THE UNIVERSITY OF MICHIGAN

INDUSTRY PROGRAM OF THE COLLEGE OF ENGINEERING

A STUDY OF ENGINE-COOLING CONTROL FOR ARMY VEHICLES

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PREFACE

This report includes a general discussion of methods for controlling engine cooling. The many unforeseen factors which are peculiar to military applications may weigh for or against the various proposed designs differently than have been anticipated. They are listed in the order in which it appears they should be classified, but future modification or unforeseen difficulties may change this order of classification.

The available products and companies mentioned in the report by no means constitute a complete list but are examples of some presently available equipment and a possible source of future information and new designs. Testing and examination of available products for their suitability for military use are not a part of this study.

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I. INTRODUCTION AND DISCUSSION

An engine's source of power is the fuel. In converting the chemical energy stored in the fuel into useful work, only about one-third of the energy in the fuel is obtained as shaft power. One-third is released in the heat of the exhaust gases and one-third makes up the cooling load of the engine. This cooling load must be removed at the rate it is generated, with an ideal system resulting in some constant engine temperature which allows the engine to operate at maximum efficiency.

Some means of removing the heat load, created by engine operation, must be included in the design. This may or may not include control over the rate of heat removed. Uncontrolled systems can vary by the amount of heat generated or by some function of the engine load or even by the engine speed. One system circulates coolant by a change in coolant density which results when the coolant absorbs the heat. The motorcycle has a cooling system which depends on the speed with which it moves as one factor in determining the cooling rate. A third type is demonstrated in the small lightweight-air-cooled utility engine in which the air flow varies with the rpm. In all these cases, the cooling must be adequate under the worst conditions, and losses which occur under more favorable heat-transfer conditions must be tolerated to obtain simplicity and low initial cost.

These uncontrolled systems are subject to various engine operating temperatures depending on the environmental surroundings in which the engine operates. Extreme cold weather can lower the operating temperature to a point where crankcase dilution, accelerated wear, inefficient operation, and icing demand that some type of control be incorporated in the system.

A. Engine-Cooling Sensing Mechanisms, Controls, and Sources of Power

To obtain proper cooling of an engine, it is necessary to sense the amount of heat produced in the engine and also the ability of the coolant to carry the heat away. A measurement of the engine temperature senses both the heat produced and the heat dissipated and is thus an ideal control source. Sensing the heat produced in the engine could also be accomplished by a measurement of shaft horsepower or fuel consumed, but these do not relate the heat produced to the ability of the coolant to dissipate it. For this reason only temperature-sensing devices will be discussed.

The temperature device must transform the temperature indication into a form useful for control. Bellows-operated valves are by far the most common, directly supplying a force which controls the coolant flow. Bellows-type switches and bimetallic units are also common, causing electrical contacts to close or open. It is also possible for these devices to control the voltage or current in an electrical system or to vary the pressure or flow of a fluid circuit. Some of the vapor pressure (bellows-type) thermostats have sufficient force to activate control; otherwise it is necessary to obtain power from an outside source.

One particularly convenient source of limited power with slight engine modifications is the engine lube supply. This is usually accessible at several points and will give reliable pressure for control purposes. A second source is the electrical system which requires no extensive changes if it has sufficient capacity to handle the additional load. Other designs may require auxiliary hydraulic pumps, generators, or take-off shafts for the power source. Simple modification can be made on engines which have fan belts for the installation of an additional pump or generator, whereas a take-off shaft may be more suitable in a new design.

In some cases the control should include a number of elements to sense the temperature of the engine oil, engine cylinder heads, transmission oil, and engine coolant. Under these conditions the control unit must be capable of acting when any one of the units needs more cooling or when all of them need less cooling.

B. Coolant Control

1. FLOW RESTRICTION

This type of control is the most common of all controls and is the standard method used in automotive design. The flow of the coolant is regulated by changing the resistance to flow by valves, louvers, or other throttling devices. The power source, such as the pump or fan, continues to operate at a predetermined speed, with the power requirements changing only by the design characteristics of the unit and the restriction to flow. This type is the most inefficient of any control system but has low initial cost and simplicity.

a. Liquid-Cooled Systems

In the liquid-cooled engines the coolant is constantly circulated by a pump; cooling takes place in a heat-exchanging device when a thermostat opens the passage to the exchanger, and which, when closed, limits the circulation mainly to the engine block. This method is simple, cheap, and reliable. Difficulty can be encountered in extremely cold weather when the forced cooling caused by the movement of the vehicle can bring the engine operating temperature below the control limits of the thermostat. For additional control under these conditions, the engine compartment must be protected from the outside air which can most easily be done by covering all or part of the air inlet. Nothing further is necessary if the thermostat can then control the engine temperature. If outside temperatures vary greatly and frequently, then an automatic control could be installed to vary the amount of inlet-air restriction.

The fan drawing the cooling air through the heat exchanger or radiator is usually belt-driven, and the speed is a function of the engine speed. Recently, variable-maximum torque-clutch arrangements have been designed for automotive use which allows fan slippage at higher speeds when the ram-cooling effects are sufficient to carry the heat load.

b. Air-Cooled Systems

Air-cooled systems do not have the buffer effect of the thermostat in a secondary cooling system to aid in control, so that in these cases the flow of cooling air must be variable and automatic.

c. Power Loss

Inlet- or exit-air passages may be closed but if possible, the inlet control is most desirable to reduce the power consumption when the system is partially closed[1] (see Fig. 1). These curves are typical of centrifugal-type fans.

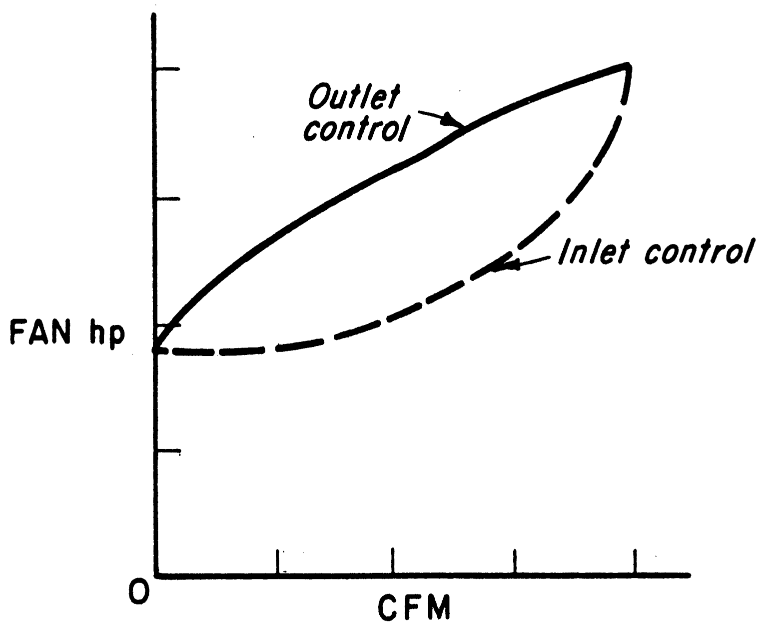


Fig. 1. Power absorption with a louver-controlled centrifugal fan.

The power losses vary with different fans depending on their characteristics. Axial flow fans increase in power requirements with increase in restriction and no saving is obtained with restricting air flow. Use of this system should be only to control the temperature when over-cooling is detrimental to operation (Fig. 2).

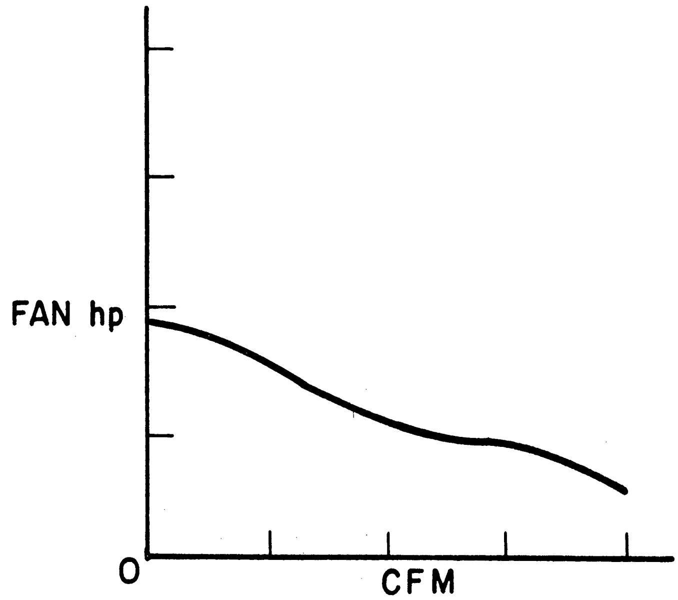


Fig. 2. Power absorption with a louver-controlled axial fan.

Damper control has the effect of altering the total resistance of the duct. A given position of the damper requires a particular head and flow from the fan, the higher head pressure being wasted in heating the air. The horsepower required to drive the fan thus has a higher percentage of loss coupled with the additional loss caused by the fan which may be operating in a region of lower efficiency (see Fig. 3).

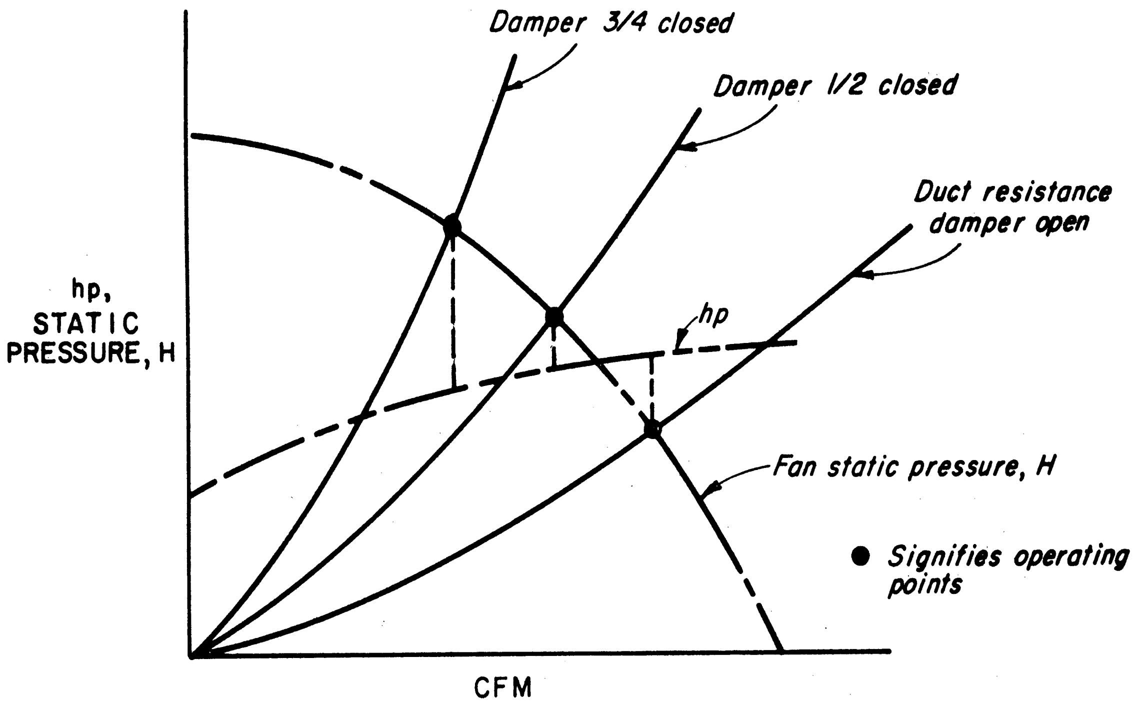


Fig. 3. Centrifugal fan curves with damper control.

2. FAN-DRIVE CONTROLS

Fans can be controlled by controlling the power-driving source. An uncontrolled drive system will transfer power determined by the engine speed and the fan characteristics. Complete control may be affected by an infinitely variable system which would allow any fan speed (and thus capacity) necessary, or a partial control system may be developed with step-type operation.

a. Step Systems

The simplest of fan-drive controls is a two-step system which operates the fan when needed and disengages it when it is not needed. This involves a means of isolating the fan from its power source and engaging it again on command of the chosen control. Some elements involved in this design problem are as follows:

1. inertia of fan and the associated equipment;
2. possible range of engine rpm;
3. cyclic frequency of operation;
4. control sensitivity;
5. heat dissipation of engagement unit;
6. size and weight of components;
7. efficiency; and
8. effect of sudden changes in power absorption on other operating equipment, including vehicle control.

The cyclic frequency mentioned as part of the design problem is dependent on the following:

1. engine load;
2. coolant temperature;
3. capacity of cooling system;
4. sensitivity of the control; and
5. controlled temperature range.

Assuming an air-temperature control, the cycling has the relationships shown in Fig. 4.

The graphs show some of the fundamental problems in determining the cyclic frequency. First (Fig. 4a), the range of the engine temperature must be decided. Control of any additional oil coolers, etc., must also be considered. The control sensitivity is then determined by this range and the lag in the cooling system which will cause the undershoot and overshoot of the temperature. With regard to an air-cooled engine, the engine-head-temperature overshoot may be negligible, while in a liquid-cooled engine or oil-temperature regulating control the increased time delay makes the overshoot problem important. Generally, the control mechanisms themselves can be designed with negligible delays, particularly electric types operating a solenoid or similar fast-acting system. In some cases when the

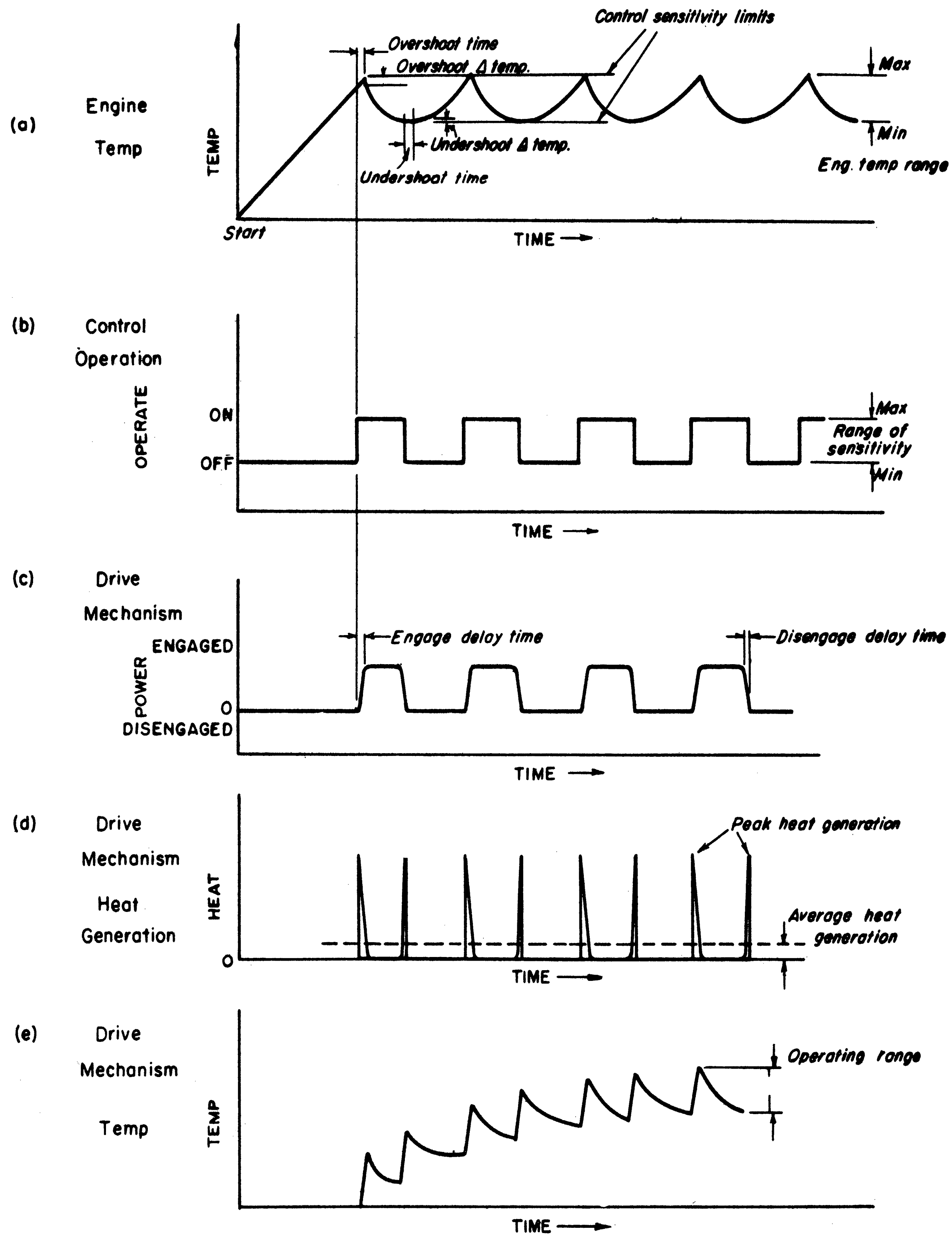


Fig. 4. Heat relationships in an engagement clutch.

control operates an engagement clutch, slower movement may be desirable to reduce the shock load placed on the engaging mechanism and power source.

If the drive mechanism is a complete lockup device, heat generation will occur only when engagement or disengagement occurs as shown in Fig. 4d. If there is some slip during normal running operation, heat will be generated at a level below the peaks depending on the efficiency and the load. The average heat generated per unit time is then a function of the engagement, disengagement losses; the running losses, if any; and the period of cyclic operation. The heat generated under these conditions must be dissipated by sufficient cooling to maintain a safe operating temperature.

Several designs of this type are discussed in Section III of this report.

The next step system to be considered is that using three steps, two fan speeds and stop. Any step system beyond this becomes too complicated to be considered for vehicle fan drives.

Considering changes in speed, the fan laws[2] should be kept in mind, that is, for the same fan keeping the fan diameter and density constant:

air flow (Q) is proportional to speed, N^1;
static pressure (H_s) is proportional to speed squared, N^2;
required fan power (hp) is proportional to speed cubed, N^3; and
efficiency (E) is the same.

The air flow is then roughly proportional to the speed. To provide a minimum cycling of the unit from the low speed to the high, the low-speed range should be sufficient to handle the majority of normal load conditions, with the high speed engaging near the full-power range.

One system of this type was used on the Navy Landing Craft. The fans were driven at either of two speeds and a disconnect by magnetic friction clutch when the vehicle entered water. A sketch of this system is shown in Section III. This design has recently been abandoned and a two-step, on-off system employing only a clutch arrangement is now used.

b. Variable-Drive Systems

In many manufacturing processes and in the military field (i.e., gun-control systems), precise control of the elements involved is essential. On the other hand, the cooling process on engines allow considerable latitude in their temperature and any precise control which entails increased cost, size, maintenance, or inefficiencies would not be good design. It is true that a system requiring variable amounts of cooling air can be obtained efficiently and easily by variation of fan speed if the fan system is considered separately. A fan can maintain a high efficiency with speed variation but the losses and costs remain with the variable drive.

(1) <u>Electric-Type Drives</u>.—To cover the gamut of variable-speed drives would be to include a system which is completely electric. This would consist of an electric generator as a power source and an electric motor as the fan-driving source. An amplidyne generator with the field controlled by the temperature could

vary the driving motor, giving excellent speed control. However, the immediate objection to an all-electric system such as this is the low efficiency. Both units, the generator and motor, have maximum efficiencies of about 85%, giving an overall system efficiency no greater than 70% (i.e., 85% x 85%). This is the percentage of usable fan power, the losses adding to the heat load of the system.

A modified electrical system which has possibilities is the eddy-current coupling. This type of control, along with some others which will be discussed later, are constant-torque transmitting devices. Since the torque output is equal to the input, the loss is then in the reduction of rpm or so-called slip loss. This is well-suited to characteristics of fans. Fan power varies about as the cube of the speed. At low fan speeds, where the slip rpm is high, the power requirements of the fan is low, and therefore transmitted power loss is low. Of course, at high-speed, high-power requirements, there must necessarily be some slip also, to develop the transfer torque through the coupling, and with a simple eddy-current coupling this loss must be accepted. Some couplings have reduced this loss by having cooling water circulate through the air gap which aids in producing torque.[3] This type of water cooling would be suitable for liquid-cooled engines and would allow the heat of the coupling to be taken out through the normal engine radiator. In the case of air-cooled engines, oil may possibly be substituted for water if the coupling were properly designed. This would enable the lube oil to be used for cooling.

Horsepower loss for this type of coupling is easily explained as follows (Fig. 5):

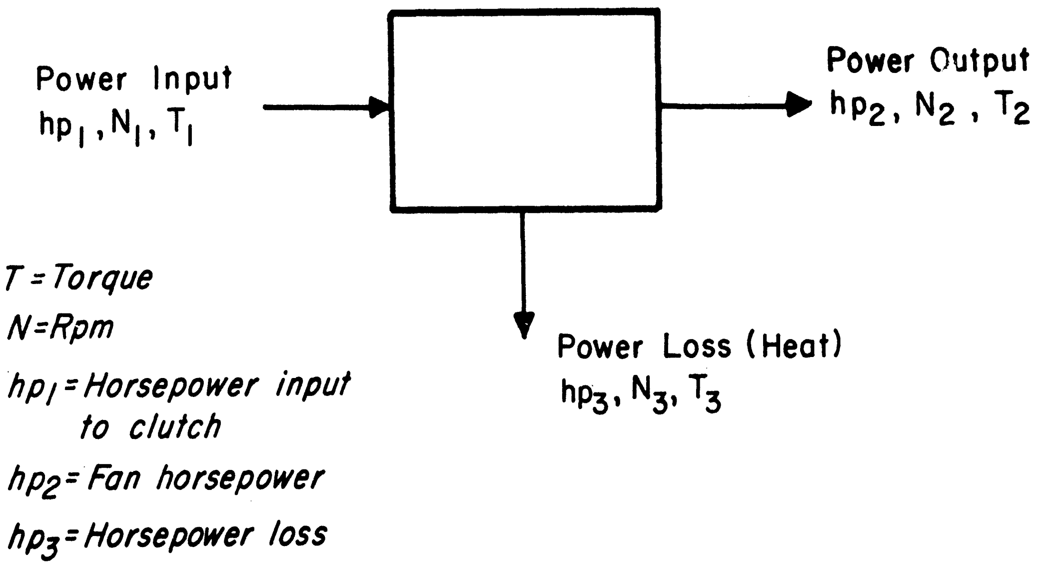


Fig. 5. Block schematic of power relationships using slip clutches.

$$hp_1 = hp_2 + hp_3$$

$$N_1' = N_2 + N_3$$

where N_3 is the slip loss in rpm

$$T_1 = T_2 \quad \text{constant torque}$$

$$hp_3 = hp_1 - hp_2$$

$$hp_3 = \frac{T_1N_1}{5250} - \frac{T_2N_2}{5250}$$

Since $N_2 = N_1 - N_3$ and $T_1 = T_2$,

$$hp_3 = \frac{T_1N_1}{5250} - \frac{T_1(N_1-N_3)}{5250} = \frac{T_1N_3}{5250}$$

$$hp_3 = \frac{T_1N_3}{5250} \times \left(\frac{N_1}{N_1}\right) = hp_1 \frac{N_3}{N_1}$$

$$\text{or hp loss} = \text{hp input} \times \frac{\text{slip rpm}}{\text{input rpm}}$$

$$hp_3 = \frac{T_1N_3}{5250} = \frac{T_2N_3}{5250} = \frac{T_2N_3}{5250} \times \left(\frac{N_2}{N_2}\right) = hp_2 \frac{N_3}{N_2}$$

$$\text{or hp loss} = \text{fan hp} \times \frac{\text{slip rpm}}{\text{fan rpm}}$$

The rated hp represents a given engine speed and the equivalent fan speed with 0% slip (i.e., direct drive). The equation is illustrated in graph form in Fig. 6.

The maximum loss in the coupling unit is 15% of the rated fan hp at 2/3 of rated speed. Comparing this to the power required if the fan were running in direct drive, a substantial saving can be had in all but the maximum fan-load requirements where direct drive would be advantageous.

A further refinement of this system would be to incorporate a magnetic friction clutch with the eddy-current clutch. The eddy-current clutch could then operate when less cooling is needed and the magnetic clutch could give mechanical lockup at peak loads when highest efficiency is desired.

A second modified electrical drive is that of a magnetic particle clutch. This has essentially the same operating characteristics as the eddy-current clutch with one important exception: the unit can operate at zero slip removing the loss at the maximum speed condition. Both types, of course, face the problem of heat dissipation when slipping and this must be considered in the design. As an example, a fan absorbing 60 hp at a given speed could be reduced by the control to 2/3 speed through clutch slippage, whereupon the clutch would absorb 15% of the

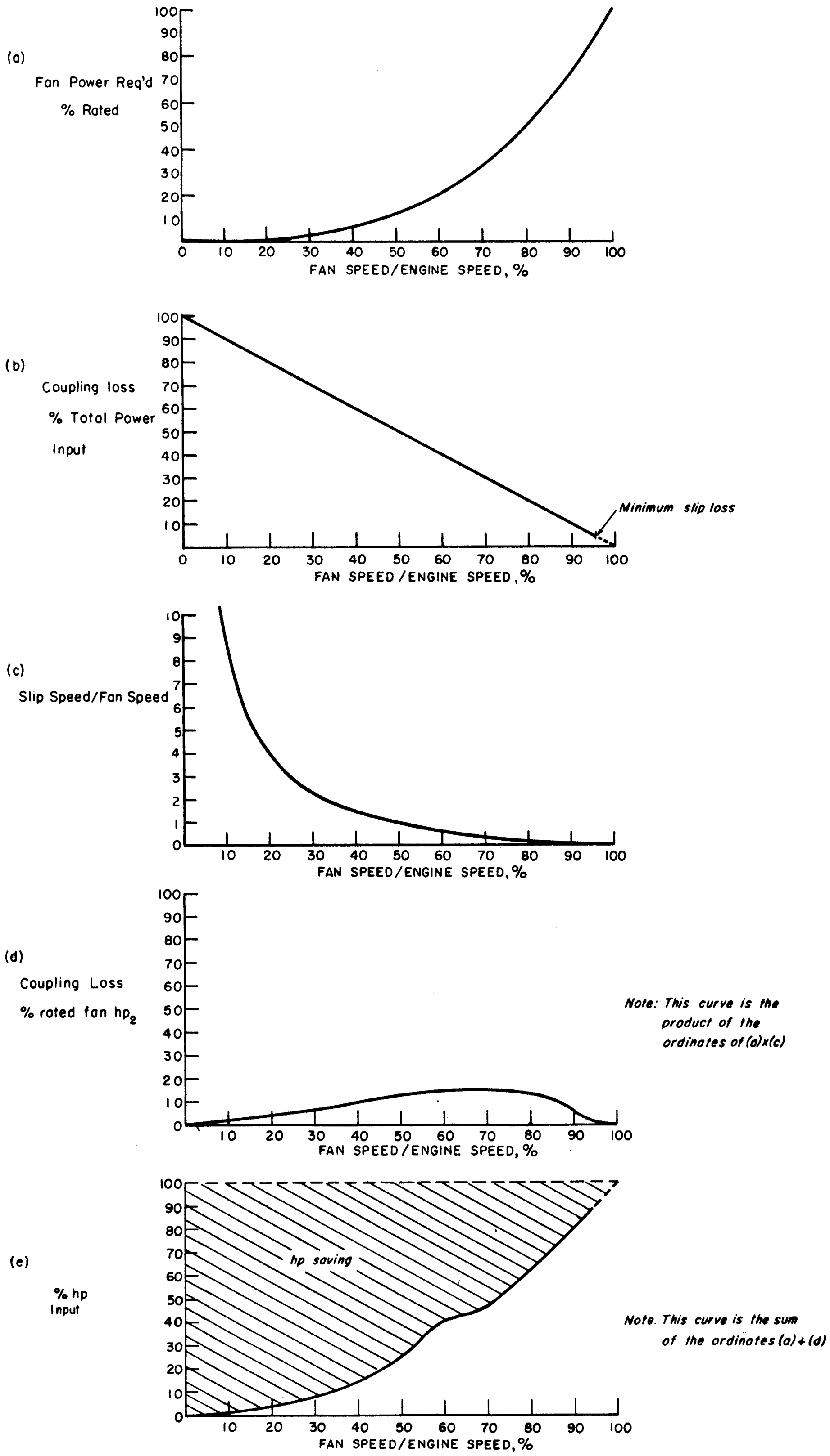


Fig. 6. Curves of power relationships in a slip-type clutch.

60 hp or 9 hp which is equivalent to about 6700 watts. Cooling air of 5000 cfm would increase about 4°F from this dissipation alone, with the major problem being that of heat transfer to the cooling air. Fan power would be reduced to about 18 hp, resulting in a total reduction of 33 hp in shaft power into the clutch-fan combination.

(2) Hydraulic-Type Drives.—Use of hydraulic power parallels the electrical systems. A hydraulic pump and motor are capable of control similar to the electric generator and motor but with equally poor efficiency.* The typical design involves a hydraulic pump delivering directly to a motor which is driving the fan. Control is effected through a swash plate adjustment in the pump or some other means of changing the pumping capacity of the pump. A second method, resulting in greater losses, is to provide a relief valve in the connecting line and bleed off a portion of the flow necessary to reduce the fan to the desired speed. Any method of this type will result in a nice control package, but the high power loss remains.

The fluid coupling, however, similarly corresponds to the eddy-current and magnetic particle couplings and is therefore another possible usable control. Most fluid couplings are used as shock-absorbing devices, allowing the driving source to get up a relatively high starting speed and gradually overcome a high inertia load. A modified form of this simple coupling can be used to vary speed.

The basic elements of the coupling are the driving rotor, the receiving rotor, and the casing. Power is transmitted from the driving rotor to the receiving rotor by the entrapped fluid. As in the electrical couplings, this coupling has torque output approximately equal to the torque input, making the losses a function of the slip rpm.

$$hp_3 \text{ (loss)} = hp_2 \text{ (fan)} \times \frac{N_3 \text{ (slip rpm)}}{N_2 \text{ (fan rpm)}}$$

Most couplings have a reservoir incorporated in the design which modifies this relationship. Under high slip conditions, fluid is stored in this space and the result is a dropoff of torque at higher slip (see Fig. 7). This prevents overloading when starting.

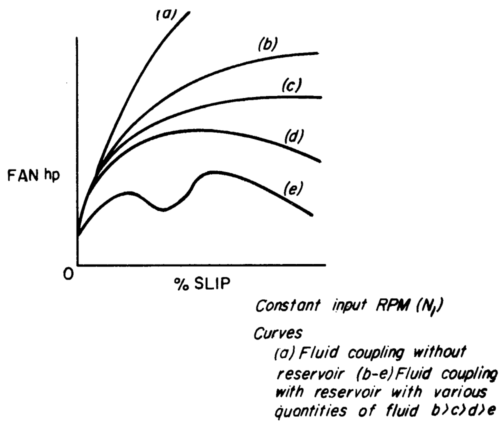


Fig. 7. Effects of fluid quantity in a fluid-coupling circuit.

*Under some conditions, efficiencies can be improved (see Design Section II).

The operation of the fluid coupling depends on the entrapped fluid transferring power at a rate dependent on the mass of fluid which is circulated across the rotors and the fluid velocity. The velocity is determined by the difference in the two rotor speeds. The mass flow is determined by the amount of fluid in the active system and the circulating velocity. Since it is desirable to have the fan speed independent of the engine speed and determined only by the air flow required, the difference in speeds and the resulting fluid velocity is fixed for any set of conditions, and thus it is necessary to change the quantity of fluid involved in the power train to obtain stability at any engine-fan speed relationship. This is done in a variable-speed coupling by a dump and fill mechanism which can regulate the quantity of oil as needed. This mechanism can work through any of the following three methods:

1. Regulating oil flow into coupling with a restricted flow outlet.
2. Regulating oil flow at outlet of coupling with restricted flow inlet.
3. Regulating inlet and outlet flow.

The common method is to regulate the outlet flow by a scoop tube which adjusts in length and thus removes all oil within the radius of the scoop inlet. This method directly relates the position of the scoop to the quantity of the oil in the coupling.

If only two-speed control is desired, two conditions must be met in the coupling, that is, coupling full for the fan on, and coupling empty for disengagement. In this case, a variable scoop would be an expensive solution to the problem as a stationary scoop or drain hole with control valve could easily do the job.

The ability of the coupling with a fixed amount of fluid to transmit power varies with the slip (Fig. 7). The amount of slip depends on the amount of fluid in the coupling (b) containing the greatest amount of fluid in the reservoir-type coupling with the curves showing decreasing amounts to curve (e) which has an insufficient amount of oil to fill the fluid circuit in the vanes and is an unstable condition. The control unit, therefore, can vary the amount of oil to one of an infinite number of curves in the stable region between curves (b) and (e). Then for any given engine speed, the fan speed will stabilize at the intersection of the fan-power curve with the capacity curve. Figure 8 shows the operation of a variable fluid coupling and the torque requirements of a fan drive. Operating points are at the intersection of the fan-torque curves with the coupling curves.

An advantage which the fluid couplings may have over other slip-type couplings is the method of heat dissipation. All couplings ultimately must have the heat removed by the cooling air. The fluid coupling can have a secondary source through a common oil supply with the engine and could dissipate the heat through a common oil cooler. Continuous circulation of oil or thermostatically controlled circulation would be necessary.

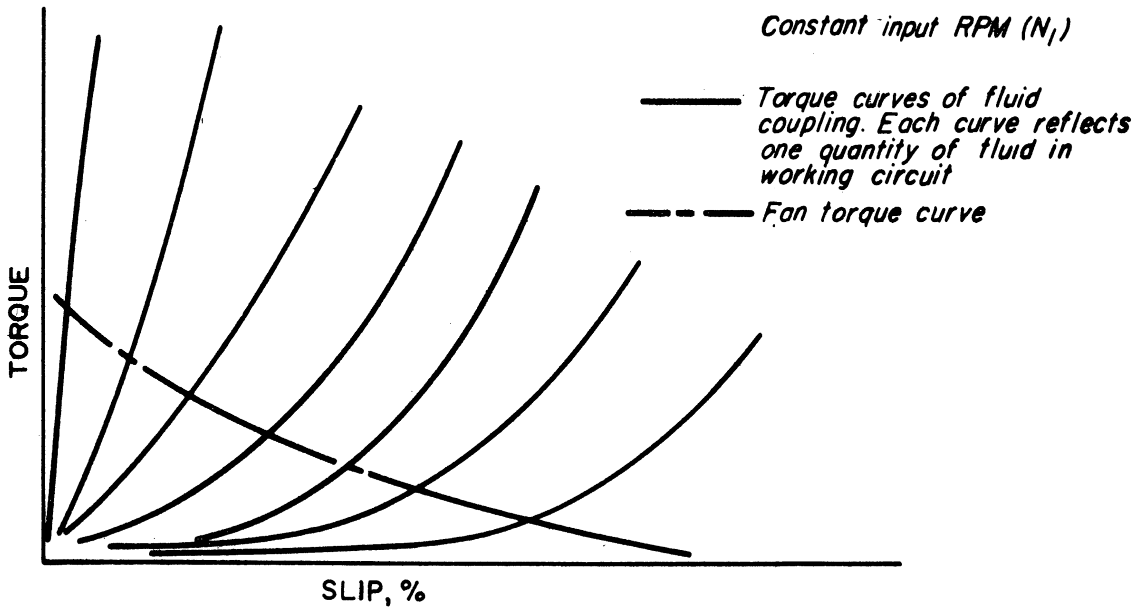


Fig. 8. Characteristics of a variable-speed fluid coupling.

All slip couplings have the advantage of low heat generation at times when the engine is hot under high load conditions. Maximum engine-power requirements result in maximum engine-heat generation and maximum air flow. This means minimum slip loss and greatest efficiency. Exceptions may be only in extremely cold climates or emergency conditions where high power is required before engine warmup is completed.

(3) Mechanical Drives.—Mechanical drives, with reliable gear trains, remain the most efficient of all power-transfer methods.[4,5] But even this method loses some of its desirability when variability is included as a design criterion. The complexity of variable mechanical drives results in high cost and many parts. If a number of gear engagements are involved in the process, the efficiency also can drop below that of a fluid coupling operating at the minimum slip condition.

A common design now in use is a differential drive arrangement where the power is conducted through two parallel paths. One is a completely mechanical gear setup having the low power loss, and the second is the variable-speed transmission with a higher loss. To maintain the highest efficiency, it is necessary to pass the greatest power through the fixed mechanical path and as little as possible through the variable path. The variable path could also be mechanical or hydraulic coupling or other types. With the large power ratios, efficiency is maintained; however, the control of the variable system on the output diminishes as this power ratio increases, and for fan application it has limited application.

3. FAN- AND PUMP-CHARACTERISTICS CONTROL

Control of the cooling system is accomplished by controlling the coolant. We have previously discussed changing of the flow restriction for control of the coolant, and of changing the fan speed. Both of these systems are distinct from the fan itself. The remaining control system possible is within the fan, by a change in the fan characteristics.

Fans are classified into two general types, centrifugal type and axial-flow type. They are further classified by the amount of pressure they are to develop, their relative position in a duct system, and the construction details which determine their characteristics.

a. Centrifugal Fans

Centrifugal fans vary in the position and shape of their blades. Radial types have the outer parts of the blades fall on radii. Backward-tipped blades are curved away from the direction of motion and forward-tipped blades curved toward the direction of motion (Fig. 9).

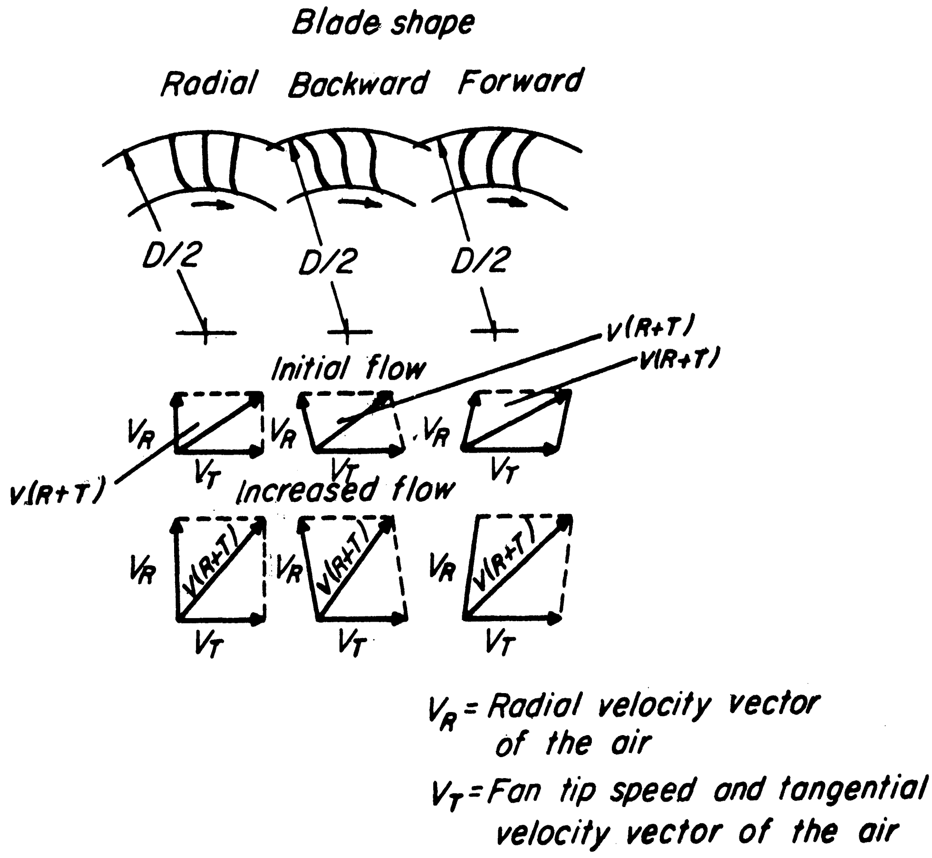


Fig. 9. Velocity relationships in the typical centrifugal fans.

Figure 9 indicates what might be expected in the characteristics of the various fan types because of the shape of the blades. The vector diagrams show the radial component of the velocity which is dependent on the quantity of air flowing through the fan and in the direction which the blades force it to flow. The tip-speed velocity imparts tangential motion to the air dependent on the tip speed which in turn is dependent on the speed and fan diameter. The vector sum of these two velocities (V_R+V_T) gives the absolute velocity of the exit air leaving the blades. The tangential component of this resultant vector V_{R+T} is roughly proportional to the total pressure (H_t) rise of the fan. Generally speaking, the characteristics of the three fan types will follow the curves of Fig. 10.

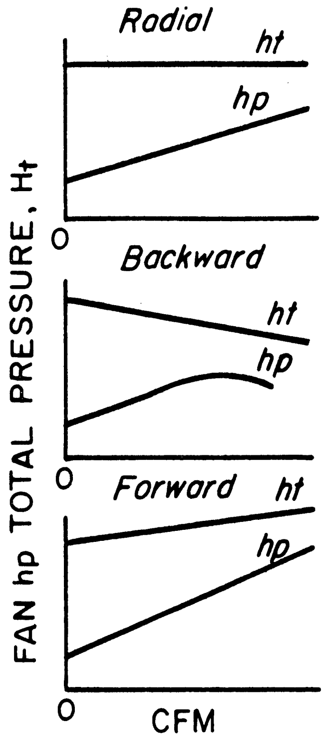


Fig. 10. Characteristics of the three general types of centrifugal fans.

The radial fan, having a nearly constant tangential velocity component of V_{R+F}, will have constant total pressure H_t and with an increase in air pumped will have an increase in power absorbed. As the flow increases in the backward flow fan, the tangential velocity of V_{R+T} tends to decrease, giving a drop in total pressure and a maximum point in the power curve. The fan with forward-curved blades increases in total pressure and in power with an increase in flow.

Backward-curved fans generally have higher efficiencies, higher speed, and larger size than forward-curved fans; the radial blades are between the two extremes.

The centrifugal fan would lend itself well to the T-95 type of tank installation where ducting forces the cooling-air flow to turn a sharp 90° bend. This is a natural point to have a centrifugal fan as in its operation the flow outlet is turned 90° from the inlet flow.

Damper control and speed control, already discussed in their respective sections, would work well on centrifugal fan applications. The common characteristics control remaining on centrifugal fans is that of inlet-vane control. Since the performance of a fan is related to the inlet-air velocity vector as well as the action and position of the rotor blades, changing the inlet vanes can affect the performance. If the inlet vanes cause prerotation of the entering air, fan capacity is reduced. If this prerotation is in the direction of the fan rotation, the power as well as the capacity is reduced (Fig. 11).

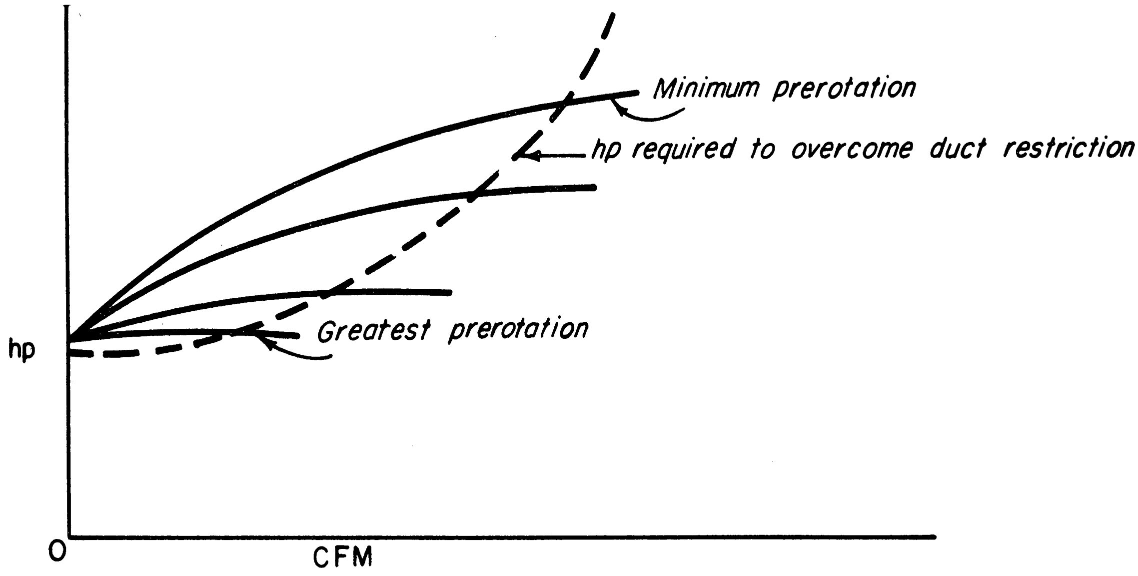


Fig. 11. Effects of inlet-vane position on fan characteristics.

With this type of control, some of the efficiency is sacrificed at the lower capacities but the result is still a decrease in shaft horsepower.

b. Axial-Flow Fans

Axial-flow fans were originally used where large volumes of air were transferred against low resistance. Developments have since led to their use in applications requiring high head and operation at high efficiencies. This has been accomplished with properly designed air foil blades, directing vanes, and proper inlet and discharge duct work.

Axial-flow fans have characteristics similar to those shown in Fig. 2, that is, a decrease in head pressure and power with an increase in flow. This rules out a damper-type device for any economically running system. Variable speed gives similar results with axial-flow fans as with centrifugal fans. Changing fan performance, by changing fan characteristics, which is the discussion in this section, can be varied by blade length, blade angle, or vane control.* Of these three, the most promising is the blade-angle control. Blade control maintains a high efficiency whereas the vane control does not. There is also the possible advantage of partially reversible flow. Installations which discharge air over hot muffler

*Nomenclature varies for the various elements involved depending on their use as turbines, compressors, or ventilating equipment. For the discussion here, the term "blade" is defined as the projections on a moving wheel which impart energy to the air. Vanes are the nonrotating segments which may be before or after the fan wheel and act as straightening devices. "Blade angle" is a general term applied to the angle of attack of the blade similar to the pitch angle of an aircraft propeller.

surfaces, such as the T-95 tank, would then be able to have a fast preheat of the engine compartment immediately on starting. Inlet air would enter through the rear of the tank through the exit grill. In climates where snow and ice are troublesome, this exit grill is more apt to be free of obstructions. The fan would then exhaust through the normal inlet grills on top of the tank and aid in removing the ice.

Variable-blade-angle fans have been used in large installations for a number of years. Aircraft propellers with variable pitch blades are a common item using similar principles of control.

Capacity can be varied to meet any condition from zero, or even small negative values, to full-rated fan capacity, the flow varying directly with the blade angle. Efficiency remains high at all capacities where any significant amount of shaft power is required (Fig. 12).

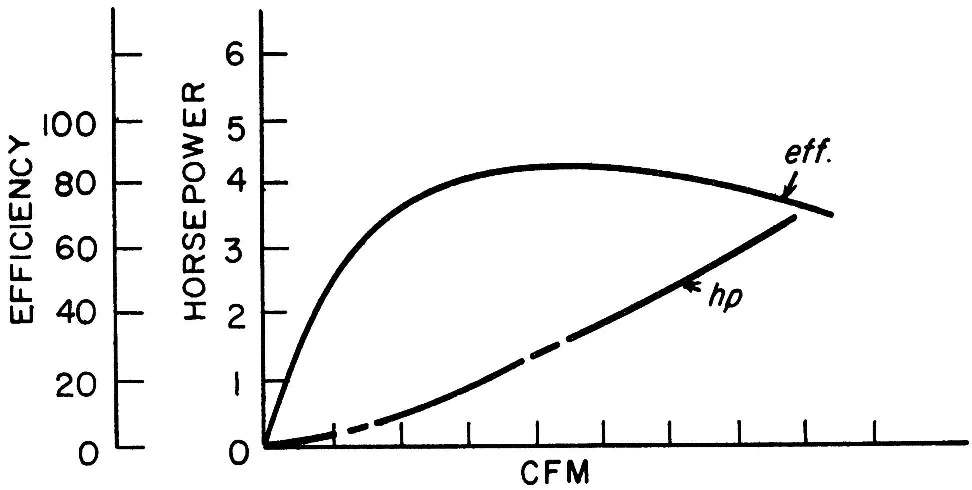


Fig. 12. Effects of blade-angle control on performance.

The curves shown in Fig. 12 are the result of proper blade angle at the various quantities of air flow which allow the fan to operate near the peak efficiency. The individual curves such as those in Fig. 13 are the individual performance curves at set blade angles. The maximum points in an infinite number of these curves make up the single curves of Fig. 12.

Control of the blade angle can be by electrical, mechanical, or hydraulic means. Many devices using these methods of control have been developed for use with variable pitch propellers. Automatic propeller controls fix the angle by sensing the engine rpm and power required. This is accomplished by using a centrifugal governor and modulating this signal by the position of the throttle control. In this manner the engine can operate at full power on take-off with a low blade angle, high rpm, which is some 15% more power than available with a fixed propeller. Lower power requirements needed when cruising increase the blade angle and keep the rpm down to an economical level.

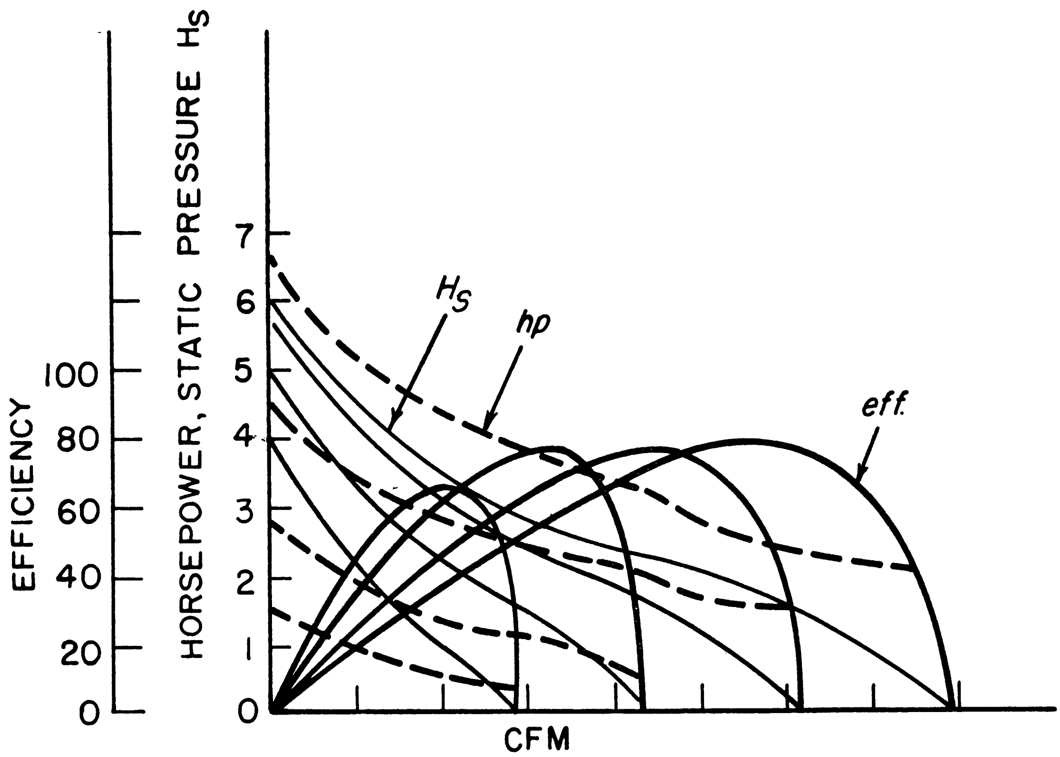


Fig. 13. Individual curves showing effects of blade angle on fan performance.

The actuating devices to change the blade angle are located in the propeller hub. Hydraulic types have a piston which operates cams. The oil pressure on the piston is fed through the prop shaft and is adjusted by a pressure-regulating control valve and oil pump located in the engine compartment. Electric types have a reversible motor located in the hub with the control current fed through slip rings. Some mechanical devices have an auxiliary shaft running at the same basic rpm as the prop shaft. Slight changes in the rpm of the auxiliary shaft result in a difference rpm between it and the prop shaft which powers a gear mechanism. Another mechanical type has a nonrotating collar about the drive shaft and which can move along the axis of the shaft. With the desired blade angle, no engagement or wear occurs. If a change in angle is desired, the collar moves and engages in an inclined projection on the propeller hub, causing movement in a ratchet mechanism changing the blade angle. Movement of the collar in the opposite direction engages a second projection which reverses the ratchet rotation.

All these systems mentioned above are fairly expensive devices. Perhaps a development program would simplify the design and reduce the cost.

One possible design which is a simplified approach to blade-angle control is described in Section III (variable blade angle, bellows control). This design proposes air-temperature control directly to an actuating bellows thermostat. Such a design has no outside fan connections and would be adaptable to all present air-cooled engines by adding an adapter to the engine-fan drive shaft. Some of the problems involved in the design of such a fan control are discussed in the design section of this report.

c. Variable-Flow Pumps

Variable-flow pumps may have some pertinence to the liquid-cooled engine installations or perhaps in the hydraulic control circuits.

Piston-type pumps with swash plates are one form of variable-flow pump and common in control systems. The angle of the swash plate can be varied with respect to the piston position and in this way the stroke is varied which in turn varies the capacity.

The radial-type pumps consist of two cylinders, the smaller cylinder of which rotates inside of a larger stationary cylinder. Vanes are attached to the rotating cylinder and extend radially outward, forming pockets between the inner rotor and outer casing. Then the volume of the pockets between the vanes varies as the rotor rotates, depending on the degree of eccentricity of the two cylinder centers. Thus varying the eccentricity will vary the flow. Both of these variable-flow pumps are produced at the present time (see Appendix).

A third type has recently been introduced by Thompson Products which uses temperature control. The vanes of a rotary-type pump are made of bimetallic materials, so that the temperature of the fluid being pumped determines the pump capacity. Depending on the placement of the bimetallic vanes, the capacity may increase or decrease with an increase of fluid temperature. Such a system may be very well suited to liquid-cooled engines.

II. DESIGN INFORMATION

A. Dynamic Effects of Control[6]

Rapid development of control systems has taken place in the last 15 years and they now occupy a prominent place in the engineering art. Information on the subject now fills many volumes and is quite complex. There are, however, a few facts which should be kept in mind concerning controls and their operation.

First, control systems can be classed as open or closed systems. An open system in cooling control would involve a temperature sensing mechanism feeding into a controller and power amplifier which would actuate the final control element (i.e., louver, speed reducer, etc.). The closed system has a means of feeding the position of the final control element back to the controller. By this means the instantaneous position of the element is compared to the new desired position and thus the rate of change of the element can also be controlled. Rapid changes can be used where the error is great and slow movement can be had when error is small. This can eliminate overshoot tendencies and still make for a rapid, accurate system.

Secondly, control systems respond to the input signal in a manner which is affected by the natural frequency and the makeup of the component parts.

The input signal may be a sudden (step function) change in voltage, current, mechanical position, or hydraulic pressure. There may also be a steady increasing or decreasing change (ramp function), or one of many other types or combinations.

The response of the control system then depends on the control-system design. If it tends to oscillate freely, that is, if there is little resistance in the electrical circuit or little friction in a mechanical or hydraulic circuit, the unit will tend to overshoot and oscillate at the natural frequency of the system about the new steady-state level. The amount of oscillation and the rate of decay of the oscillations are dependent on the friction of the system and the nature of the input signal. Under some conditions this could make a control very unsatisfactory and cause rapid wear on the parts involved. To avoid this, damping is added to an electrical circuit by addition of electrical resistance, to a hydraulic circuit by addition of an orifice, and to a mechanical circuit by addition of friction. For a given natural frequency the response will be most rapid with the least damping (i.e., the least resistance or friction), but will take longer to reach a steady state because of the oscillations. Increasing the damping increases the time for the control element to reach a desired level but reduces oscillations after reaching the desired steady-state level. For these reasons a control system near the critically damped value is often used which results in minimum time to reach the steady state without oscillating. Overdamping in addition to this critical value only makes the system insensitive and it never reaches the desired steady state.

Another method of changing the overall system response is to change the natural frequency. Higher natural frequencies will reduce the response time. The proper selection of various elements, their natural frequency, and amount of damping desired depends on the design.

A completely different aspect of these dynamic problems is the cyclic rate of operation of the on-off type systems. The general operation of a drive clutch in this regard has already been discussed in Section I, with the problems of temperature control and clutch-heat dissipation shown in Fig. 4. Engine-cooling problems can be divided into two types. One type is the completely closed cooling system which depends entirely on the fan for cooling, and the other type is that which is aided by the cooling effect of the moving vehicle. It is conceivable, in the case of the open systems, that a fan would never be engaged when favorable speeds and loads were maintained. This is not the case in the closed system, and without the aid of supplementary cooling, cycling must take place under all conditions after initial engine warmup.

Some estimates of what type of cycling rate may be expected in a system are shown in the calculations which follow on an AOS-895 engine.

Engine weight - 1894 lb
Heat rejected to cooling air - 14,800 Btu/min at 2800 rpm F.T.
Estimated heat rejection (under a normal condition of about half load) - 8000 Btu/min

Assume head temperature to be controlled between 425°-475°. If the cooling surfaces (i.e., cylinder heads and cylinder) comprise 1/5 of total engine weight,

$$1/5 \times 1894 = 379 \text{ or } 380 \text{ lb} \quad .$$

These are assumed all aluminum with a specific heat of .22. 50°F x .22 x 380 = Btu to raise temperature from 425° to 475°:

$$\frac{50 \times .22 \times 380}{8000} = .522 \text{ min to raise } 50°\text{F, assuming no radiation, uniform temperature rise} \quad .$$

This indicates that the "off" portion of the fan cycle would be about 1/2 minute. Similarly, the "on" portion of the cycle is also 1/2 minute. The cooling system at half load is running at one-half capacity and will reduce the temperature at about the same rate as it had increased. F ctors which tend to make the actual cycling rate less than estimated may be the concentration of heat in a more local area of the cylinders, error in estimating the mass involved, and the fact that a portion of that mass is steel rather than aluminum (steel has the lower specific heat). A 50° allowance in temperature variation is also a broad range. With a 25° temperature control, the cycle rate is increased to once every 30 seconds. A 500-hour life expectancy multiplies this figure to 60,000 operations.

A by-product in a two-step fan control on an engine such as this is the effect on vehicle operation. Fan engagement and disengagement has the effect of changing the vehicle driving power 10 to 15% which could prove annoying to a driver if not detrimental to vehicle control.

B. Noise Control

The noise level and noise characteristics of the cooling system are directly affected by any attempts made to control the cooling air flow. These effects are great enough to warrant consideration in the decision to control cooling on the large air-cooled engines. The fan noise level is very high and presents one of the major noise sources in tank vehicles.[8]

Two types of noise are generated by the cooling fan, rotational noise, and vortex noise.[7] Rotational noise produces a fundamental tone and harmonics of that tone by the impulse of each blade as it passes through the air; for symmetrically spaced blades, the fundamental tone is equal to the number of blades which pass a given point in a unit time.

$$\text{Fundamental frequency (cps)} = \frac{\text{fan rpm x no. of fan blades}}{60}$$

The turbulent noises generated in a random pattern across a wide frequency range are the second group of noises. These noises are generally caused by eddies around the fan blades.

Overall sound power varies with the fan conditions as follows:

$$70 \log_{10}\left(\frac{D_2}{D_1}\right) + 50 \log_{10}\left(\frac{S_2}{S_1}\right)$$

$$20 \log_{10}\left(\frac{D_2}{D_1}\right) + 25 \log_{10}\left(\frac{H_2}{H_1}\right)$$

$$10 \log_{10}\left(\frac{Q_2}{Q_1}\right) + 20 \log_{10}\left(\frac{H_2}{H_1}\right)$$

where

D = fan diameter or size
S = fan tip speed
H = static fan pressure
Q = air flow

These relationships are based on the fan laws:

$$\frac{Q_2}{Q_1} = k_1 \left(\frac{D_2}{D_1}\right)^3 \times \left(\frac{S_2}{S_1}\right)$$

and

$$k_2 \left(\frac{D_2}{D_1}\right)^2 \times \left(\frac{P_2}{P_1}\right)^{1/2}$$

$$\frac{P_2}{P_1} = k_3 \left(\frac{D_2}{D_1}\right)^2 \times \left(\frac{S_2}{S_1}\right)^2$$

$$\frac{hp_2}{hp_1} = k_4 \left(\frac{D_2}{D_1}\right)^5 \times \left(\frac{S_2}{S_1}\right)^3$$

and

$$k_5 \left(\frac{D_2}{D_1}\right)^2 \times \left(\frac{P_2}{P_1}\right)^{3/2}$$

where k = constant..

To obtain 1/2 air flow by changing the blade angle, the head (H) reduces to between 1/2 to 1/4 the initial value. Assuming 1/3 as an approximate value, the noise level is:

$$10 \log_{10} \left(\frac{Q_2}{Q_1}\right) + 20 \log_{10} \left(\frac{P_2}{P_1}\right)$$

$$= 10 \log_{10} \left(\frac{2}{1}\right) + 20 \log_{10} \left(\frac{3}{1}\right) = 3 + 9.6 = 12.6 \text{ db} \quad .$$

The static pressure effect is difficult to determine in this case. In laminar flow the resistance to flow varies directly as the rate of flow. In turbulent flow, the resistance varies as the square of flow. Thus the noise reduction could possibly vary from 3 db to 15 db lower at 1/2 flow conditions, depending on the degree of turbulence.

If the fan drive is controlled and the speed then varied as needed, the fan running at 1/2 speed and flow would give a reduction in noise level:

$$50 \log_{10}\left(\frac{N_2}{N_1}\right) \equiv 50 \log_{10} 2 \approx 15 \text{ db} \quad .$$

For a comparison of recorded and calculated results, data obtained by K. A. Beier and T. J. Weir[9] can be used (see Appendix).

Fan Speed, rpm	Recorded Noise Level, db	Tip Speed Ratio, S_1/S_2	$50 \log_{10} \frac{S_1}{S_2}$ = Calculated* Noise Reduction, db	Recorded Noise Reduction, db*
2880	91	---	---	0
2500	88	1.15	3.1	3
1950	84.5	1.475	8.5	6.5
1600	80	1.8	12.8	11
1350	75	2.13	16.4	16

*Reduction in noise level, db, from highest level at 2880 rpm.

C. Reduction of Fan-Power Requirements[1,10,11]

Considering cooling control and power consumption in the engine-cooling process, the air flow required for any given set of conditions is determined by the engine design. If the fan control limits the cooling-air flow to this minimum, then the fan-power losses will be a minimum for that design. Possible additional gains may be realized if the complete basic cooling system would be re-evaluated in the light of new concepts which have arisen in the past few years. If an increase in the rate of heat transfer between the engine and the air could be realized, a reduction in air flow and fan power could be accomplished. Such an increase in efficiency may be possible by introduction of such innovations as pin-type cooling surfaces which are capable of greater heat transfer than conventional fins. Modified pin arrangements may be made practical in the original finned castings or perhaps fabricated after casting by some cheap process.

Another possibility in power conservation is to increase the efficiency of the fan. No doubt many types of fans have been tested and the fans now in use are the best performers under the present conditions. Some basic research on fan placement in relation to the engine, size, speed, hub diameter ratio, vane positioning, blade angle, and contour may discover new and untried possibilities in this field.

A third possibility is a more closely integrated engine and engine compartment which would reduce the resistance to air flow. Elimination of restrictions, bends, variations in effective duct cross sections, and an inspection of ballistic grills may lead to power reductions. Along this line automotive manufacturers have gotten into difficulty with the addition of power steering, power brakes, and air-conditioning equipment which is obstructing the cooling air flow. This has led to an increase in fan size and an increase in fan speed, neither of which is conducive to economy.

D. Design Components

Three basic block schematics may be constructed illustrating the three possible types of control:

1. flow-restriction control (Fig. 14);
2. fan-drive control (Fig. 15); and
3. fan-characteristics control (Fig. 16).

The basic building blocks of these systems are:

1. sensing device;
2. controller;
3. controller power source;
4. fan or pump; and
5. fan or pump power source.

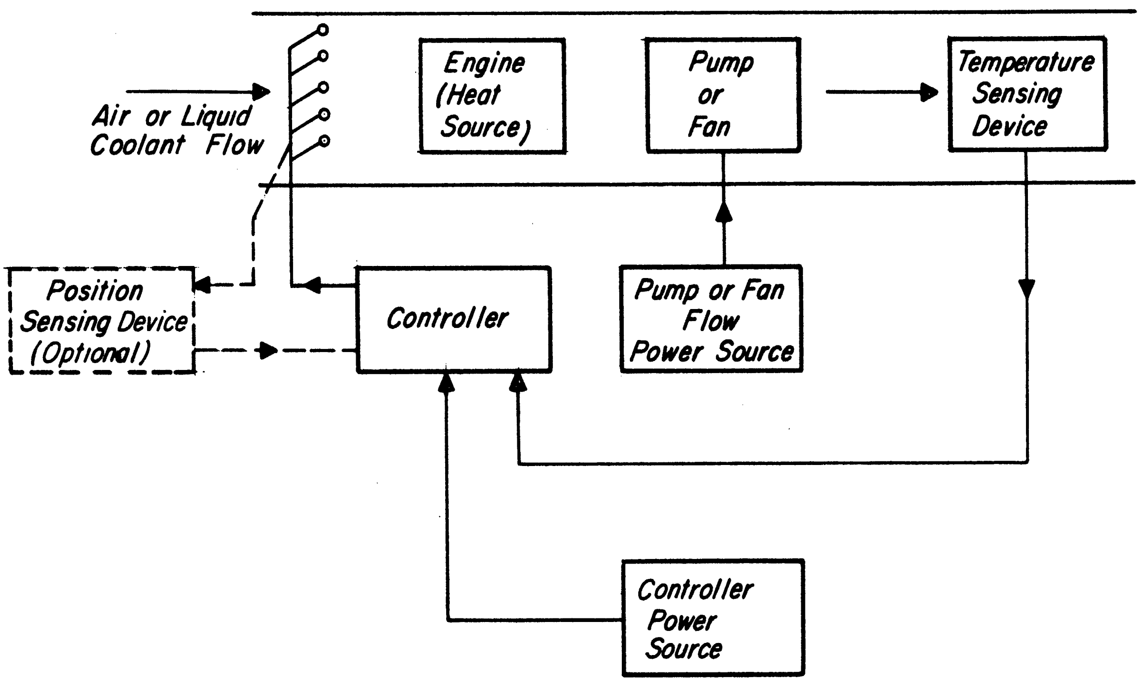


Fig. 14. Block schematic of flow-restriction control.

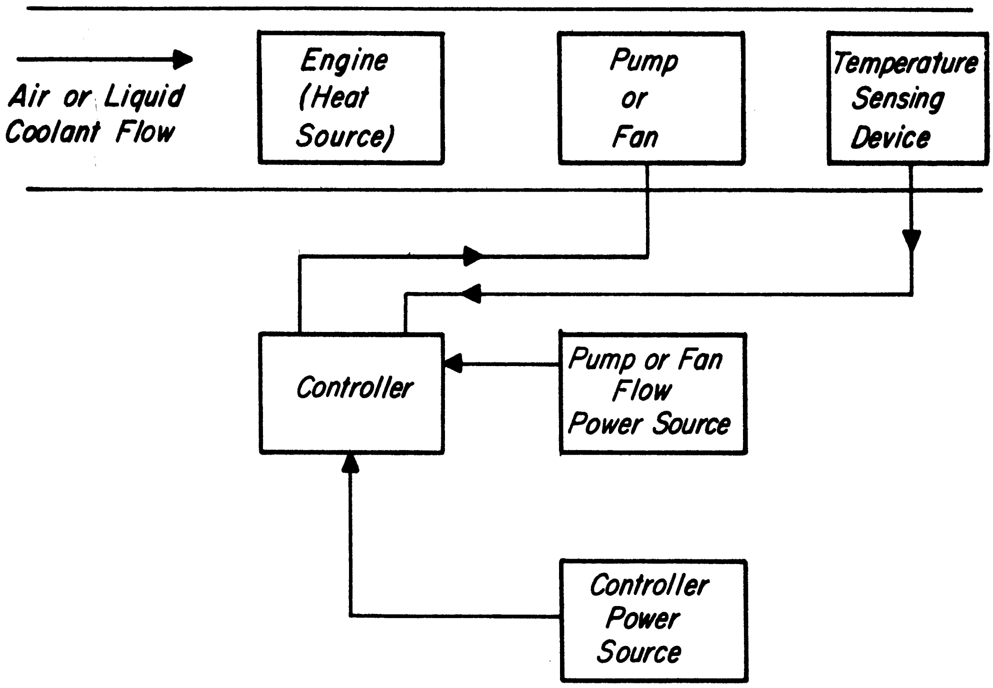


Fig. 15. Block schematic of fan-drive control.

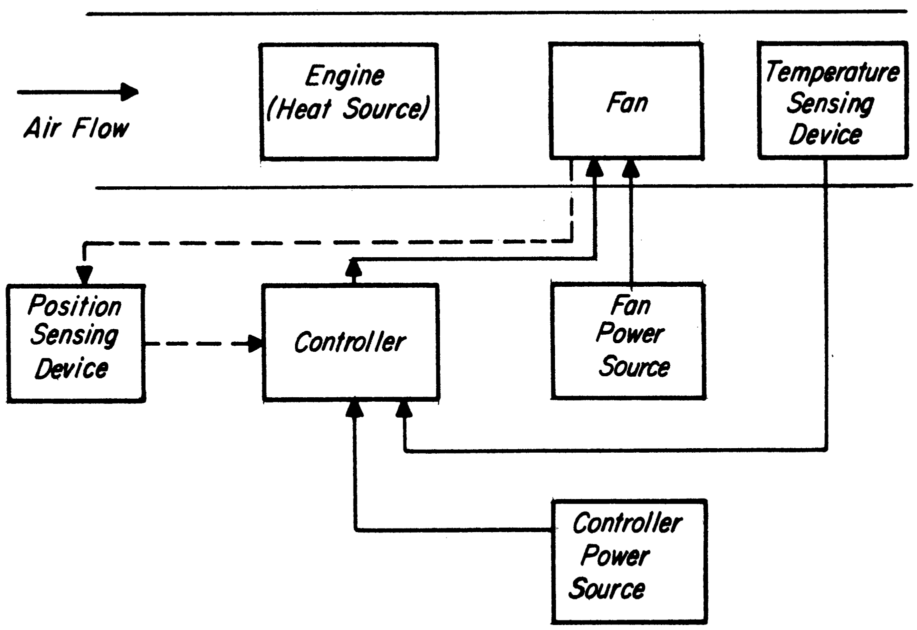


Fig. 16. Block schematic of fan-characteristics control.

Some items which can be used as building blocks for these systems can be found on the following pages. Many of the items are designed for use in servomechanisms, and in fact, some of the usable fan controls are servomechanisms. The term servomechanisms covers all power-amplifying systems. This includes the control mechanisms which have a receive unit following the action of a send unit with power amplification between the two units. For cooling control the sender is positioned by the temperature and the receiver varies louvers, fan drive, or fan-blade angle. Power is supplied externally.

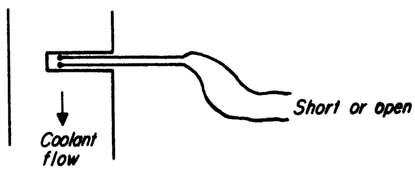
1. TEMPERATURE CONTROLS

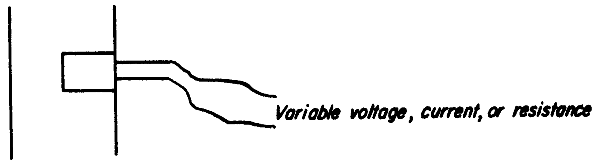
The type of temperature-sensing device used depends on the nature of the application. In a truck engine usually only one temperature is of interest, the engine-coolant temperature. Tank vehicles, however, may be concerned with the head temperature, engine-oil temperature, and transmission-oil temperature.

In the case of the truck engine with one temperature to control, one sensing device will do the job. Controls in tank engines must be approached separately for each design. In all cases the system is cooled by air-flow paths which merge into one path through a fan (the AV-1790 uses two fans). Thus control of the single path by changing fan capacity or restriction of the single path will reduce the air flow through each of the cooling units proportionally. This is a desirable condition if the heat generation in each of the units is also proportional. A single sensing device could then handle the control with warning devices to cover any abnormal condition.

Where heat generation in the units is not proportional or the control of temperature may be critical, more than one sensing device are needed. Under these conditions electric actuated controls offer the simplest solution. Two or more switches can be placed in parallel for on-off control of a solenoid or relay. It is also possible to feed several varying voltages or currents into a device to obtain control by the unit which has the most critical temperature (see sections a, b, and c following).

a. Bimetallic Controls



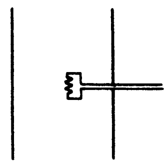


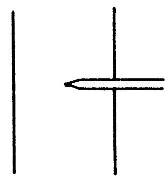
Notes: Very complete information on uses and operation of bimetals is given in Wilco Blue Book, distributed by the H. A. Wilson Co., Union, N. J. A copy is included in the Appendix.

Advantages: Any temperature or range. Good for temperature sensing of solids (i.e., head temperature of air-cooled engine). Multiple hookups are simple. May have anticipator control to hold overshoot tendencies.

Possible Supply Sources: Fenwall Inc., Ashland, Mass.; Control Products, Inc., Harrison, N. J.; H. A. Wilson (Bimetallic Materials), Union, N. J.; United Electric Controls, Watertown, Mass.

b. Thermistor and Thermocouple Controls





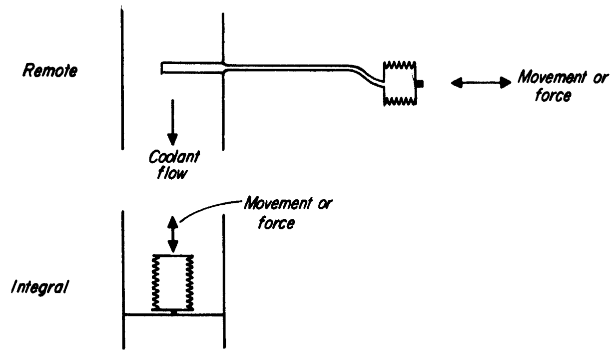
Notes: Bridge circuit and/or amplifier required.

Advantages: Variable output for variable control. Multiple control. Fast response range - 100°F to 300°F. Contact head temperature possible for air-cooled. Extremely small, very sensitive (thermistor).

Disadvantages: Additional components needed.

Possible Supply Source: Fenwal Inc., Ashland, Mass.

c. Vapor-Pressure Controls



Notes: Considerable information available from Bridgeport Catalogs (see Appendix).

Advantages: Considerable force and motion available. Very simple operation. Air or liquid coolant proportional control or on-off.

Disadvantages: Complicated hookup for mechanical control if more than one sensor is needed.

Possible Supply Source: Flexonics Corp., Maywood, Ill.; Fenwal Inc., Ashland, Mass.; United Electric Controls, Watertown, Mass.; Robertshaw-Fulton, Bridgeport Thermostat Division, Milford, Conn.

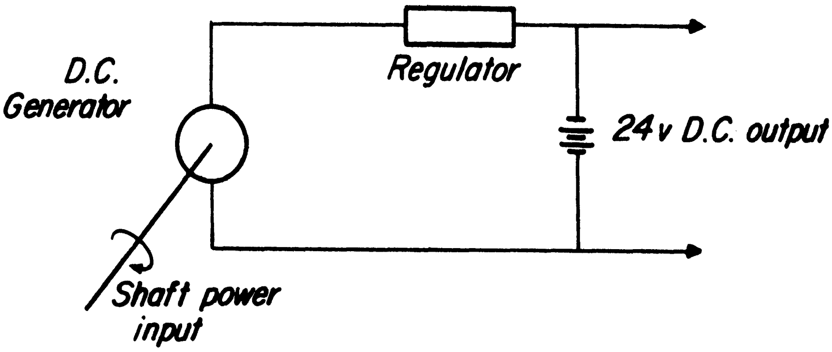
2. CONTROL POWER SOURCES

The temperature-sensing device is not directly capable of control without some means of power amplification. An exception to this may be the vapor-pressure thermostats which can do considerable work. Usually a small voltage or current change is received from the temperature device and power amplification can be supplied electrically. This is the simplest method in the case of off-on controls. For variable control systems, it is usually better to use hydraulic or pneumatic power as it affords a greater variety of simple mechanisms for this type of action. Thus if the original sensing device were mechanical (bimetallic or vapor pressure) or electrical with enough force to operate a pressure- or flow-regulating valve, complicated control systems are avoided.

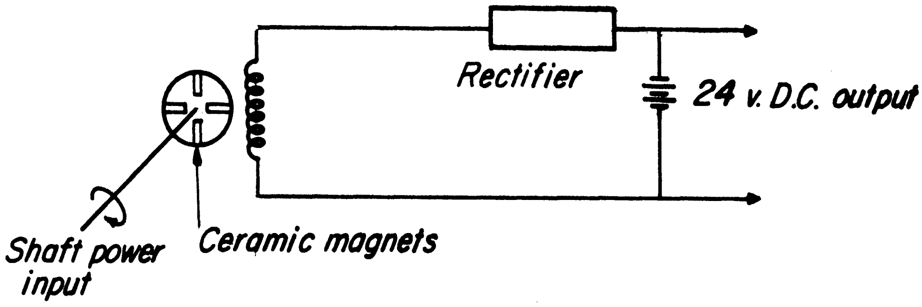
A few sources of electrical power are shown and are common items on mobile equipment. The Syncro alternator is expected to appear as standard equipment on a leading automobile manufacturer's equipment in 1960.

Hydraulic and pneumatic systems of many types are covered in Conway's book,[12] which includes flow and pressure controls, valves, servos, and electric control systems. No design details or equipment construction are given, but the principles and operation of fundamental mechanisms are explained very simply (see the following page, section a).

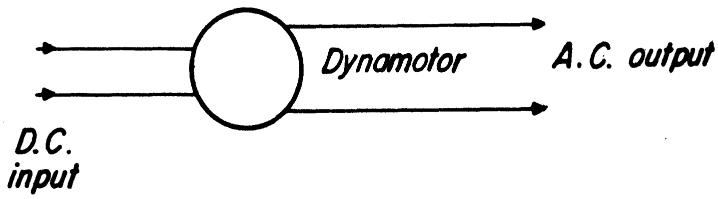
a. Electrical Control and Controller Power Sources



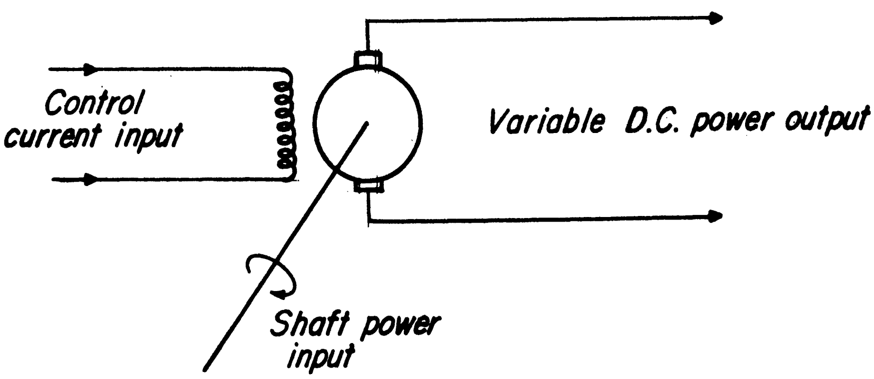
Notes: Normal power supply in all vehicles.



Notes: This generator is being developed by Syncro Products, Oxford, Mich. The construction is a salient pole generator incorporating resonant circuits which are magnetically controlled. The generator has permanent ceramic magnets and no brushes. Present prototype can deliver 70 amps at 15 volts, constant duty with a maximum output of 2100 watts. The unit measures 3 in. deep x 8 in. diameter with a weight of 15 lb. Efficiency is considerably better than a standard generator. May be used on 24-volt system by changing connections (see Appendix).



Notes: 400-cps output. Common in military equipment. This unit has the usual motor generator losses.



Notes: This unit is similar to a large power amplifier. Low input signals can control power output which can operate large power devices.

b. Vapor Pressure*

There seems to be a distinct possibility that the use of vapor-pressure control may be extended beyond the present applications. Vapor-pressure control is now mainly limited to its use as bellows actuators which adjust valves or operate switches. These systems require low forces and are reliable operating devices.

Much more force is available than is indicated. An examination of vapor-pressure-temperature curves will show this (see Appendix, Bridgeport Company). Common charging liquids, such as the Freons, increase in pressure in a straight-line relationship on semi-log coordinates. At 14.7 psi, F113 (trichlorotrifluoroethane) has a b.p. at 118°. At 54 psi the b.p. rises to 200°F. Thus a temperature change of 82° increased the pressure 39 psi. An additional increase in temperature of 82° to 282°F increases the pressure an additional 91 psi, making a 130-psi increase.

These pressures can do work. A piston, depending on the diameter, then has a force of pressure x area with a stroke limited by the liquid charge and the mechanism. This is entirely different from the gas pressure-volume-temperature relationship. Changes in volume are independent of the pressure and temperature.

There appear to be several difficulties which may be the reason for the limited use of vapor-pressure systems to this date. Since considerable pressures are present, a problem appears in attempting to contain the gas. Flexible bellows are either small or, if they are as large as 5 in. or so in diameter, they are limited in the maximum design pressure. To have effective control of such items as variable-pitch fans and variable-pitch pulleys, considerable force will be required. This means bellows of large size, say 6 to 15 in. OD to work in any reasonable range. In these sizes bellows cannot take the pressure.

A partial investigation along this line indicates the possibility of containing the liquid gas in butyl rubber containers which are then enclosed in a piston. Rubber is permeable to such gases; the degree of permeability varies with the type of rubber (including the filler), rubber thickness, type of gas, and the pressures. The metal piston enclosure would reduce leakage to a satisfactory limit at all points except perhaps at the piston-cylinder joint where movement must take place. In this area, normal sealing devices such as "O" rings or piston rings along with increased rubber thickness in the container may prove the design feasible.

Again, the problem of time lag presents itself, particularly in the variable-blade fan. Use of the necessary container, rubber, steel, or otherwise, isolates the gas from the varying temperature. Improving heat transfer by fins, and circulation of air by the fan, may solve this problem.

*References: Bridgeport Bulletin No. 125 (see Appendix); Robertshaw-Fulton.

3. CONTROLLERS

a. Hydraulic Pumps and Motors[12]*

Hydraulic pumps and motors lose power by friction loss in the moving fluid and mechanical parts and by energy dissipation due to leakage. The latter is a factor in the term "Volumetric Efficiency."

The volumetric efficiency decreases with an increase in operating pressure but the overall efficiency tends to increase. This indicates that high-pressure systems are the more desirable from an efficiency standpoint. In any case, most of the literature indicates a maximum of 80% efficiency where a pump and motor may be used for a fan drive.

For such use as a fan drive, either or both of the pump and motor units may be variable to provide speed control, or a control value may be used with non-variable units.

Piston pumps may operate at a pressure as high as 10,000 psi, gear pumps to 2500 psi, and vane pumps to 1500 psi, using oil as the fluid.

Hydraulic systems are very popular in the control field, and components are available in many varieties.

For an economical system, the oil system should be integral with a presently available oil supply; otherwise separate pumps, sumps, temperature-warning devices, and additional maintenance will be required. All vehicles will have at least the lube-oil supply available while others may have additional hydraulic systems for gun control or other purposes which would be suitable.

The advantage of the lube-oil supply lies in its availability on all vehicles and also in its capability of carrying some heat load. The hydraulic supply present on other vehicles has the advantage of a more suitable oil for control cylinder and pumps, but may not be able to take a heat load such as a fluid-drive coupling would generate.

Physical size and capacities: 6 x 12-in. output, 24 hp; weight 50 lb.

b. Clutches

(1) Friction Clutches**.—Friction clutches are essentially no-slip devices. Slippage is limited to brief periods of engagement while a load is brought up to

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*Sources: Many manufacturers produce products in this line. Some products and manufacturers are included in the Appendix.

**References: Design handbooks. Sources: Rockford Clutch Div., Borg-Warner, Rockford, Ill.; Dana Corp. (Spicer), Toledo 1, Ohio; Formsprag Co., Warren, Mich.; Twin Disc Co.

the driving shaft speed. When used within the maximum torque rating of the clutch, no slipping occurs and the unit is 100% efficient.

Design considerations use the maximum torque as the basic design criteria with speed entering as part of the dynamic balance problem.

Heat generation is determined by the cyclic rate and nature of the load (Section I).

Operation may be by mechanical, hydraulic, electrical, or centrifugal means. Several manufacturers produce electric clutches and many hydraulic friction clutches are used on a multitude of machines. A good source of this type of clutch would be the automotive suppliers. Use of a presently produced clutch over one of special design would greatly decrease cost.

Clutch engagement by centrifugal means is a very applicable means of supplying a protective device for fans when they are submerged while fording. Centrifugal force increases as the square of the speed as does the needed torque to drive a fan. It is, therefore, possible to have the maximum slip torque of the clutch closely follow the fan torque with a designed margin of safety. Exceeding the clutch torque, which would occur on entering the water, would cause the fan to slip and reduce the driving torque to some small initial amount imposed by springs.

Physical size and capacities: A 6 x 6-in. unit can handle 100 ft-lb. Size includes operating mechanism for engagement.

(2) Fluid Couplings*.—Fluid couplings are ideally suited to high inertia loads. Basically, they are constant torque devices, but may be altered in design to produce torque multiplication (torque converter) or a decrease in torque with increased slip (reservoir coupling). The latter condition would be preferred in fan-drive design as a protective device.

Fluid couplings are useful for fixed or variable-drive applications. Losses do occur under slip conditions and this must be considered in the economy of the design as well as in the problem which then occurs in the additional heat load. Minimum slip can be limited to about 5% of the input speed.

As stated in the discussion, a 60-hp fan reduced to 2/3 speed by fluid-coupling slippage must absorb 9 hp in heat. This is about 6700 watts or 380 Btu/min. Considering the AOS-895-3 engine, 2160 Btu/min is rejected into the cooling oil at 2800 rpm, full throttle. Assuming that 2/3 fan speed will satisfy the engine-cooling requirements at 2/3 full throttle (2800 rpm), the engine heat rejection should be about 2/3 x 2160 or 1440 Btu/min. This is now 720 Btu/min less than the oil cooler must handle at full throttle. The problem is—can the oil cooler dissipate the 380 Btu/min (in addition to the 1440 Btu/min) created in the hydraulic coupling with the air flow through the cooler reduced to 2/3 its normal value? The answer lies in the heat-transfer characteristics of the

*References: Edmonson, Glenn V., Hydraulic Machinery, Ann Arbor, Mich., 1957.
Sources: Twin Disc Clutch Co., Racine, Wis.

cooler and the likely safety factor which is in the design of the cooler itself.

Some of the oil-cooler load may be relieved by heat dissipation from the coupling directly to the air. To aid in this heat transfer, cooling fins may be incorporated into the design. If the oil cooler can handle the additional heat load, the fan drive may be controlled by a device which senses engine temperature only.

Basic Equations:

$$\text{hp input} = \frac{TN_1}{5250}$$

$$\text{hp output} = \frac{TN_2}{5250}$$

$$\text{hp loss} = \frac{T(N_1-N_2)}{5250}$$

$$\text{Eff} = \frac{N_2}{N_1} \times 100 = \% \text{ (constant torque only)}$$

$$\text{Capacity} = f(D^5)$$

$$\text{Capacity} = f(N_1^3)$$

$$\text{Slip} = f(W)$$

$$F = \frac{W}{g}(V_2-V_1)$$

$$F_t = \frac{W}{g}(V_2 \cos \gamma_2 - V_1 \cos \gamma_1)$$

$$T = \frac{W}{g}(R_2V_2 \cos \gamma_2 - R_1V_1 \cos \gamma_1)$$

N_1 = driving rpm

N_2 = driven (fan) rpm

D = rotor diameter

W = mass of fluid in active working circuit

F = fluid force

F_t = effective fluid force producing torque in coupling

V_1 = mean velocity of fluid W leaving driven member

V_2 = mean velocity of fluid W entering driven member

R_1 = effective radius of fluid W leaving driven member

R_2 = effective radius of fluid W entering driven member

γ_1 = angle of fluid velocity from tangent to R_1

γ_2 = angle of fluid velocity from tangent to R_2

Physical size and capacities: Fluid couplings are larger than the comparable friction clutches including their actuating devices. A 12-in.-diameter x 5-in. coupling can handle 50 hp at 2400 rpm, and a 10-in. diameter will handle 50 hp at 3600 rpm. It is very likely that a variable speed coupling can be designed to fit within the hub diameter of an air-cooled engine fan without difficulty.

(3) Eddy Current*.—The discussion of the performance of the eddy-current coupling parallels the fluid-coupling discussion. Slip losses and output torque equalling the input torque are similar to that of the fluid coupling. The method by which power is transferred is, of course, entirely different.

Basically, a d-c current is passed through a coil which builds up magnetic flux in an iron field core surrounding the coil. The magnetic field, to complete the circuit, passes through a second iron path (drum) which can rotate relative to the first. Then movement between the two units induces eddy currents in the drum. The eddy currents in turn develop a secondary field of magnetic lines of force which tend to force the drum to follow the movements of the field core. The tendency of one to follow the other is dependent on the strength of the magnetic field created by the coil and the difference in the speed of the two parts.

To transfer torque through the coupling, the field and drum must slip relative to each other. Minimum slip is about 5%.

The field coil may be a rotating member or stationary member. The rotating field receives the excitation current through slip rings. The advantage of having a rotating field coil is the fewer number of air gaps in the magnetic circuit, requiring less current excitation. The advantage of a stationary field coil is the absence of slip rings.

Physical size and capacities: Estimates indicate the tank fans may be driven by eddy-current coupling which could fit in the fan hub.

(4) Magnetic Particle**.—This is a constant-torque clutch similar to the eddy current and fluid coupling. The operating principle is somewhat similar to the eddy-current clutch. A flux is created between two parts, but the air gap is partially filled with magnetic particles forming chains which are sheared when the clutch slips. The heat generation will then occur in the particles. This differs from the heat generation in the drum of the eddy current clutch and forces a more difficult heat-transfer problem into the design.

Clutches of this type are generally limited to applications where slippage occurs infrequently or for short periods. Some magnetic particle clutches are in use on European automobiles; however, the success of the design is not known.***

Physical size and capacities: Equal to or greater torque per unit size than the eddy current. Limited slip clutch 11 x 6 in. capable of handling 1000 to 2000 in.-lb. Maximum heat generation limited to 500 watts for one of the intermittent duty clutches produced.

*Sources: Dynamatic Div., Eaton Manufacturing Co., Kenosha, Wis.

**References: Lear, Inc., Grand Rapids, Mich. Sources: Lear, Inc.

***Smiths-Magnetic Particle Clutch-"Selectroshift."

c. Variable Pulley*

Variable pulleys have become quite popular on small-horsepower machinery. They are used on drill presses, wood lathes, and band saws. One common control varies the distance between driving and driven centers, forcing a spring-loaded pulley to change the effective pulley diameter. A second type controls the space between the pulley faces of one pulley, the second pulley being spring-loaded. The belt follows the diameter of the adjusting pulley and the takeup spring of the second pulley then adjusts the belt tension. Two similarly sized pulleys of this type vary the speed as the square of the diameter because as one pulley increases diameter, the other decreases accordingly. A third type varies the faces of a single pulley with the belt tension taken up in an idler pulley.

To accomplish any great ratio of change in pulley diameter, say 2:1, the angle of the pulley faces must be small or the pulley must be slotted to accommodate considerable side travel (see Fig. 17).

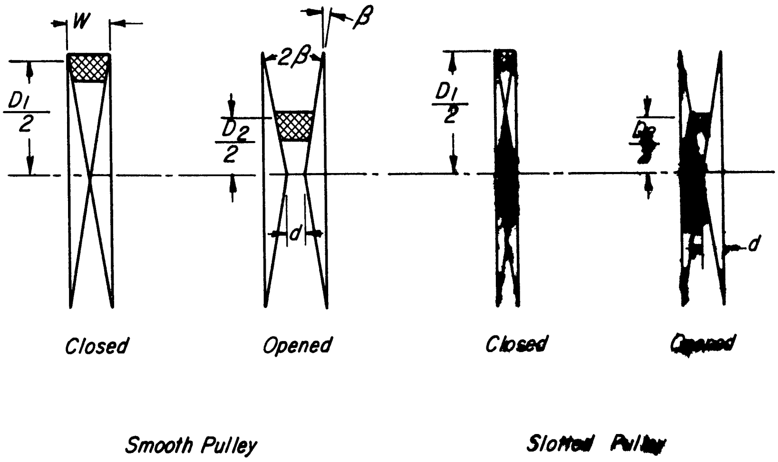


Fig. 17. Types of variable-speed pulleys.

$$\text{Tan}\,\frac{\beta}{2} = \frac{d}{\frac{D_1}{2} - \frac{D_2}{2}}, \quad \beta = 34°$$

$$\text{Tan}\ 17° = \frac{2d}{D_1 - D_2} \text{ with ratio } \frac{D_1}{D_2} = \frac{2}{1}$$

$$\text{Tan}\ 17° = \frac{2d}{2D_2 - D_2} = \frac{2d}{D_2}$$

$$d = \frac{D_2\ \text{Tan}\ 17°}{2}$$

$$d = \frac{3.5\ \text{Tan}\ 17°}{2} = \frac{1.07}{2} = .54\ \text{in.}$$

*References: Design handbooks, belt and pulley manufacturers. Sources: Allis Chalmers; Speed Selector Inc., Chagrin Falls, Ohio; The Falk Corp., Milwaukee, Wis.; Hoover Transmission Co., Williamstown, Mass.; and Lovejoy Flexible Coupling Co., Chicago, Ill.

The pulley motion must then be capable of .54-in. motion along the axis to accommodate a 2:1 ratio of speeds. The 34° angle is a common V Belt design and the 7-in.-diameter pulley is a common diameter for automotive fan pulleys.

A smooth pulley designed for 2:1 operation would necessarily need a wider belt to operate. Increasing the ratio of speed change or decreasing the belt width would require a slotted pulley.

A disadvantage of the slotted pulley is the increased belt wear caused by the belt operating on the slotted surface. No data were found to indicate the seriousness of this situation. A method which would tend to relieve this condition is to place the slotted adjustable pulley on the driving member. The belt would then ride on the slotted surface when the fan speed and thus the horsepower were at the minimum.

Multiplication of the effect of the variable pulley can be obtained as stated previously by having two variable pulleys. Then a 2:1 speed ratio (with equal diameter pulleys) can be obtained with a diameter ratio of 1.4:1.

Then

$$D_1 = 7 \text{ in.}$$

$$\frac{D_1}{D_2} = \frac{1.4}{1}$$

$$D_2 = \frac{7}{1.4} = 4.95 \approx 5 \text{ in.}$$

$$\text{Tan } 17° = \frac{d}{\frac{D_1 - D_2}{2}} = \frac{d}{\frac{7 - 5}{2}} = d$$

$d = \tan 17° = .306$ in., the necessary pulley face movement

This allows the belt width to be reduced. SAE standards for this pulley diameter recommend a belt width of 11/16 in. or .687.[13] Whether this belt is suitable depends on other factors.

Some other considerations which enter into a V Belt design are:

1. strength of belt,
2. belt speed,
3. belt size/pulley diameter,
4. type of load,
5. power,
6. distance the belt is in contact with pulley, and
7. bearing loads.

Considering a typical design and using some commonly accepted design practices,[14] the calculations may follow this pattern.

Taking 20 hp at 2880 fan rpm, which is a value found in fan tests by Beier and Weir,[9]

$$F_1 - F_2 = \frac{33{,}000\ 12\ \text{hp}\ K_s}{d_p\ N\ u}$$

F_1 = Total tension on tight belt side, lb
F_2 = Total tension on slack side, lb
K_s = Service factor = 1.1
N = rpm of small pulley (drive or driver)
d_p = pitch diameter of small pulley
u = no. of belts in the drive

$$F_1 - F_2 = \frac{33{,}000 \times 12 \times 20 \times 1.1 \times 1}{6.5 \times 2880}$$

$$= 148 \text{ lb}$$

This has some error due to the pitch diameter d_p being something less than the pulley OD. This will vary on belt size with an error of approximately 3/8 in.

Initial tension (Fi) in belt without load is

$$Fi \approx \frac{(\sqrt{F_1} + \sqrt{F_2})^2}{4}$$

If F_2 approaches complete slackness in operation,

$$Fi \approx \frac{(\sqrt{F_1})^2}{4} = \frac{\sqrt{150^2}}{4} = \frac{150}{4} \approx 38 \text{ lb}$$

To find a suitable belt for the load, manufacturers' tables are next consulted. Industrial belts require a belt of 1-in. width to handle this load with a minimum pulley diameter of 9 in. Newer automotive belt types are smaller and will reduce the belt width and minimum pulley diameter considerably.

Following also the manufacturers' recommendation of arc of contact, maximum belt speeds, and service factors, a suitable belt or combination of belts can be selected. Selection is made in these cases at the maximum load conditions with the variable pulley being considered also at the minimum diameter and arc of contact.

d. Gear Drives*

Gear systems remain the most efficient method of changing the speed-torque relationship in a power transmission. The small losses can be attributed to tooth engagement, shaft friction, and oil agitation.

Engagement losses vary with the number of teeth in the gears and the gear ratios. The type of tooth profile and pressure angle has a negligible effect. Efficiency loss varies from .1% to 2.6% of the power transmitted.[4]

In the case of planetary gear systems and other parallel power distribution arrangements, calculation of gear loss must be based on the true power being transmitted through the gears. This may be considerably higher than the input to the system due to the circulating power which may be built up in this type of design.

Many gear combinations, complete transmissions, and planetary sets including clutches and controls are available. Special designs would involve standard gears and parts with the major expense being the construction of the gear case.

Likely sources of complete gear assemblies are the suppliers of automatic transmissions for automobiles. The units are very compact, handle adequate power for the largest fans, and include actuating clutches which are hydraulically operated. Production of such items greatly reduces costs, and also solves the problem of obtaining spare parts.

4. FANS

Variable-Blade-Angle Axial-Flow Fans[1,10,11,17]

Variable-blade-angle fans have several design problems to be solved before they can become a practical method of controlling cooling-air flow. The majority of problems have already been solved by the designers of the variable-pitch propeller. Some of the differences and similarities are listed below:

Propeller	Fan
2 to 4 blades	12 blades
Pitch angle change about 100°	Blade angle change about 40°
2000 rpm	4000 rpm
Large blades, small hub	Large hub, small blades
Throttle and speed controlled	Temperature and speed controlled

*Reference: Design handbooks.[4,5,15,16] Sources: Automotive Suppliers; Gear and Transmission Manufacturers (Link-Belt Co., Chicago, Ill.).

Hydraulic, electric, or mechanical power available	Hydraulic, electric, or mechanical power available
Abrasive operating conditions	Severe abrasive operating conditions
Vibration of large masses at relatively low speeds	Vibration of small masses at relatively high speeds

Fan design has developed along the pr nciples of the aircraft wing using many of the same basic laws. The last few years have seen the fan design evolve on its own to adjust to the differences between s n le-wing design and multiple-blade design. Multiple blades cause interaction of air flow, changing the characteristics of flow. This interaction is known collectively as cascade effects.

Design of axial-flow fans as well as of pumps is based, among other things, on the specific speed.

$$U_s = \frac{(rpm)(cfm)^{1/2}}{H^{3/4}}$$

As this specific speed increases, the number of blades and the hub diameter ratio decreases. The application which calls for high volumes of flow against a low head then requires a fan with few blades, which makes a variable-blade-angle control a simpler problem. Present engine-cooling-air requirements and restrictions indicate specific speeds from 3,000 to 5,000 which require from five to twelve blades.

Design of axial-flow fans follow this general procedure:

1. select rpm;
2. calculate the specific speed;
3. determine hub ratio (diameter of hub/diameter of fan);
4. determine chord ratio (blade width/distance between blades);
5. determine number of blades;
6. choose design point based on head and capacity which will give the highest efficiency;
7. choose discharge angle for vane at mean effective diameter (this will be a range of angles if a variable-blade fan is designed);
8. calculate peripheral velocity values, axial velocity, and establish vane inside and outside diameters;
9. design blade profile.

This method is similar to that given in Stepanoff.[1] Considering item 7, which is a range of angles in the variable-blade-angle fans, high efficiencies have been obtained at a wide variation of blade-angle settings. One fan test by Schlimbach[1] varied the blade discharge angle from 11° to 40° maintaining efficiencies near 80% while air flow was reduced as much as 75%. Tests by Keller[11] showed similar results.

One method to provide fail-safe and simple operation is to use a torsion-bar type of suspension of the fan blades. This eliminates one of the two bearings needed to support the blade as well as any springs which would be necessary for high blade-angle return in case of operational failure.

To provide a more complete meaning to the fan curves and fan laws which appear throughout this report, it would be well to mention that they are based on an equivalent orifice or point of rating. The curves are developed through fan tests where the restriction of the duct is changed to provide various degrees of flow resistance. Any point on this curve can then be a point of rating. The fan laws apply to a fixed point of rating or, as it is sometimes called, a fixed orifice.

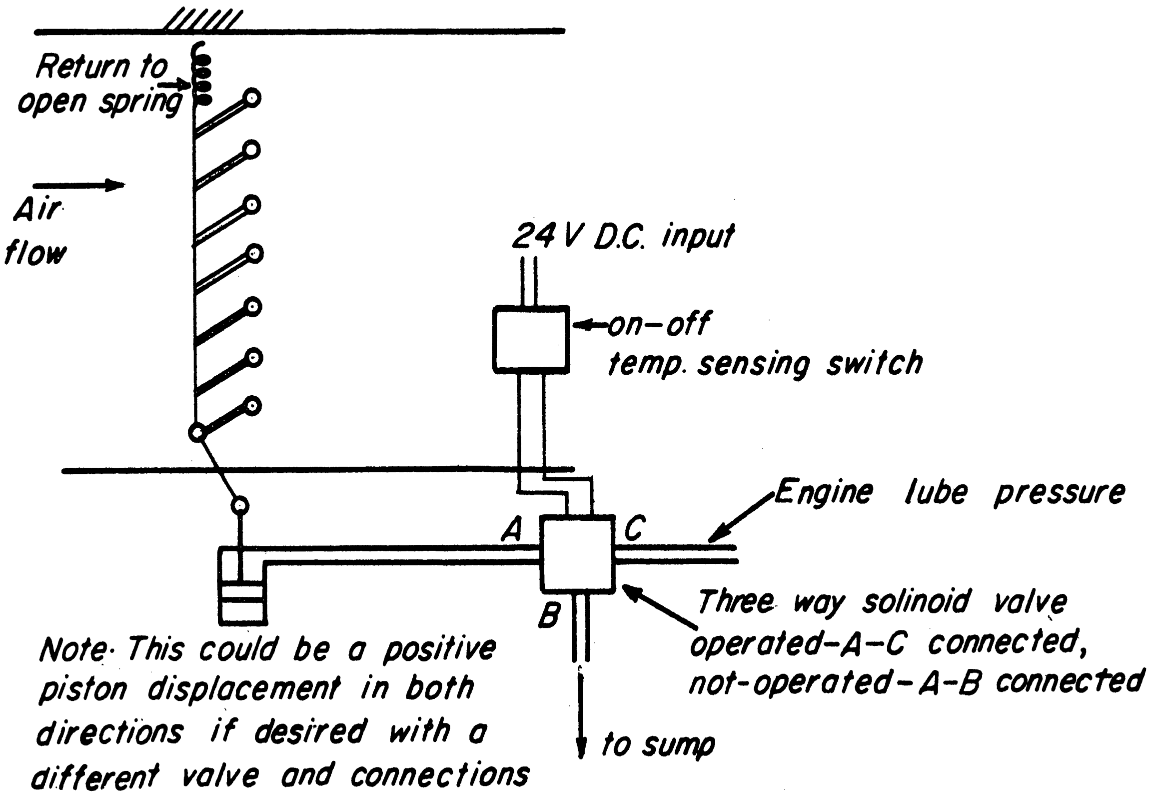
III. COOLING SYSTEMS—DESCRIPTIONS AND EVALUATIONS

A. Flow-Restriction Control

Five basic designs are incorporated in this section. Two are two position-louver controls actuated by electric and hydraulic power. The other three are variable louver positioning controls. Many modifications of these types can be used depending on the components selected to make up the system.

Regulation of air flow only in this manner is not recommended but can be a very good method of additional control in cold climates. An exception to this recommendation may be the case of auxiliary engine power units or small vehicles with fans using one or two horsepower, particularly those equipped with centrifugal cooling fans.

1. LOUVER SYSTEM
   HYDRAULIC-ACTUATED—ELECTRIC CONTROL
   TWO STEP



GENERAL DESCRIPTION: Air flow is controlled by restriction with open or closed louver conditions only. Low temperature closes electrical contacts operating solenoid. Oil pressure of lube system then operates piston closing louvers. Open circuit allows spring to open system, dumping cylinder oil back to sump. Operation may also be pneumatic if air pressure is available.

ADVANTAGES: Positive air flow in case of failure—very good as additional control in cold climates to supplement for control.

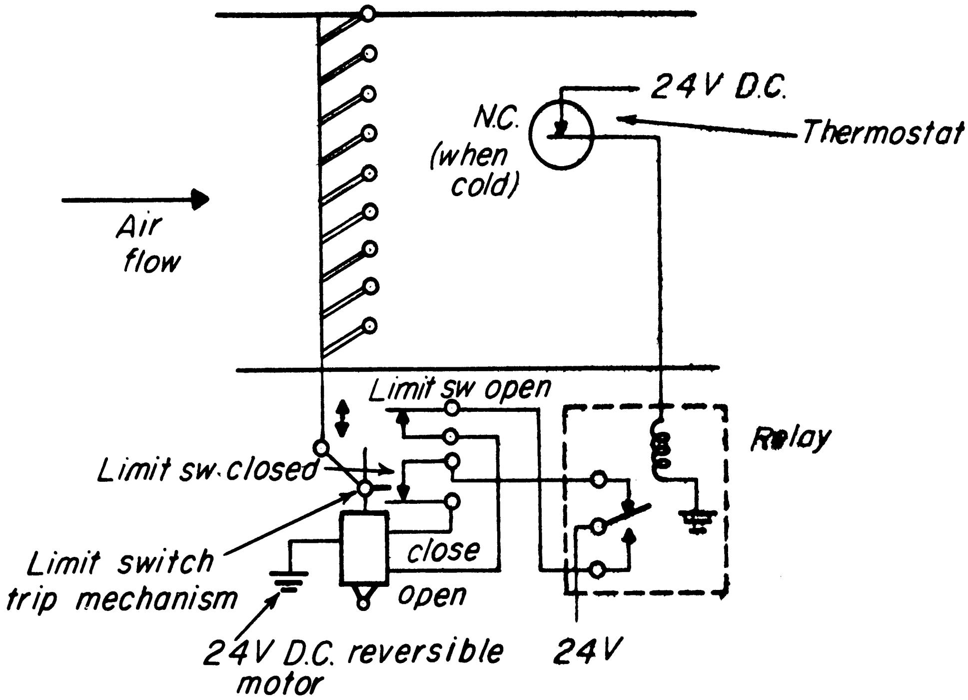
DISADVANTAGES: Power loss with any restriction.

DESIGN INFORMATION: See pages 2-4, 27-31.

AVAILABILITY: Made to order from standard parts.

CLASSIFICATION: Ultimate—this is an ideal system for additional control to supplement fan control in extremely cold climates.

2. LOUVER SYSTEM
TWO-STEP ELECTRIC



GENERAL DESCRIPTION: This is an open or closed louver device; the operating temperature is determined by the temperature range in which the thermostat contacts operate. Limit switches control the open and closed positions.

ADVANTAGES: Simple—reasonable cost—reliable—very good additional control to supplement fan control in cold climates.

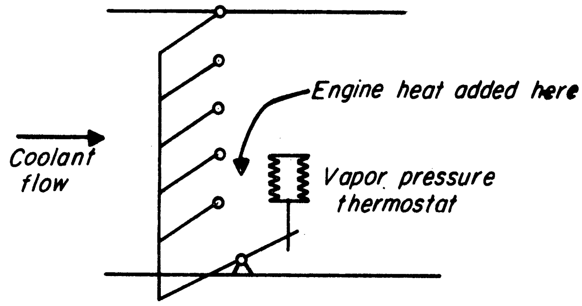
DISADVANTAGES: Power loss with any restriction.

DESIGN INFORMATION: See pages 2-4, 27-31.

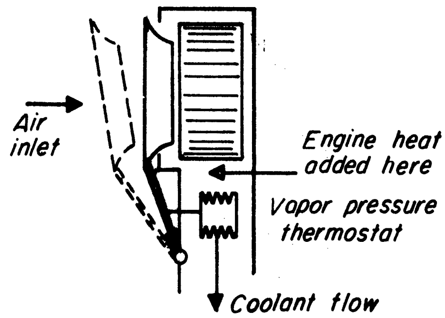
AVAILABILITY: Made to order from standard parts.

CLASSIFICATION: Ultimate—an additional cooling control in extremely cold climates.

3. LOUVER SYSTEM
DIRECT VAPOR-PRESSURE CONTROL
VARIABLE RESTRICTION



TYPE I



CENTRIFUGAL FAN CONTROL
ON THE VW ENGINE
TYPE II

GENERAL DESCRIPTION: Air flow is controlled by changing restriction to flow. Vapor-pressure thermostat is directly connected to louver system. No other connections or equipment are necessary.

ADVANTAGES: Simplicity—low original cost—very good as an additional control—this system is the least expensive of the louver controls.

DISADVANTAGES: Power loss at any restriction.

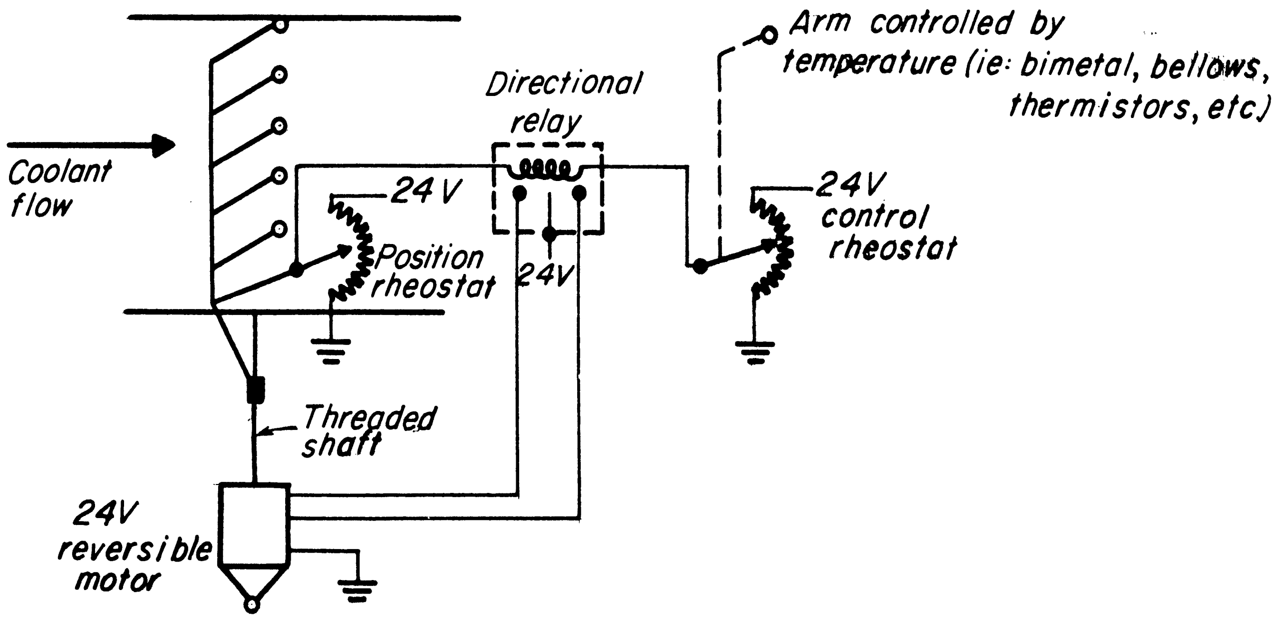
DESIGN INFORMATION: See pages 2-4, 31.

AVAILABILITY: Made to order.

CLASSIFICATION: Satisfactory—for a single control on centrifugal fans which use little power—use on inlet only—may be used with axial fans if fan is controlled or power loss is not important.

ULTIMATE: As an auxiliary control.

4. LOUVER SYSTEM
VARIABLE ELECTRIC



GENERAL DESCRIPTION: The sketch shows a simplified servo-type mechanism. Many variations of this type of control are possible, providing any degree of regulation that may be required. In the sketch, the temperature controls the voltage on the control rheostat arm. If this voltage is different from that of the position arm on the louvers, current flows, actuating the relay. This places 24 v on the motor which changes the louvers and the position arm until the position and control arms are corresponding. When in similar positions, the voltages are equal, current flow stops through the relay and the motor is stopped.

ADVANTAGES: Completely variable—large amounts of power available.

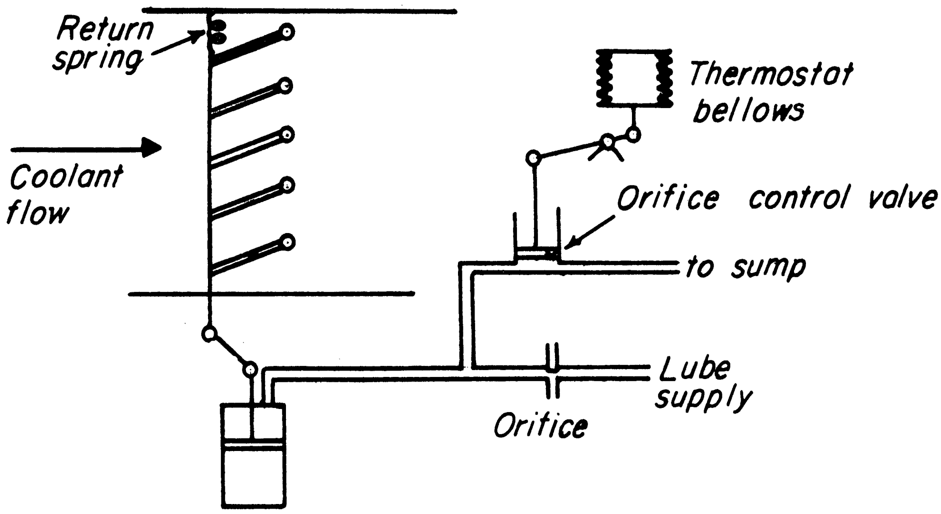
DISADVANTAGES: Power loss in the control circuit (this can be overcome by other type of controls). Fan power loss with any restriction.

DESIGN INFORMATION: See pages 2-4, 27-31, and Chestnut and Meyer.[6]

AVAILABILITY: Made to order from standard parts.

CLASSIFICATION: Satisfactory—for centrifugal fans which use little power—use on inlet only—may be used with axial fans if fan is controlled or power loss is not important.

5. LOUVER SYSTEM
VARIABLE HYDRAULIC



GENERAL DESCRIPTION: Position of piston-actuating louvers is controlled by oil pressure. Oil pressure is determined by position of restrictive valve which is controlled by temperature-actuated bellows. Many modifications of this design are possible.

ADVANTAGES: High power capacity.

DISADVANTAGES: Power loss with any restriction to air flow—some bleedoff of lube oil required for control.

DESIGN INFORMATION: See pages 2-4 and Conway.[12]

AVAILABILITY: Made to order from standard parts.

CLASSIFICATION: Acceptable—on small centrifugal fan. Satisfactory—as an additional control in extremely cold climates.

B. Fan-Drive Control

All designs in this section change air or liquid coolant flow by control of the pumping mechanism drive. This type of control can be used on any system. Brief comments on the drives can be summed up as follows:

On-off Control:

- Electromagnetic friction clutch—simple, compact.
- Pneumatic or hydraulic friction clutch—simple, compact.

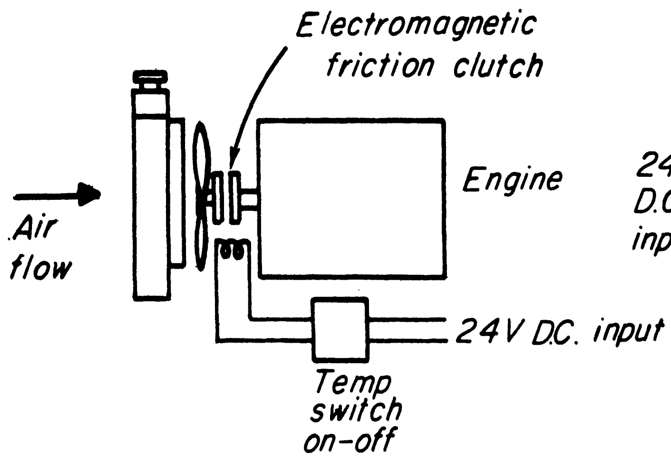
Three-Speed Mechanical:

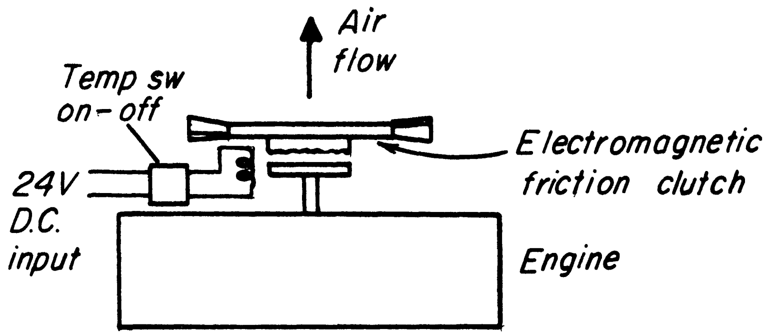
- Complex—high initial cost.

Variable:

- Pulley—very good on belt drives.
- Slip types:
  - Viscous fluid—compact, but durability should be investigated.
  - Eddy current—very simple unit, no wear.
  - Magnetic particle—simple no-slip at rated torque 1:1 ratio.
  - Hydraulic fluid coupling—well-established type, fairly simple.
- Pump-Motor Combinations:
  - Electric—bulky, expensive, and complex.
  - Hydraulic—complex—satisfactory for remote fan installations—likely maintenance problem.
- Mechanical:
  - Complex (no design included because of the remote possibility of their use—see discussion).

1. ELECTROMAGNETIC FRICTION CLUTCH
ON-OFF CONTROL





GENERAL DESCRIPTION: Electromagnetic clutches operate on d-c current with 24 v being one of the standard types. Clutch may have stationary or moving field coils. Moving coils require slip rings and have two gaps in the magnetic circuit. Stationary coils do not have rings but have four gaps in the magnetic circuit. The four-gap circuits require higher field coil current for a capacity equal to that of a two-gap clutch.

ADVANTAGES: Simple—compact—economical initial cost—efficient—may be designed for any type of fan—ideal for low-inertia stamped-steel fans in ram air-cooled vehicles—may be a high-low speed design with the addition of a gear set (see three-speed mechanical, pages 51-52).

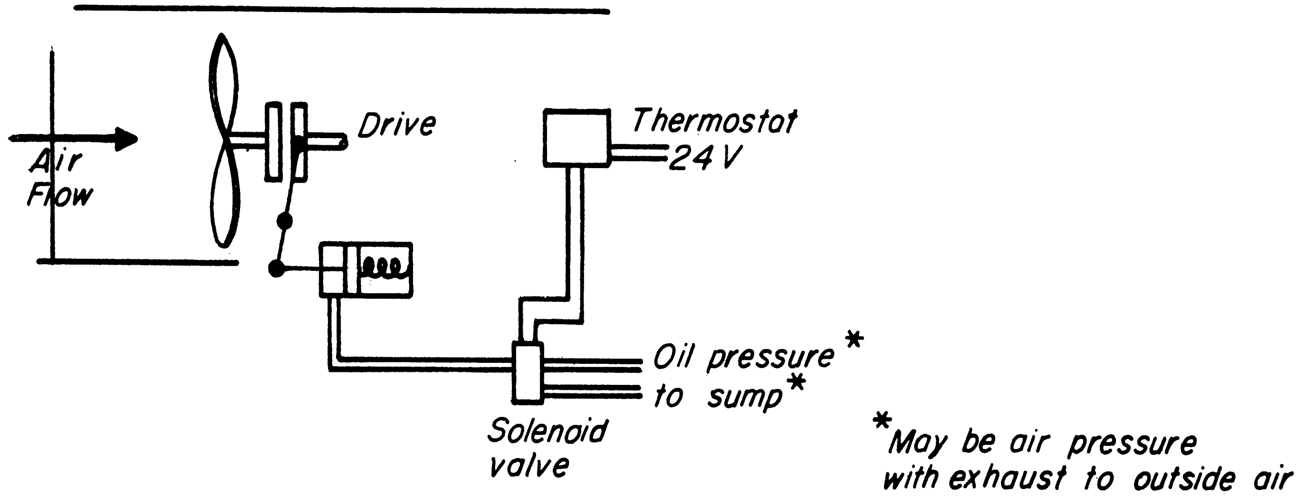
DISADVANTAGES: Not a fail-safe device unless spring-loaded engagement and magnetic disengagement.

DESIGN INFORMATION: Design components, pages 5-7, and Appendix.

AVAILABILITY: Many types available for immediate application. Others may need some modifications. Warner Electric Brake and Clutch, Eaton Mfg., Stearns.

CLASSIFICATION: Satisfactory for large truck engines.
Ultimate for small engines with ram cooling.

2. MECHANICAL-DRIVE FRICTION CLUTCH
PNEUMATIC OR HYDRAULIC—ELECTRIC-ACTUATED ON-OFF CONTROL



GENERAL DESCRIPTION: Mechanical clutch is engaged when piston receives oil under pressure. The pressure is controlled by position of switch thermostat which operates solenoid valve. Actual design incorporates hydraulic piston and clutch together on drive shaft. May operate pneumatically if desired.

ADVANTAGES: Simple—compact—no drive loss—fan loss when engaged is the same as an uncontrolled fan drive—air-pressure control eliminates oil-leakage problems—torque capacity of this type of clutch can exceed the electromagnetic types.

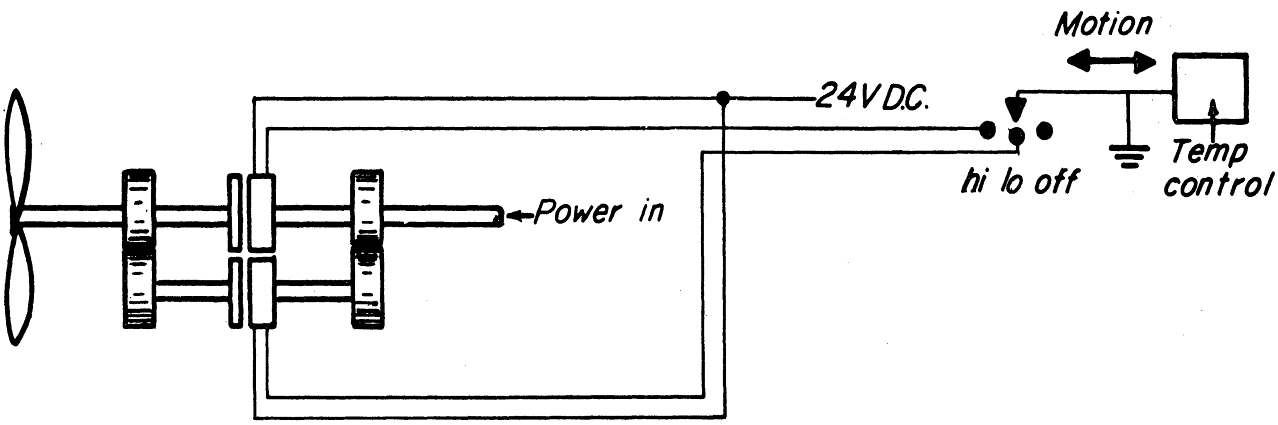
DISADVANTAGES: On-off control only.

DESIGN INFORMATION: Schwitzer drawings in Appendix.

AVAILABILITY: Many designs for engines now available—used on large off-the-highway vehicles—Schwitzer Corp.

CLASSIFICATION: Ultimate: can be used for any fan type but recommended for large truck applications and vehicles with ram cooling. Satisfactory: for closed forced cooling systems such as the tanks, if the design can endure the high cyclic engagement rates.

3. THREE-SPEED MECHANICAL—STRAIGHT GEARING



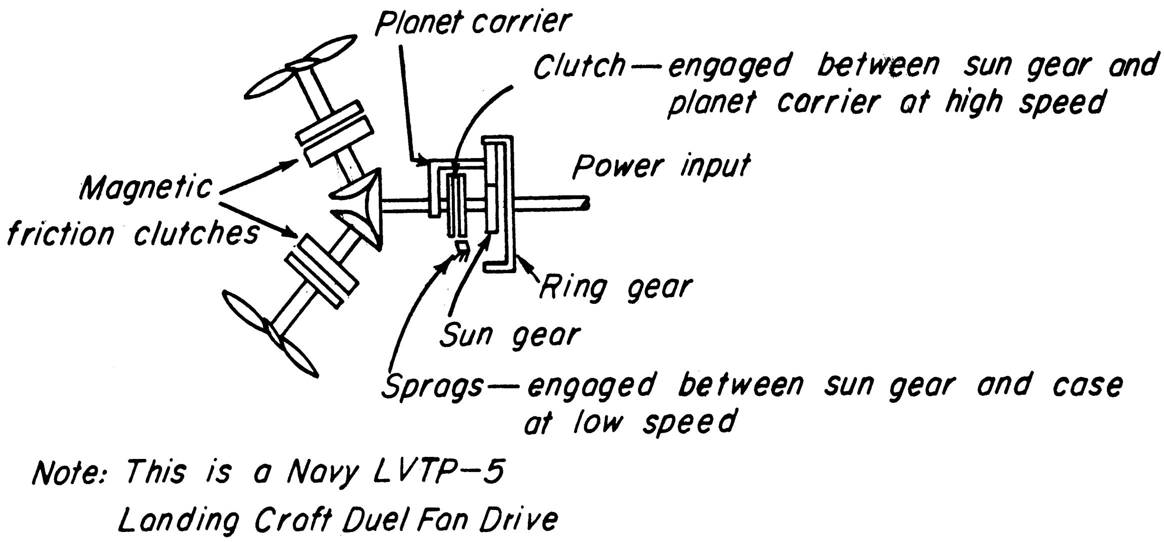
GENERAL DESCRIPTION: The speeds are controlled simply by clutches. Engagement in high speed connects the input and output shaft without gearing. Engagement of low-speed clutch and disengagement of the other clutch causes a speed reduction through the gearing. The off position disengages both clutches. This system may also operate hydraulically or pneumatically. A specific design using planetary gearing can be found on the next page.

ADVANTAGES: This type would be the most compact and economical of any change speed mechanical drive. Small losses through the gears of about 2%. Lower cycling frequency than an on-off system.

AVAILABILITY: Made from standard available parts. Mountings and gear case would necessarily be a special part.

CLASSIFICATION: Satisfactory—for large engines with forced air circulation cooling.

4. THREE-SPEED MECHANICAL-PLANETARY



GENERAL DESCRIPTION: The magnetic friction clutches control the on-off portion of the drives. For high speed the clutch is engaged by oil pressure locking the sun gear and planet carrier. This causes the output shaft to rotate at input speed. For low speed the clutch is disengaged and the carrier drives the fan through the normal planetary reduction. The center sun gear is prevented from rotating by engagement of sprags to the gear case.

Capacity is about 120 hp with the physical size of 7 in. diameter by 12 in. long. Original gearing was used in Studebaker automatic transmission (Borg-Warner).

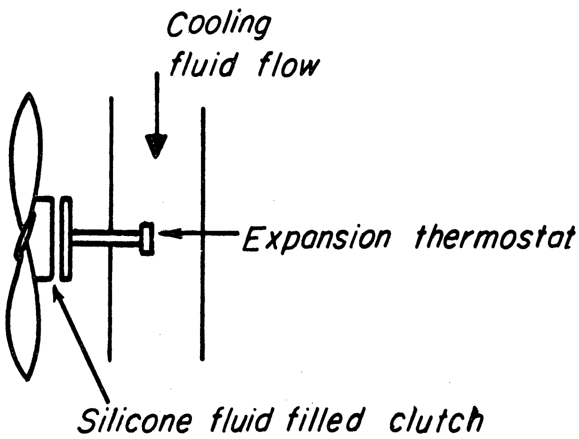
ADVANTAGES: Two speeds available at high efficiency.

DISADVANTAGES: Cost and complexity—12-in. length is also a disadvantage in most cases.

AVAILABILITY: Made to order; the basic planetary unit is probably available to suit design.

CLASSIFICATION: Satisfactory for the AV-1790 engine with dual fans—not practical for lower hp applications.

5. VARIABLE-SPEED DRIVE SYSTEM
SLIP CLUTCH, TEMPERATURE-CONTROLLED



GENERAL DESCRIPTION: Unit is essentially a friction clutch. To prevent wear of clutch parts, slippage takes place in a silicone fluid which separates the friction plates of the clutch. Varying the thickness of the fluid film varies the maximum slip torque. Thickness of film is controlled by thermostat in liquid circulation system (usually installed as part of the water pump) which determines the distance between the friction plates.

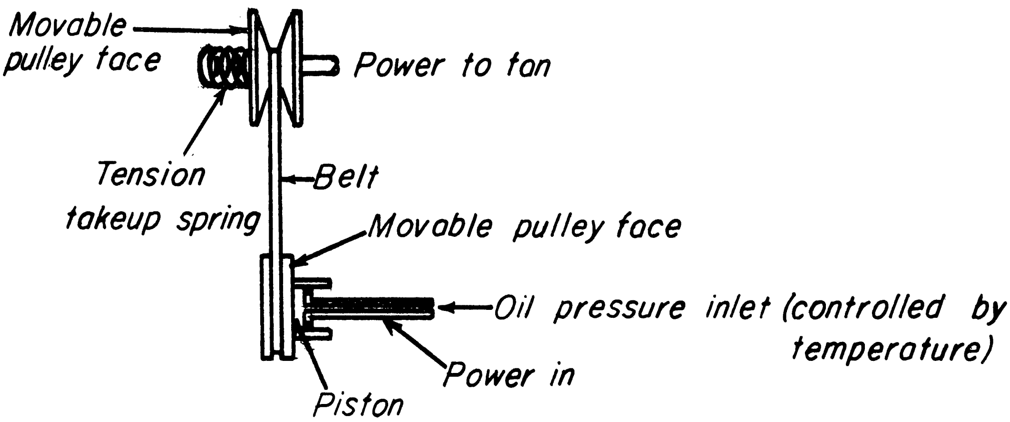
ADVANTAGE: Very compact unit.

DISADVANTAGES: Always some power loss—loss is similar to all other slip-type clutches—must have adequate heat dissipation.

AVAILABILITY: Immediately available from at least two companies for a number of engines—see Appendix for drawings—Schwitzer Corp., Indianapolis, Ind., and Eaton Mfg. Co. Pump Division, Detroit, Mich.

CLASSIFICATION: Satisfactory, if product can withstand a rigid test program.

6. VARIABLE-SPEED PULLEY



GENERAL DESCRIPTION: Speed can be varied as high as 8:1, offering adequate fan control. One pulley sheave is controlled while the second adjusts diameters through a change in spring tension. Belt remains aligned if opposed faces of the pulleys are made adjustable.

ADVANTAGES: No power loss—the variable speed may also serve as a possible control for generator if on the same belt—battery drain from starting engine replaced at high rate when engine is cold—after warmup and battery is charged, the generator rpm is reduced to lower output level.

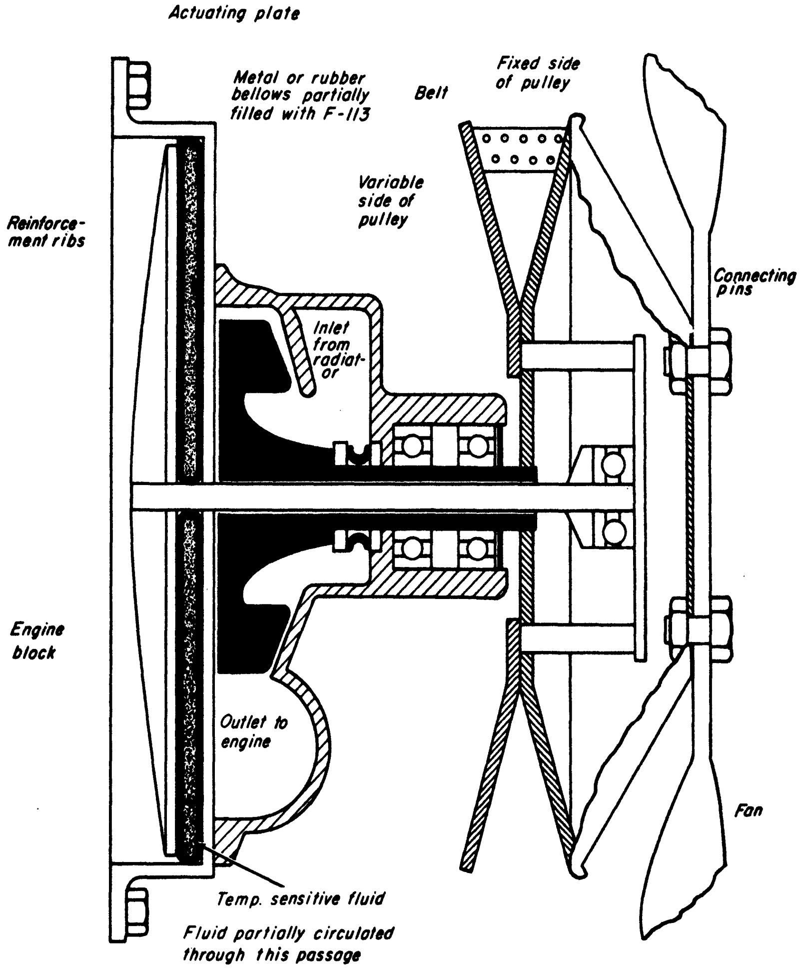
DISADVANTAGES: Those systems with additional equipment driven by the same belt would necessarily vary speed also or must be transferred to another belt.

DESIGN INFORMATION: Pages 37-38 and Appendix.

AVAILABILITY: Several companies made variable pulleys with hand- and automatic-operated controls. No automatic control systems found—some simple design work would be necessary.

CLASSIFICATION: Ultimate for ram-cooled belt-driven systems.

7. VARIABLE-SPEED PUMP AND FAN ASSEMBLY
TEMPERATURE-CONTROLLED VARIABLE-DIAMETER PULLEY



GENERAL DESCRIPTION: Pulley is separated into two halves, one of which is attached rigidly to the driving shaft and the other of which can move axially along the shaft. Control of axial movement is determined by vapor pressure in expansion bellows which is controlled by water temperature.

ADVANTAGES: Speed may be controlled within a ratio of 2.5:1 which would vary the air volume 2.5: 1 and power requirements about 15:1—no loss in control system, and power saving in both fan and pump—ratio can be squared if spring-controlled pulley is introduced on the driving end—fail-safe.

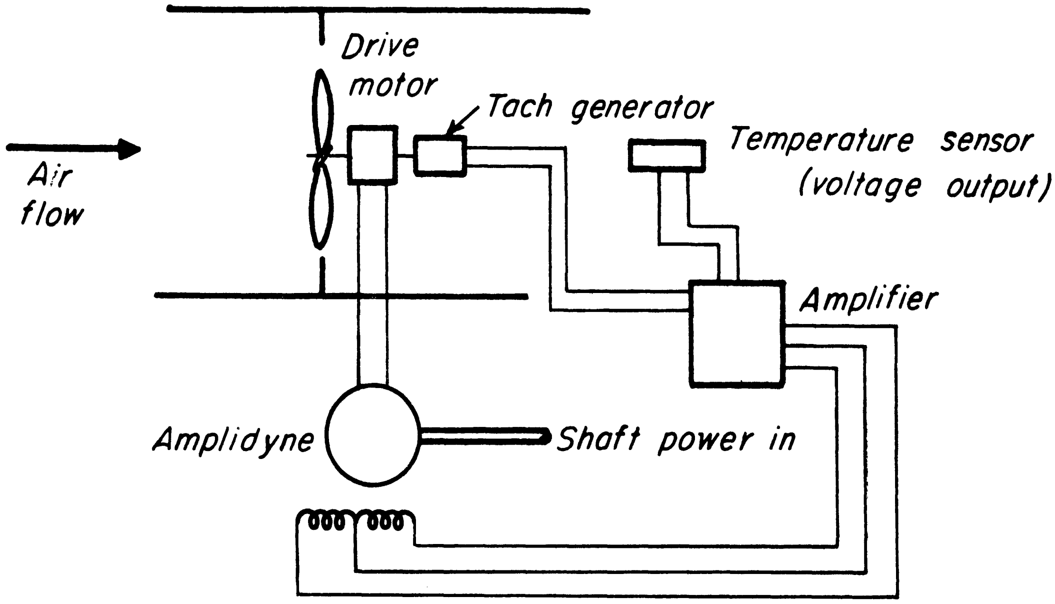
DISADVANTAGES: May require a fairly large diaphragm diameter to obtain sufficient force to operate pulley control.

DESIGN INFORMATION: See Design Section II—no similar device is known to be manufactured.

AVAILABILITY: None.

CLASSIFICATION: Ultimate for liquid-cooled belt-driven vehicles—design must be developed as no known device of this type is available.

8. VARIABLE SPEED
GENERATOR-MOTOR COMBINATION



GENERAL DESCRIPTION: Temperature sensor and tach generator feed signals into amplifier. Amplifier controls input to amplidyne which in turn controls speed of motor.

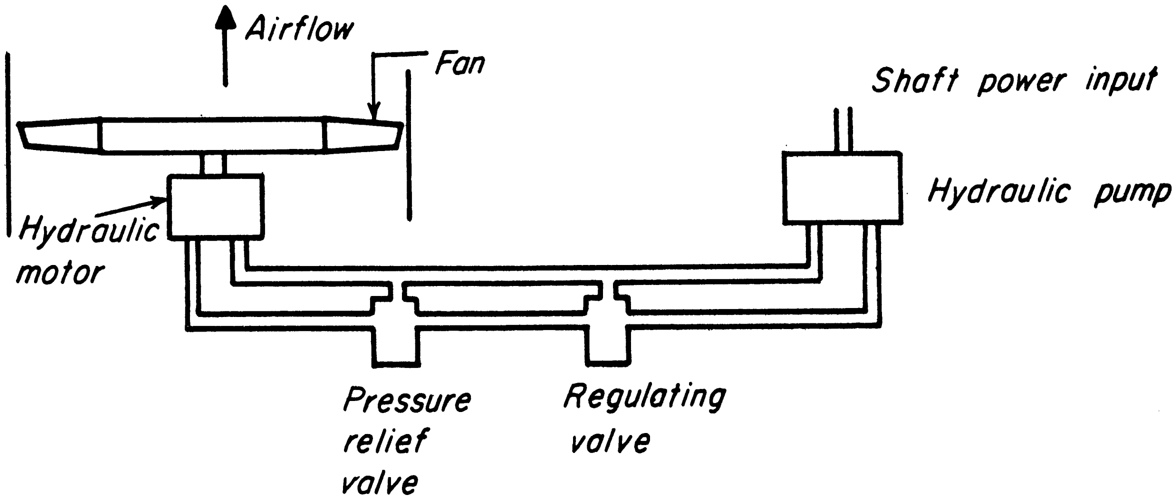
ADVANTAGES: Complete variability.

DISADVANTAGES: Low efficiency (70%)—high initial cost—considerable occupied space. The drive motor may be reduced in size if run at very high speeds, and the output reduced through a gear box to a suitable fan speed.

DESIGN INFORMATION: Chestnut and Mayer.[6]

CLASSIFICATION: Unsuitable.

9. VARIABLE SPEED
HYDRAULIC PUMP AND MOTOR



GENERAL DESCRIPTION: Hydraulic pump is driven from engine and supplies oil under pressure to hydraulic motor. Speed regulation is obtained by adjusting oil flow to motor. This can be done by control valves which results in losses or by regulating output of the pump. Two methods of pump control are given in Design Section II. Either pump or motor or both may be variable devices if necessary for greater speed variation.

This system is presently in use by the Kenworth Motor Truck Corp., Seattle. They employ a Denison TMC balanced vane pump belt driven from the engine. The fluid power obtained drives a Denison TMB hydraulic motor. Mounted directly on the shaft of the motor is the fan blade. Speed regulation is accomplished with a Denison RA relief valve.

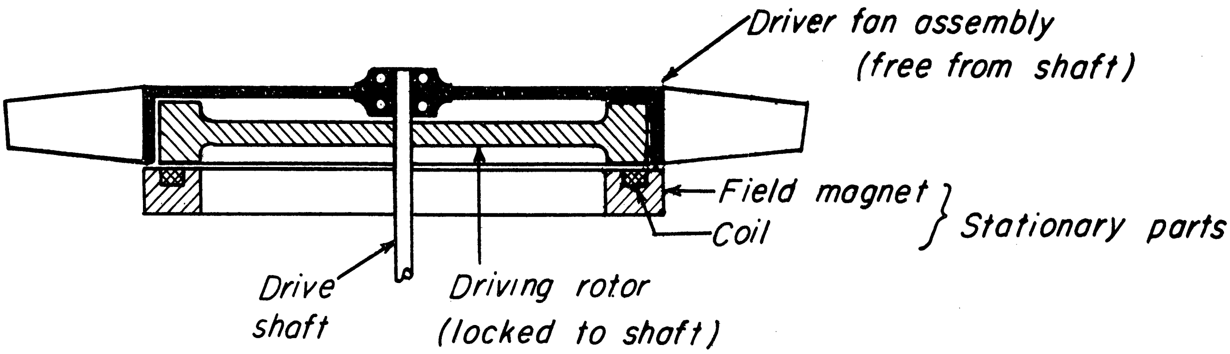
ADVANTAGES: Completely variable speed—fan can be remote from power source—very small sizes for power output; 10-hp motor in 6 x 6 x 6-in. space.

DISADVANTAGES: Low efficiency (70-80% maximum)—system is quite complex for a cooling fan drive.

AVAILABILITY: Many sources and types available.

CLASSIFICATION: Acceptable for closed cooling systems with high inertia fans, where fan position and power source position cannot be readily connected by mechanical drives.

10. VARIABLE SPEED
EDDY CURRENT



GENERAL DESCRIPTION: The fan assembly in this case is completely free from the driving shaft. Torque is transferred to fan through eddy-current effects between rotor and fan depending on the current in the stationary field coil. The control device must then control the current. This type is infinitely variable, will slip when entering water or if fan is otherwise obstructed. Heat generation can be located in the fan assembly by slotting the driving rotor. This will enable the heat to be dissipated directly to the cooling air.

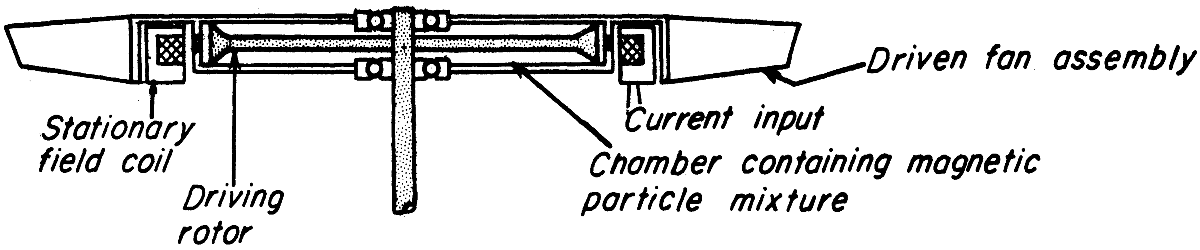
ADVANTAGES: A simple reliable control device—no wearing of parts.

DISADVANTAGES: Possible high current requirements to provide sufficient torque for a system such as this with three air gaps—this may be reduced by having coil on moving members, which would require slip rings—slip clutch type of power losses.

AVAILABILITY: Similar devices have been used, but some design and development work will be necessary—Dynamatic Division, Eaton Mfg. Co., Kenosha, Wis.

CLASSIFICATION: Satisfactory—for large tank fans. Not recommended for ram air-cooled vehicles.

11. VARIABLE SPEED
MAGNETIC PARTICLE



GENERAL DESCRIPTION: Comments on the magnetic particle clutch are the same as on the eddy-current clutch with one exception: the unit can attain a no-slip condition which is not possible with the eddy-current clutch.

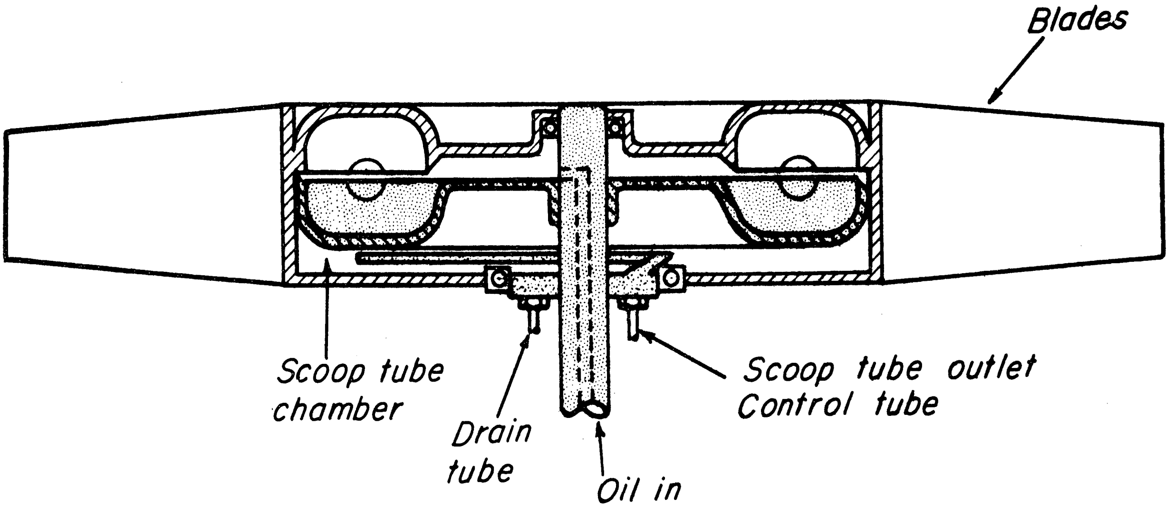
ADVANTAGES: Fairly simple device—some provision must be made for retaining the magnetic particles but sealing is not as difficult as a hydraulic system—lower current requirements than the eddy-current types.

DISADVANTAGES: Slip clutch type of power losses except when locked in. The problem of heat dissipation will probably prevent the success of this design.

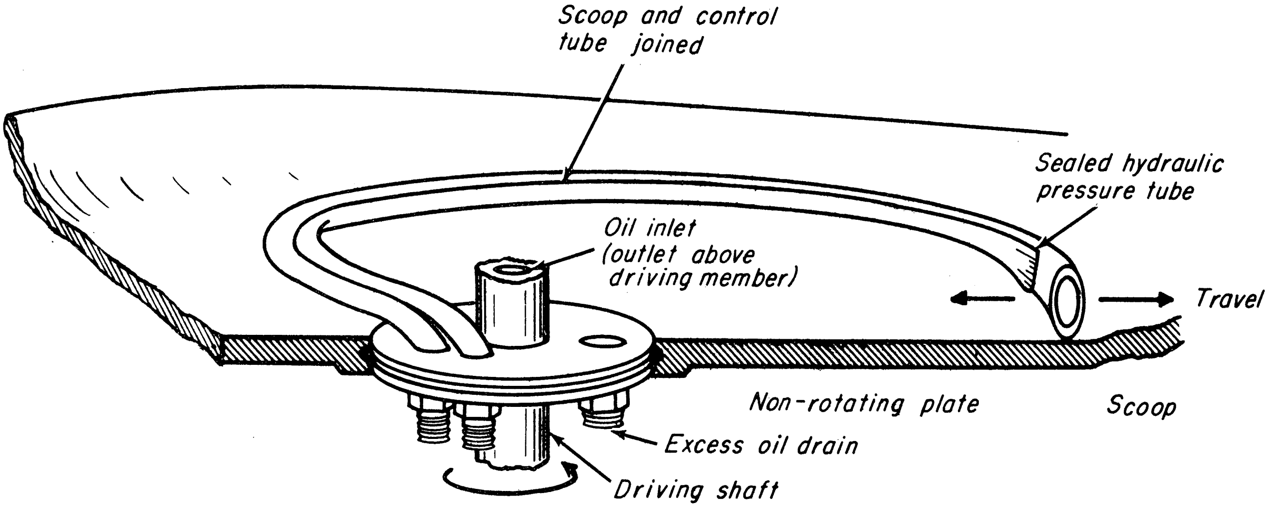
AVAILABILITY: Similar devices have been used, but some design and development work will be necessary.

CLASSIFICATION: Satisfactory for large tank fans; not recommended for ram air-cooled vehicles. This recommendation is with the qualification that the heat-dissipation problem can be solved.

12. VARIABLE-DRIVE SYSTEM
DUMP AND FILL FLUID COUPLING



FLUID COUPLING SCOOP DETAIL



GENERAL DESCRIPTION: Speed is controlled by amount of fluid in working circuit of coupling. Scoop tube varies the amount of fluid by changing the effective radius of the inlet of tube. This is controlled by hydraulic pressure variation in scoop tube control. Heat is absorbed in the flowing oil. For river fording, indicator to control can disengage fan completely.

ADVANTAGES: Efficiency good, 95% at minimum slip—15% of rated fan power at maximum loss (plus oil pump loss)—completely variable—disengages completely.

DISADVANTAGES: Always some power loss—oil-seal problems.

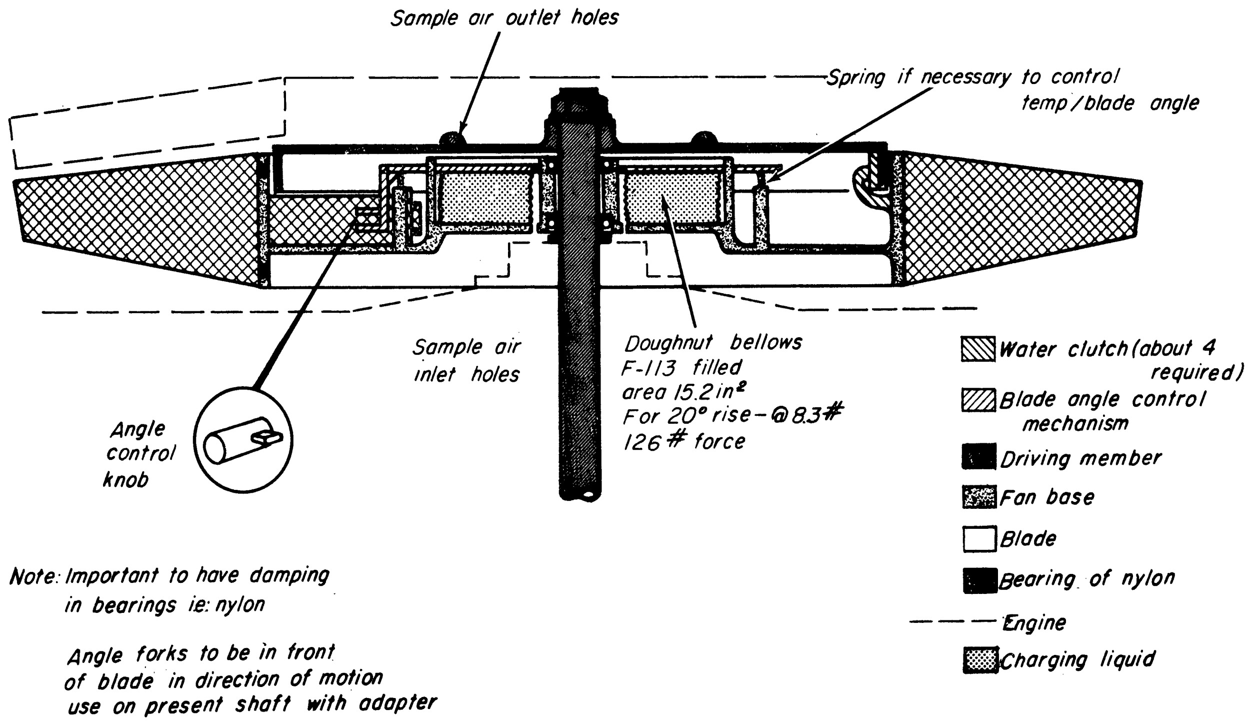
DESIGN INFORMATION: Design Section II; supply sources: Twin Disc Co., Racine, Wis.

AVAILABILITY: Some modification of present systems will be necessary—many sources and types available.

CLASSIFICATION: Satisfactory—for the larger fan installation on tanks or other vehicles with only forced fan cooling involving heavy cast fans. Not recommended for ram air-cooled vehicles or those with low-inertia steel-stamped fans.

C. Fan-Characteristics Control: Variable Blade Angle

1. BELLOWS CONTROL



GENERAL DESCRIPTION: Blades are mounted on nylon sleeve bearings and adjusted by control forks actuated by bellows piston. Air flowing by bellows determines vapor pressure in bellows and thus the blade angle. Design must include damping sufficient to prevent vibration fatigue.

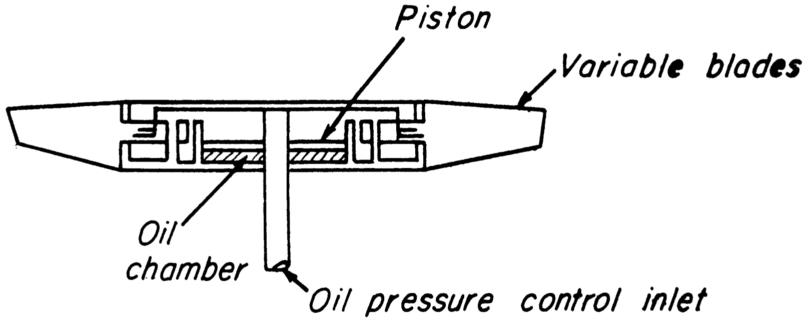
ADVANTAGES: No controls are needed—no power is lost under any condition—the blades are replaceable—noise reduction at lower blade angles—reverse flow is possible.

DISADVANTAGES: Forces needed to control pitch may be greater than that obtainable with bellows which could fit inside hub diameter. Noise reduction will be somewhat less than if the speed were reduced.

AVAILABILITY: Design and development work necessary. Bridgeport Thermostat Division of Robertshaw-Fulton does make metal bellows up to 18-in. OD with a welding process.

CLASSIFICATION: Ultimate; for large volume flow fans, in forced circulation systems.

2. HYDRAULIC CONTROL



GENERAL DESCRIPTION: Blades are mounted on nylon sleeve bearings and adjusted by control forks actuated by a hydraulic piston. Oil pressure is delivered to piston through center of shaft drive. Pressure is regulated by a temperature-controlled valve.

ADVANTAGES: Positive blade-angle control and thus accurate control of air flow—no power is lost at any condition—replaceable blades—noise reduction at lower blade angles—reverse flow possible.

DISADVANTAGES: Noise reduction will be somewhat less than if speed were reduced—leakage problems.

AVAILABILITY: Design and development work necessary.

CLASSIFICATION: Ultimate for large forced circulation systems.

IV. CONCLUSIONS

With the diversity of air-cooled and water-cooled engines now in use and future anticipated use, no one cooling control system is feasible.

For engines using small light-weight fans such as those normally driven by belts on the water-cooled-truck engines, the fan can be controlled by a magnetic friction clutch and a thermostatically actuated on-off switch. This type of control is easily adaptable to present installations. Other similar types are pneumatic or hydraulic friction clutches. These differ from the electric type only in the means of actuation. Friction clutch plates are the engaging means in all cases.

A second, more smoothly operating system for these engines would be a variable-diameter fan pulley. This allows the speed to be adjusted continuously to satisfy conditions. Variable-speed pulleys are in common usage on many types of equipment but no specific case has been found where they have been applied to control cooling on vehicles. One small automobile produced in Europe uses the system as a part of the transmission, driving each rear wheel with a belt on a variable pulley.

For vehicles which are partially cooled by air flow due to their movement, fan-speed reduction or even disengagement may not be sufficient control. For extremely cold conditions, additional control can be obtained by installation of louvers on the air inlet, either hand- or automatically operated.

Engines using only forced cooling involving considerable horsepower and fan inertia are not as well suited to friction clutch application, and in this regard clutches are third choice. Ideally, variable-blade-angle fans are the answer to cooling control where large volumes of air are pumped and heat dissipation is high. Variable blade angles will permit widely variable air flow, maintain good efficiency, and involve little modification of the engines or components. The design and modifications would be limited to the fan itself and the control equipment. This type of control may also have the advantage of heating the engine compartment under very cold conditions by reverse flow of the cooling air. In some tanks reversing flow would cause the air to pass over the muffler, giving instant heat after starting.

Another choice for the forced cooling systems is the slip-type clutch. Hydraulic, eddy-current, and magnetic particle types can be used; the magnetic particle type is the only one which can operate with a no-slip condition. This no-slip advantage can be useful only when the engine needs full cooling capacity, a condition which would be quite rare with a cooling fan which is adequate to meet engine demands under extreme heat. The advantage of a no-slip clutch then depends on the need of full fan cooling in a given design.

Electric generators and motors or hydraulic pumps and motors are generally unsuitable as drive systems because of their large losses. Slip-type clutches, such as magnetic particle, eddy current, and fluid coupling, are possible control devices despite their inefficiencies at high slip rates. This is mainly due to the nature of fan horsepower curves. Fan horsepower is sharply reduced when speed is reduced, and the maximum loss with a slip coupling is 15% of the rated fan power (i.e., fan power is directly driven). This results in considerable savings of power previously wasted in additional air flow on unregulated fans.

Power savings will depend on the type of control chosen and the operating conditions under which it is used. Direct drive operating at maximum capacity is the most efficient when maximum flow is needed. For less than the maximum flow, top efficiency is obtained at all times with variable-speed belt drives, and variable-blade-angle controls. On-off clutches are also near top efficiency at all load conditions, the loss occurring in the operating of the clutch.

Multiple-step speed drives are at maximum efficiency only when the air flow needed corresponds to one of the available speeds. An added loss is present in the speed-changing device which in gear designs is slight.

Slip clutches vary with the operating range of the vehicle. The less slip required, the less the percent of the fan power lost. It is important in this type to operate as near minimum slip as possible, and that the capacity of the fan is the least required to handle the load.

Hydraulic pump and motors have a more constant loss with the efficiency varying with the fluid pressure, action of by-pass and control valves, etc., as well as the horsepower transferred.

For the slip clutch to be more efficient than a constant-loss hydraulic pump and motor, the clutch slippage should be 30% or less. If the design contemplates the drive to be in various percentages above and below 30%, the factor of fan horsepower varying with the cube of the speed weighs heavily for the slip clutch. The slip clutch requires less power at the times when the need for maximum engine power is required—an important factor when net horsepower output of the engine is being considered.

The table following is a brief outline of some of the possible design schemes included in the report.

SUMMARY OF POSSIBLE DESIGNS

Type	Design	Use*	Classification	Adaptability	Availability	Efficiency	Control: Actuator	Control: Operation	Relative Costs (Low Nos. - Low Costs): Design**	Relative Costs (Low Nos. - Low Costs): Purchase	Remarks	Design Page Location
Air-Flow Control	Louvers	Open and Closed	Ultimate as Auxiliary Control	No Serious Problems	Parts Available	No Loss as Auxiliary Control	Electric Motor	Two Position	1	1	All flow-restriction controls should be used only for auxiliary control in cold weather operation. If used as the main control on axial fans, power loss is high. Some power saving can be obtained with centrifugal fans.	44
								Completely Variable	1	2		46
							Vapor Pressure	Completely Variable	1	1		45
							Hydraulic Piston	Two Position	1	1		43
								Completely Variable	1	2		47
Fan-Drive Control	Friction Clutch	Open	Satisfactory	No Serious Problems	Available	Good	Electro-magnetic	On-Off	1	1	Mainly used for open systems. Cycling rate is more serious with closed systems and is recommended in these cases when minimum initial cost is desired.	49
							Pneumatic or Hydraulic		1	1		50
						Fair	Vapor Press. or Bimetal	Variable Speed	1	1	This is a slip clutch device similar to fluid coupling, eddy current, and magnetic particle.	53
	Mechanical	Open and Closed	Satisfactory	Space Difficulties	Parts Available	Good	Clutches	Two or Three Speed	3	5	A variety of designs available. Space limitations will probably require special design using shelf parts.	51
								Two or Three Speed Planetary	3	5	Similar to above; however, a separate design sheet is used to emphasize the planetary systems.	52
	Pulley	Open and Closed	Ultimate	No Serious Problems	Some Design Work Necessary	Excellent	Hydraulic Pneumatic Electrical	Variable Speed	2	2	A smoothly working system for belt drives.	54
					Untried		Vapor Pressure		3	2	As above, but specifically using vapor pressure. This system is untried but appears feasible.	55
	Electric	Closed	Unsuitable	Space and Design Difficulties	Parts Available	Poor	Electric Current Low	Variable Speed Motor and Gen.	2	5	Size, initial cost, and inefficiencies rule against this system for fan application.	56
			Satisfactory	No Serious Problems	Some Design Work Necessary	Fair		Eddy-Current Clutch	3	4	A very simple principle for brakes and clutches which seems to get little attention.	58
								Magnetic Part. Clutch	3	4	A relatively new type. Common in servos. Has good possibilities with solution of heat problems.	59
	Hydraulic	Closed	Acceptable	Space Difficulties	Parts Available	Poor	Electric Pneumatic Hydraulic	Variable Speed Motor and Pump	2	4	A useful drive for a remote fan, otherwise other systems appear more suitable.	57
			Satisfactory	No Serious Problems	Some Design Work Necessary	Fair	Electric Hydraulic	Variable Speed Fluid Coupling	3	3	An accepted popular clutch. Present clutches can probably be modified for variable control.	60
Fan-Characteristics Control	Variable Blade Angle	Closed	Ultimate	No Serious Problems	Special Design (Untried)	Excellent	Vapor Pressure	Variable Flow	5	7	Two methods of control are given. The vapor-pressure system is untried. Leakage and pressure problems must be solved which could result in a neat compact fan unit with built-in control. Hydraulic control follows standard design procedure.	61
					Special Design		Hydraulic Pneumatic		5	8		62
	Variable Blade	Closed	Satisfactory	Space Difficulties	Special Design	Good		Two Rates of Flow	4	6	Design requires a number of the blades to be attached to a separate concentric rotor. Blades would lock in proper position in driving fan by a clutch mechanism.	Not Shown

*This refers to type of cooling system in vehicle, open for ram air-cooled systems (automobiles, trucks, etc.) and closed for those with forced circulation only (tanks).
**The term design implying only development of unit for new vehicle or without extensive modifications of present vehicles.

V. RECOMMENDATIONS

The cooling systems can be divided into two broad groups:

I. Ram cooling systems or open systems - those in which the cooling air flow is aided by the vehicle movement.

II. Closed cooling systems - those in which the cooling must depend on fan circulation alone.

Those systems in group I can most easily be controlled by a two-step on-off fan drive or variable pulley drives. Friction clutch plates are recommended with actuation by electromagnetic, pneumatic, or hydraulic means. The electromagnetic design is the most versatile and can be controlled by one or more thermostatic switches. If the system is belt-driven, the use of variable pulleys should be investigated.

The systems in group II are necessarily divided into two subgroups.

A. Low-inertia stamped-steel fans with low hp.

B. Large-inertia cast fan assemblies.

Those in subgroup A can be controlled in a manner similar to group I. If belt-driven, variable-speed drive is recommended. Subgroup B, which includes the high horsepower fans on tank engines, should be variable air-flow devices. This can be done by varying the fan-drive speed or varying the pitch.

If future engine designs continue to use large cast fans and the numbers produced amount to two or three hundred a year, it is recommended that an investigation and development program be initiated to design a variable-pitch fan. If conditions do not warrant this development, a variable-speed fluid coupling should be used. Other slip couplings such as the eddy current and magnetic particle may also be used, but it is felt that the development program of one of these types would be more costly than that of the fluid coupling. Fluid couplings have been used for this type of application by many companies and the problems in design are well known.

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