off the turbine shaft.

*** The more general equation, $\Delta p = \frac{G^2 v_1}{2 g_c} \left[ (K_c + 1) + \sigma \left(\frac{v_2}{v_1} - 1\right) + f \frac{A}{A_c} \frac{v_m}{v_1} - (1 - \sigma - K_e) \frac{v_2}{v_1} \right]$ has been simplified by the assumption that the specific volumes at entrance and exit of the exchanger are equal, or $v_2 = v_1 = v_m$.

TABLE 1

| $G$ | $N_{Re}$ $10^{-3}$ | $j$ | $f$ | $h$ | $\eta_{o\,a}$ | $U_a$ | $V_{tot}$ |
|---|---|---|---|---|---|---|---|
| (lbm/sec-ft$^2$) | (-) | (-) | (-) | (BTU/hr ft$^2$ $^oF$) | (-) | (BTU/hr-ft$^2$-$^oF$) | (ft$^3$) |
| 5 | 2.62 | 0.009 | 0.038 | 49.4 | 0.869 | 39.3 | 4.10 |
| 4 | 2.09 | 0.0097 | 0.042 | 42.5 | 0.871 | 33.8 | 4.70 |
| 3 | 1.57 | 0.011 | 0.045 | 36.1 | 0.878 | 29.3 | 5.36 |
| 2 | 1.045 | 0.013 | 0.057 | 28.4 | 0.908 | 24.2 | 6.56 |
| 1 | 0.523 | 0.017 | 0.079 | 18.6 | 0.938 | 17.7 | 8.96 |

$$N_{Re} = \frac{G\,(4r_h)}{\mu}$$

$4r_h = 0.00696$ ft

$\mu = 0.048$ lbm/hr-ft at $133.5^oF$

$c_p = 0.2405$ BTU/lbm $^oF$ at $133.5^oF$

$N_{p_r} = 0.70$ at $133.5^oF$

$$V_{tot} = A_a/\alpha_a$$

$$A_a = 3 \quad 10^4/V_a$$

$$\alpha_a = b_a \;\beta_a/b_a + b_s + 2a$$

$$\eta_{o\,a} = 1 - \left(\frac{A_f}{A}\right)(1 - \eta_f)$$

$$\frac{A_f}{A} = 0.892$$

$$\eta_f = \frac{\tan h\ ml}{ml}, \quad l = \frac{b}{z}, \quad m = \sqrt{\frac{2\,h}{k\,\delta}}$$

$\delta = 0.006$ in

$k = 100$ BTU/hr ft $^oF$

Assume: $a = 0.012$ in

$\beta_a = 514$ ft$^2$/ft$^3$

$b_a = 0.413$ in

$b_s = 0.125$ in

TABLE 2

(Data taken from graph on page 1917 in Marks' Handbook)

Typical Twenty-Inch Diameter Fan

| $w_a$ (lbm/sec) | Volume (cfm) | Total Pressure (in $H_2O$) | Horsepower (hp) | Total Efficiency (%) |
|---|---|---|---|---|
| 0 | 0 | 8.5 | 7.8 | 0.00 |
| 1.25 | 1000 | 8.1 | 7.4 | 17 |
| 2.5 | 2000 | 6.8 | 6.3 | 33 |
| 3.75 | 3000 | 5.3 | 5.0 | 48 |
| 5.0 | 4000 | 4.7 | 4.4 | 69 |
| 6.25 | 5000 | 4.3 | 4.2 | 82 |
| 7.50 | 6000 | 3.1 | 3.2 | 87 |
| 8.75 | 7000 | 1.5 | 2.0 | 70 |

TABLE 3

| $G$ ($lbm/sec\text{-}ft^2$) | $K_c$ (-) | $K_e$ (-) | $A_{fr}$ ($ft^2$) | $\Delta p$ (in $H_2O$) |
|---|---|---|---|---|
| 4 | 0.43 | -0.01 | 1.91 | 37.9 |
| 3 | 1.09 | -0.44 | 2.55 | 19.3 |
| 2 | 1.09 | -0.44 | 3.82 | 9.1 |
| 1 | 1.09 | -0.44 | 7.63 | 1.04 |

$w_a$ = 5 lbm/sec

$Q$ = 4,000 cfm

## CONCLUSIONS

This paper, involving the design of a practical steam sports car, is quite encouraging. We feel that solutions offered for the boiler and condenser designs are particularly satisfactory. The condenser, our original design, must of course be tested to determine its actual performance. Since performance specifications were calculated somewhat conservatively, however, the condenser should live up to expectations.

Flash boilers have been built by Charles Keen of Madison, Wisconsin and William J. Besler of Oakland, California, for their private steam cars. Since these boilers are capable of supplying the steam required by our engine, we considered this component to be adequately available for our design.

Because of the many design variables peculiar to a small high pressure, high temperature steam engine, the thermodynamic calculations obtained concerning the engine are somewhat uncertain. However, since existing data is based largely on moderate pressure and temperature conditions and conventional sizes, our approach was still the most feasible. Several of the unknowns encountered were:

1. valve throttling losses due to high speed
2. larger clearance, because of port areas vs. small displacement
3. high conductivity loss and radiation, due to small relative size

4. above normal exhaust temperatures and reevaporation of expanding steam, due to metal conductivity in a small engine.

Out of necessity then, the engine would have to be built and tested to determine its true performance. In addition to determining the above unknowns, the maximum number of expansions could be determined by determining the minimum practical cut-off. Shorter cut-offs would, of course, result in increased economy. Another promising possibility is the use of higher temperatures and pressures which the use of shorter cut-offs would allow. Some research with lubricants at temperatures above 700$^{\circ}$F in connection with this application is highly recommended.

One final aspect of lubrication remains. The cylinders of most steam engines operating at high temperatures are lubricated by means of force-feed lubricators that inject oil directly to the cylinders and valves. Since this lubricating oil is eventually carried out of the cylinder with the exhaust steam, it must be removed before the condensed steam can be returned to the boiler. This is normally done by using oil separators, which are simple in design and operation. However, in a closed system with a rapidly repeating cycle, the separation of the oil from the condensate should not be minimized and, unless such a problem is completely solved, rapid boiler failure would be encountered.

# APPENDIX A*

## OPTIMUM HEAT-TRANSFER AREA RATIO IN COMPACT HEAT EXCHANGERS

The ratio of heat transfer surface area on the two sides of a heat exchanger should be chosen in accordance with the ratio of film coefficients of heat transfer on the two sides. In the usual design problem the heat transfer rate and mean temperature difference are known, so that

$$\frac{q}{\Delta T_m} = UA = \text{constant}$$

where the U and A are based on one side of the heat exchanger. This quantity may also be expressed as

$$\frac{1}{UA} = \frac{1}{h_1 \eta_{o1} A_1} + \frac{1}{h_2 \eta_{o2} A_2} = \text{constant} \qquad (1)$$

This equation can now be used to obtain the heat transfer area ratio corresponding to a minimum total heat transfer area. Although specification of minimum total heat transfer area is not precisely identical with minimum weight, the two criteria are nearly coincident for plate fin heat exchangers with identical fin thicknesses on both sides. Only for the special case where all fin and plate thicknesses are equal will the minimum total area criterion be exactly coincident with minimum weight. However, since the ratio of fin area to total surface area usually is between 0.8 and 0.9, the minimum total area consideration

* These notes are reprinted here with the kind permission of L. C. Hoagland

should be adequate for the case of the same fin thickness on both sides.

Now, to obtain an expression for the heat transfer area ratio corresponding to minimum total heat transfer area, multiply equation (1) through by the total area $A_t$ and simplify to obtain

$$\frac{A_t}{UA} = \frac{1}{h_1 \eta_{o1}} + \frac{1}{h_2 \eta_{o2} \left(1 - \frac{A_1}{A_t}\right)}$$

Taking the partial derivative of $A_t/UA$ with respect to the ratio $A_1/A_t$ and equating to zero

$$\frac{\partial\left(\frac{A_t}{UA}\right)}{\partial\left(\frac{A_1}{A_t}\right)} = -\frac{1}{h_1 \eta_{o1} \left(\frac{A_1}{A_t}\right)^2} + \frac{1}{h_2 \eta_{o2} \left(1 - \frac{A_1}{A_t}\right)^2} = 0$$

$$\frac{h_1 \eta_{o1}}{h_2 \eta_{o2}} = \frac{1 - 2A_1/A_t + (A_1/A_t)^2}{(A_1/A_t)^2} = (A_t/A_1)^2 - 2(A_t/A_1) + 1$$

$$\left(\frac{A_t}{A_1}\right)^2 - 2\frac{A_t}{A_1} + \left(1 - \frac{h_1 \eta_{o1}}{h_2 \eta_{o2}}\right) = 0$$

Solving this quadratic for $A_t/A_1$ gives

$$\frac{A_t}{A_1} = 1 \pm \sqrt{\frac{h_1 \eta_{o1}}{h_2 \eta_{o2}}}$$

Because $A_t/A_1$ must necessarily be greater than or equal to unity, the plus sign must be used. Now $A_t = A_1 + A_2$ so that the above result reduced finally to

$$\frac{A_2}{A_1} = \sqrt{\frac{h_1 \eta_{o1}}{h_2 \eta_{o2}}} \qquad (2\ )$$

Since for most heat exchanger design problems the values of $h_1$, $h_2$, $_{o1}$, and $_{o2}$ do not vary significantly with the area ratio, they can be estimated for computing the optimum heat transfer area ratio from equation (2). Again it should be emphasized that this simple result is very nearly correct for plate fin heat exchangers having the same fin thickness on both sides, and is exactly correct when the plate thickness is equal to the fin thickness.

# APPENDIX B

## SAMPLE CONDENSER CALCULATION

GIVEN: $q = 2 \times 10^6$ BTU/hr

$U_a A_a \approx 6 \times 10^4$ BTU/hr-°F

$h_s = 2000$ BTU/hr-ft²-°F

$\eta_{os} = 0.99$

$\epsilon = 0.75$

$k_{aluminum} = 100$ BTU/hr-ft-°F

$b_a = 0.413$ in. $\qquad b_s = 0.125$ in.

$\Omega_a = 17.8$ fins/in. $\qquad \Omega_s = 5$ fins/in.

$4 r_h = 0.00696$ ft

$\beta_a = 514$ ft²/ft³ $\qquad a = 0.012$ in.

$A_f / A = 0.892$ $\qquad v_1 = v_2 = v_m$

FIND: $A_{fr}$ and $\Delta p$ of the steam-to-air exchanger at $G = 1$ lbm/sec-ft² and $w = 5$ lbm/sec

SOLUTION:

A. CALCULATE $N_{Re}$

$$N_{Re} = \frac{G(4r_h)}{\mu} = \frac{1 \times 0.00696 \times 3600}{0.048} = 0.523 \times 10^3$$

B. REFER TO PLOT OF j AND f VS. $N_{Re}$ IN KAYS AND LONDON TO OBTAIN

$j = 0.017$

$f = 0.079$

C. CALCULATE h FROM DEFINITION OF j

$$h_a = \frac{j G c_p}{N_{Pr}^{2/3}}$$

$c_p = 0.2405$ BTU/lbm-°F at 133.5 °F

$N_{Pr} = 0.70$ at 133.5 °F

$$h_a = \frac{0.017 \times 1 \times 0.2405 \times 3600}{(0.70)^{2/3}} = 18.6 \text{ BTU/hr-ft-°F}$$

D. CALCULATE $\eta_{oa}$

$$\eta_{oa} = 1 - \left(\frac{A_f}{A}\right)(1 - \eta_f)$$

$$\eta_f = \frac{\tanh ml}{ml}$$

$$l = \frac{b_a}{2} = \frac{0.413}{2 \times 12} = 0.0173 \text{ ft}$$

$$m = \sqrt{\frac{2h}{k\delta}} = \sqrt{\frac{2 \times 18.6 \times 12}{100 \times 0.006}} = 26.9 \text{ ft}^{-1}$$

$$m\ell = 0.466$$

$$\tanh m\ell = 0.434$$

$$\eta_f = \frac{0.434}{0.466} = 0.932$$

$$\eta_{oa} = 1 - (0.892)(1 - 0.932) = 0.938$$

E. CALCULATE $U_a$ FROM EQUATION (5a) WHERE

$$\frac{1}{U_a} = \frac{1}{h_a \eta_{oa}} + \frac{A_a/A_s}{h_s \eta_{os}} = \frac{1}{18.6 \times 0.938} + \frac{1/0.192}{2000 \times 0.99} = 0.0564$$

$$U_a = 17.7 \quad BTU/hr\text{-}ft^2\text{-}°F$$

F. FIND $V_{tot}$ FROM $A_a$ AND GEOMETRICAL CONSIDERATIONS

$$V_{tot} = \frac{A_a}{\alpha_a}$$

$$A_a = \frac{6 \times 10^4}{U_a} = \frac{6 \times 10^4}{17.7} = 3390 \ ft^2$$

$$\alpha_a = \frac{b_a \beta_a}{b_a + b_s + 2a} = \frac{0.413 \times 514}{0.413 + 0.125 + 2(0.012)} = 378 \ ft^2/ft^3$$

$$V_{tot} = \frac{3390}{378} = 8.96 \ ft^2$$

G. FIND $A_{fr}$ AT $W_a = 5$ lbm/sec USING EQUATION (6)

$$A_{fr} = \frac{W_a}{\sigma_a G}$$

$$\sigma_a = \frac{b_a \beta_a r_h}{b_a + b_s + 2a} = \frac{0.413 \times 514 \times 0.00696/4}{0.413 + 0.125 + 2(0.012)} = 0.655$$

$$A_{fr} = \frac{5}{0.655 \times 1} = 7.63 \ ft^2$$

Also, $$L = \frac{V_{tot}}{A_{fr}} = \frac{8.96}{7.63} = 1.175 \ ft$$

H. CALCULATE $\Delta p$ OF THE EXCHANGER FROM EQUATION (7)

$$\Delta p = \frac{G^2}{2 g_c \rho_a} \left[ (K_c + 1) + f\left(\frac{A_a}{A_c}\right) - (1 - K_e) \right]$$

$$A_c = \frac{W_a}{G} = \frac{5}{1} = 5 \ ft^2$$

$$\rho_a = 0.075 \ ft^3/lbm$$

$K_c$ and $K_e$ are found from Kays and London data at $\sigma_a = 0.655$ and $N_{Re} = 0.523 \times 10^3$

$$K_c = 1.09, \quad K_e = -0.44$$

$$\Delta p = \frac{(1)^2}{2 \times 32.2 \times 0.075} \left[ (1.09 + 1) + 0.079 \left(\frac{3390}{5}\right) - (1 + 0.44) \right] = 54.3 \ lbf/ft^2$$

Converting to more useful units

$$\Delta p = \frac{54.3 \times 12}{144 \times 0.433} = 1.04 \ in.\ H_2O$$

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