LIQUID METERING SYSTEMS FOR LOW VOLUME SPRAYERS

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ABSTRACT

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by Stephen David Johnson

Pest control through the application of chemicals has been an integral part of crop production for nearly a century. The most common method of pesticide application for fruits and vegetables is dilute airblast spraying. Recently, however, this method in its present form has drawn fire from ecologists and a concerned public because of its possible detrimental effects to the environment. Alternate methods of pest control have been proposed and also alternate methods of chemical application have been tried. One such alternate method of application is ultra-low volume application. The method offers the advantages of reduced dosage and a reduced number of treatments needed in a growing season. One of the problems associated with this method of pesticide application is the lack of an accurate low-flow metering system. The purpose of this study was to attempt to solve this problem.

Several systems were considered and two systems were investigated in detail. A peristaltic pump system which included equipment for desurging the pulsating output was evaluated under laboratory and field conditions. The results of the tests indicated that this system could reliably accomplish the stated research objective. A gear pump system was also evaluated in the laboratory. It demonstrated equal and sometimes superior performance in all aspects of testing to that of the peristaltic pump system.

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LIQUID METERING SYSTEMS FOR
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DISCLAIMER

References to company names, products, or trademarks do not constitute condemnation or endorsement of these products, but merely convenient designations for selected equipment and materials.

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NOMENCLATURE

cpse	Centipoise (Viscosity)
fps	Feet per second (Speed)
gph	Gallons per hour (Output)
gpm	Gallons per minute (Output)
lbf	Pounds (Force)
lbf/in	Pounds per inch (Spring Constant)
ml/hr	Milliliters per hour (Output)
ml/min	Milliliters per minute (Output)
ml/rev	Milliliters per revolution (Displacement)
mph	Miles per hour (Speed)
psi	Pounds per square inch (Pressure)
psig	Pounds per square inch gauge (Pressure)
rpm	Revolutions per minute (Speed)
spm	Strokes per minute (Pumping Speed)
°F	Degrees Fahrenheit (Temperature)
°	Degrees (Angular Rotation)

I. INTRODUCTION

Pesticide application and the possible resultant environmental pollution are subjects of discussion in ladies' tea circles, college laboratories, and even in the chambers of state and national government. At this time there are bills in both the House of Representatives (Obey, 1971) and in the Senate (Nelson, 1971) which if passed would exact stiff regulatory measures on the use and distribution of chemicals including pesticides. Similar legislation has already become law in some provinces of Canada. This national consciousness is due to the crusading of such individuals as Ralph Nader (Nelson, 1969) and Rachel Carson (Westcott, 1966) who,unfortunately, like to dwell only on the negative aspects of pesticide application. However, this negative view that is being instilled in the general public does have the positive effect of forcing much-needed research in the field to be initiated.

Many alternatives to pesticide application in its present form have been offered. One of the more promising is a technique called "integrated control". This method calls for the introduction of organisms that are natural predators of the economically harmful insects with a greatly reduced amount of pesticides applied to keep the predator populations in check. Other methods of control that are being considered are microbial disease treatments of insects, insect sterilization and metamorphosis inhibition, sex attractant sprays to draw the insects to one place for more efficient disposal, and developing genetic resistance to insect damage in plants (Westcott, 1966). Nevertheless the application of chemical pesticides is still the most widely used means of pest control. According to 1958 statistics, over 92 million acres were sprayed for pest control (USDA 1962). Yet many of the ideas and equipment principles used on today's application equipment are the product of research that was done 25,50 or even 75 years ago.

Current research in this area has been directed toward the development of application equipment which reduces substantially the amount of chemicals introduced into the ecosystem while at the same time providing adequate crop protection. This new approach requires a redesign of all the major components of the chemical delivery system including fan and nozzle characteristics and chemical metering of the low flow rates required.

II. BACKGROUND INFORMATION

Prior to 1868, there was little or no knowledge regarding chemicals for control of plant pests and, therefore, no need for application equipment (Potts, 1958). The Colorado potato beetle was the first pest to be controlled through chemical application. The substance used was called "Paris Green" and was applied using a box with a wire-screen bottom. The box was held over the potato plants and shaken so that some of the dust fell on the leaves. In 1882, copper sulfate and lime, Bordeaux, was applied to grapes in France to prevent grapevine mildew. The first applications were made with a crude whisk broom which was dipped in the Bordeaux solution and shaken at the vines. As soon as the means for controlling these diseases became available, machines for their application began to appear. A more sophisticated revolving brush arrangement was invented to apply the Bordeaux solution. Also a number of sprayers with tanks that were carried on a person's back appeared.

In 1883, John Bean invented the first force pump sprayer at Los Gatos, California (Fronk, 1962). From then until about 1900, many models of both the knapsack and force pump sprayer were introduced. In 1911, the pressure regulator was invented and in 1914, the adjustable spray gun was introduced to the market. From that time until the late 1930's, little progress was made except to increase the capacity of existing machines. About 1937, a citrus grower in Florida toyed with the idea of using an airblast as the carrying force for the spray solution instead of hydraulic pressure. As a result, the method of dilute airblast spraying was introduced and rapidly became the dominant form of pesticide application for orchard and grove crops.

One of the reasons for the rapid acceptance of airblast spraying was that it enabled the operator to spray a large acreage of trees with a limited amount of labor. Since it operates with less pressure than hydraulic sprayers, there is less wear and tear on valves, nozzles and other parts of the pumping system. When compared with other systems commonly used in orchards and groves, speed sprayers present the following additional advantages. They provide a very uniform coverage of the plant being sprayed. The formulations used have a very low level of toxicity to humans and are dilute enough so that slight errors in calibration are not critical. Most speed sprayers provide an adequate supply of air to carry the spray to the upper portions of the tall fruit trees (Brann, 1956).

Dilute airblast spraying does have some disadvantages, however. A tremendous volume of water is required for a single application and a lot of chemical is wasted through runoff which frequently ends up in underground water supplies. Airblast spraying equipment is rather large, bulky, and heavy, causing soil compaction and traction problems during wet weather (Courshee, 1960).

Several of the problems associated with dilute airblast spraying, namely the amount of water used and the material wasted, prompted the development of concentrate spraying. In this method, the same amount of chemical is placed in one-half or one quarter or possibly less water while reducing the amount of water sprayed on each tree by the same proportion. This method applies the same amount or approximately the same amount of chemical per acre as the airblast machines while using far less water per acre and, consequently,less total spray time is spent filling the tank. The concentrate sprayer in many instances, is an airblast sprayer that has been calibrated to put out only a fraction of its original volume per acre.

However, there have been some sprayers that are designed primarily for concentrate applications. These machines are usually equipped with a squirrel-cage fan capable of producing between 130 and 140 mph winds with volumes of at least 7000 cfm (Canada Department of Agriculture, 1963).

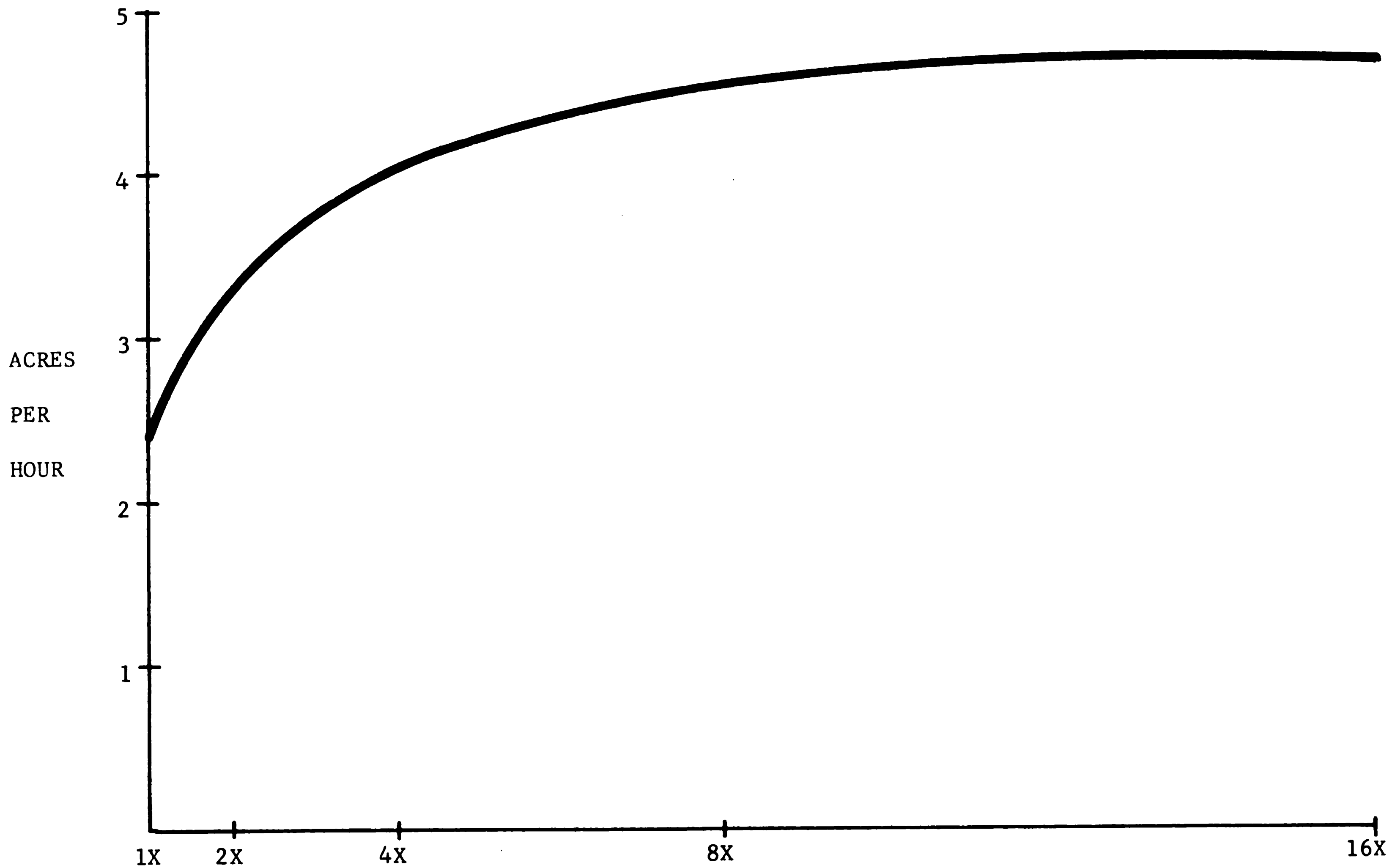
When compared with dilute airblast spraying, concentrate spraying offers several advantages (Brann, 1968). As mentioned earlier, there is a great reduction in the amount of water needed for applications and, as a result of the increased concentration of material, there is an increase in the amount of active chemical ingredient deposited per unit volume applied. With machines made primarily for concentrate spraying there is a reduction in weight of the equipment and therefore not as many problems with traction or soil compaction. There is also an increase in the number of acres that can be sprayed in a given period of time. Table 1 shows the increase in acres per hour with increasing concentration based on 500 gallons per acre as a dilute application. The savings in time reaches a point above which, however, it is negligible for any further increase in concentration as shown in Figure 2.1.1.

Concentrate spraying also has some disadvantages. One of the most crucial aspects is that slight errors in calibration are greatly magnified. The concentrated chemical solution creates more frothing in the tank while being agitated. Also the nonuniform coverage obtained with this method is not always effective against certain pests, for example mites, certain fungus diseases, etc. (Brann, 1968). The smaller droplets produced by concentrate sprayers lead to problems of drift, evaporation of carrier liquid and failure of some of the smaller droplets to impinge on the plant surface. As a consequence, concentrate applications are limited

TABLE 1.--Concentration and Acres Covered Per Hour

Concentration	Time to fill and travel	Spraying time for one tank	Gal/acre	Acres per hour
1X	12	12	500	2.5
2X	12	24	250	3.3
4X	12	48	125	4.0
8X	12	96	62	4.4
16X	12	192	31	4.7
32X	12	384	16	4.9

Source: Brann, 1968



Source: Brann, 1968

Figure 2.1.1. Concentration versus Acres Sprayed per Hour

to conditions of high humidity and low wind velocity. Another disadvantage, particularly of machines built strictly for concentrate spraying, is that they don't produce the right combination of air velocity and volume to adequately cover the tops of fruit trees (Courshee, 1960).

A new concept in pest control called ultra-low volume application began in 1962 with the introduction of the British Micronair rotary atomizer (Howitt, 1970). In a series of tests, insect control was achieved by aerial application with volumes as low as 1 pint per acre. The technical definition of ultra-low volume (ULV) is an application of material without water (neat) at a rate of one-half gallon or less per acre. The legal aspects of this definition become clear when it is observed that any application greater than one-half gallon per acre is considered low volume or concentrate. This is important since there are only a few materials that are registered as ULV materials by USDA (Farm Chem, 1967). Thus, this technical definition provides a loophole by which many growers can use other materials undiluted as long as they apply over one-half gallon per acre. The reason for the paucity of ULV formulations is due to stringent USDA registration requirements (Ag Chemicals, 1965). Before a ULV label is issued, the USDA must know the following: 1) Is it effective for all species of insects mentioned on the label? 2) Are tolerance limits met and can the public be adequately protected? 3) Will the smaller particle size and dosage adversely affect fish or wildlife? 4) Will the finely-divided particles create an unacceptable drift problem? and 5) Will following the proposed caution steps adequately protect the aerial applicator and his crew? These requirements stimulated the development of modified ground equipment for ULV application as a means of reducing the hazards of drift.

In 1965, the Michigan State University Department of Entomology modified a Buffalo Turbine sprayer for ultra-low volume applications (Howitt et al, 1966). The materials were dispersed through a fishtail fan assembly by a modification of the Micronair called a "minispin". Outstanding insect control was achieved when this sprayer was used for an annual spray program. Later ULV work (Howitt et al, 1969) brought about the development of a more precise method of producing small uniform droplets, thus making ULV spraying even more effective.

The field of ULV spraying has great promise in the future of pesticide application for a number of reasons. First, and foremost in the minds of growers, is the economy of ULV spraying. A ULV sprayer is compact and the initial cost is much less than for a large conventional air-blast sprayer. It is light and carries only a few gallons of material at a time so that it can be pulled by a small tractor. There is also a big savings in time due to the reduction in loading time. Another big factor is reduced dosages. Since ULV materials are undiluted, more active material is deposted with ULV methods than with conventional sprayers, and dosages could be greatly reduced, particularly since precise control of droplets can now be achieved. Because more active ingredients are applied per application, the residual effectiveness of these materials is being improved and this means fewer applications. Another advantage of ULV is that the formulations are less volatile than conventional solutions, resulting in less loss due to evaporation of the carrier liquid.

Ground application of ULV sprays does have disadvantages. Although not as bad as for aircraft application, there is still a considerable drift problem associated with ground applications of ULV sprays. Also the toxicity hazards to humans and animals are greater with ULV formulations (Farm Chem, 1967). The risk of overdosing some crops is also increased. One of the biggest problems is the lack of materials that are registered as ULV formulations. And finally, to date, ULV sprayers lack an accurate, reliable chemical metering system.

The problem of designing a reliable metering system has many facets. Not only must the system be able to meter rates that vary from 6 ml/min up to 250 ml/min, it must also be able to accommodate liquid viscosities that range from that of water solutions, approximately 1 cpse, to technical materials with viscosities of 650-700 cpse. To further complicate the problem, many of the chemicals are not true solutions. Some are colloidal suspensions or oil emulsions. Many exhibit non-Newtonian behavior such as thixotropy or rheopexy.

Early attempts at metering systems (Howitt, 1970) involved the use of a pressurized stainless-steel tank with controls for regulating the pressure and rate of discharge. A constant delivery was achieved by maintaining a prescribed pressure in the tank by means of a pressure regulator and air compressor, and controlling the outflow with a precision micrometer metering valve. With this system, flow rates could be controlled to within 1 ml/min. This system provided excellent results when using emulsifiable concentrates or chemicals that are liquids in the pure state. However, problems were encountered when trying to meter flowables, i.e. formulations that contain discrete particles. The small particles lodged in the orifice of the metering valve and thereby changed the flow rate. Many fungicides are flowables which are usually quite viscous due to the high particle density (Carter, 1969). Another problem was the deterioration of gaskets in the pressurized tank under the action of the caustic spray chemicals.

In 1969, the pressure tank-needle valve system was replaced by a peristaltic metering pump which could be adjusted for different rates of flow in three ways: 1) by the diameter of tubing used; 2) by varying the pump speed; and 3) by varying the amount of tube occulsion. This pump offered several advantages including the fact that discrete particles could not block the flow and also the corrosive chemical did not contact the internal pump parts. Delivery rate was adjusted by changing tubing size or varying the degree of tube occlusion since the pump was driven by a constant speed drive. Thus the pump was not always positive displacement in nature and therefore greatly affected by minute pressure changes. Also the delivery from this system was pulsating in nature which resulted in erratic distribution of spray material at very low flow rates.

III. OBJECTIVES

The objective of this research project was to develop a metering system for a ULV sprayer which:

1. has a steady and positive discharge for delivery rates from 6 ml/min to 250 ml/min;
2. can be easily and accurately calibrated for a given delivery rate;
3. can handle a wide range of material viscosities;
4. can handle the important pesticide formulations including true solutions, and flowable formulations;
5. is mechanically simple and easily maintained by the operator;
6. is economically compatible with the remaining components of the sprayer.

IV. LITERATURE REVIEW

4.1. METERING PUMPS

The problem of developing an accurate metering system may be approached in a number of different ways. One could take the peristaltic pump and modify it to provide the desired outflow characteristics. Or one could select one of the many other types of pumps available (Appendix A) and adapt it to the metering requirements. Metering pumps used in the chemical processing industry have characteristics somewhat in harmony with the requirements for ULV sprayers. According to London (1965),

> A metering pump--or, as it is called in America, a controlled volume or controlled capacity pump--is basically similar to the ordinary positive-displacement reciprocating pump. The one feature that distinguishes it as a metering pump is that the rate of delivery of the liquid can be pre-adjusted to within fine limits and will be maintained to ±1 per cent, or, under favourable conditions, to ±0.5 per cent.

Metering pumps are used to handle a wide range of substances including viscous liquids, slurries, liquified gases, molten metals, and even gases. Their internal wetted parts are made of materials which will not contaminate the liquid being pumped and are resistant to the corrosive action of the liquid. In the chemical process industry, they are primarily used as feeders or injectors of a substance into a main flow. A list of metering pump manufacturers and the types of pumps they make is given in Appendix B.

Metering pumps employ the positive displacement principle, i.e. the transportation of the fluid involved is caused by the alternate increasing and decreasing of volume of an enclosure (Wilson, 1950). Positive displacement is obtained in a number of ways including moving pistons and membranes, rotary elements, or a flexing type action (Marton, 1963). Each type has its own particular advantages and applications.

The plunger or piston pump is a reciprocating type of metering device which requires inlet and outlet valves and can work at very high pressures. The rate of delivery can be changed either by altering the length of the piston stroke, thus drawing more or less liquid per stroke, or by altering the rate of stroking with a variable speed drive, or both (London, 1965). The pumping speed is usually between 30 and 300 spm and is dependent on the type of installation and the physical properties of the liquid being pumped together with the desired flow rate.

The valving of reciprocating pumps is very important and many of the more accurate models have double inlet and double delivery valves (Figure 4.1.1). Double valves preserve the accuracy of the metering when a solid particle is passing through the pump which may prevent one valve from closing properly. There are three major types of valves used: ball, cone, and disc type. Ball valves are the most popular since they accommodate a wide variety of substances and have a long life. They can be gravity controlled, springloaded, free or guided. Cone type valves are used for liquids with viscosities less than water and devoid of suspended materials. Disc valves are standard in some pumps and can be made from a wide range of materials.

The plunger or piston type metering pump has the advantages of being self-contained, easy to operate, control, and maintain, and capable of handling a wide range of liquid viscosities (AIChE, 1960). However, the outflow of this type of pump is pulsating unless multiplexed, which is cost-prohibitive in many cases. It also requires a large amount of space, the materials of construction are limited, and it is not generally well suited for handling liquids which contain particles or abrasives.

A major breakthrough in chemical metering came about in the late 1940's with the introduction of a flexible diaphragm which kept the process liquid from contacting the plunger. This diaphragm or membrane type of metering pump (Figure 4.1.2) provided a virtually leak-proof system (Hetz, 1967). The diaphragm is actuated by a push rod, or by air or hydraulic pressure. The diaphragm is made of Viton, Teflon, or some other flexible material which can stand a pressure of about 125 psi, or it may be made of flexible metal which can take pressures up to 2500 psi (Marton, 1963).

Diaphragm pumps may not be as accurate as plunger pumps and sometimes diaphragm failures result from repeated and uneven stresses and strains (Figure 4.1.3). This problem is somewhat alleviated with the hydraulically actuated diaphragm pump which is really two pumps in one. The plunger applies a pumping action to the hydraulic fluid in a closed chamber with a diaphragm at one end (Figure 4.1.4). The diaphragm deforms causing a pumping action on the process liquid. Because pressure is exerted equally on the entire surface of the diaphragm by the hydraulic fluid, there is no excessive strain at any single point.

Where there is a danger of reaction between the hydraulic fluid and the process liquid, a second diaphragm is placed in the chamber (London, 1965). Between this diaphragm and the pumping diaphragm is placed an inert liquid, The hydraulic fluid is placed between the plunger

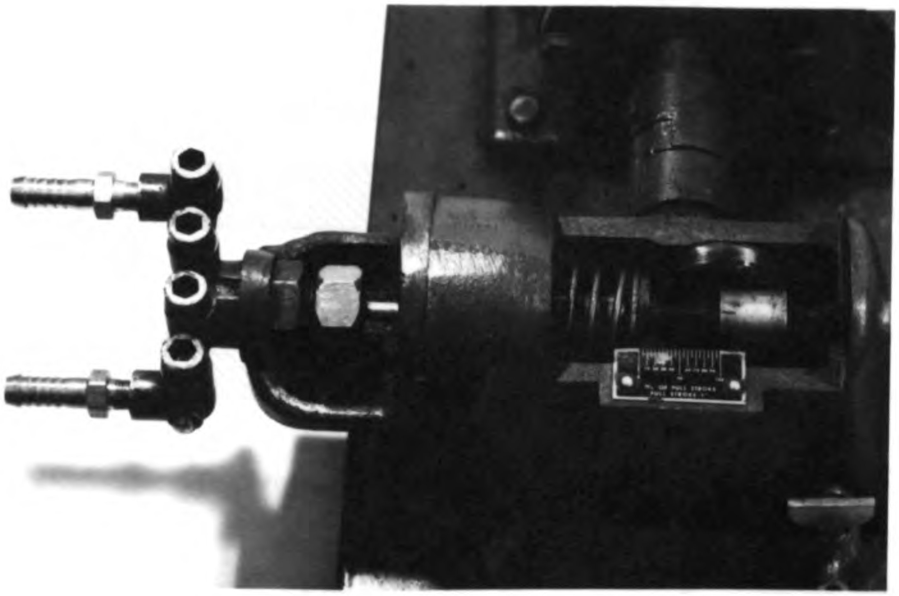


Figure 4.1.1. Milton Roy Reciprocating Pump With Double Inlet and Discharge Valves.

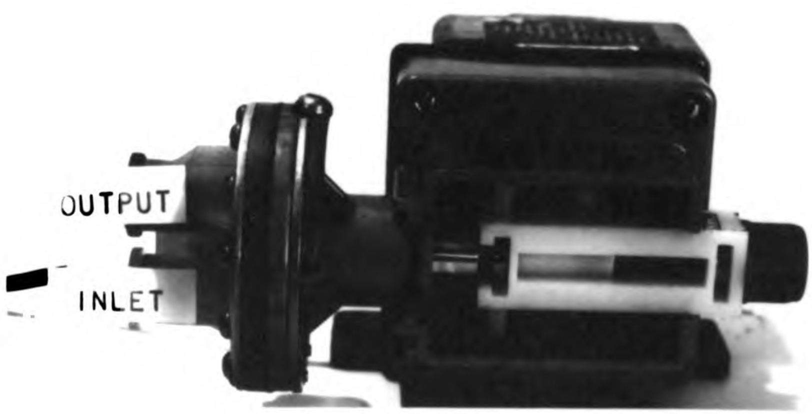
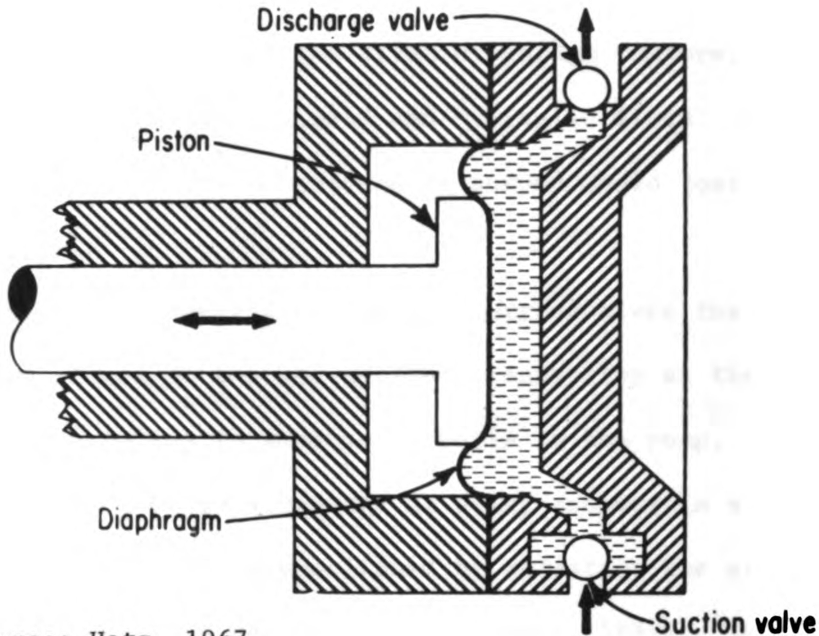
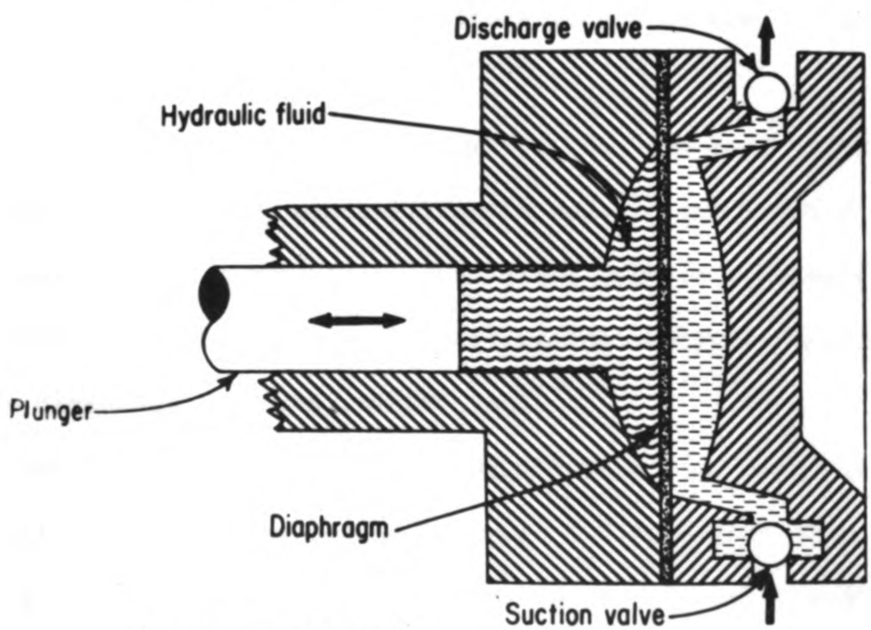


Figure 4.1.2. Jaco Mechanically-Accuated Diaphragm Pump.



Source: Hetz, 1967.

Figure 4.1.3. Liquid End of a Diaphragm Pump--Mechanical Support.



Source: Hetz, 1967.

Figure 4.1.4. Liquid End of a Diaphragm Pump--Hydraulic Support.

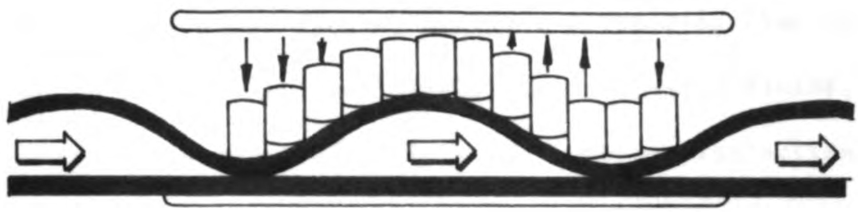
and the first diaphragm. Should the first diaphragm rupture, there would just be a mixing of inert liquid and hydraulic fluid. Should the pumping diaphragm rupture, the liquid being pumped would just be diluted with the inert liquid.

Diaphragm pumps work well for pumping slurries since the suction connection can be placed on top and delivery connection at the bottom to help maintain the solids in suspension while in the pump. In most instances they have the added advantage of being compact in size.

Changes of flow rate are accomplished by adjusting the stroke of the piston. This is done in one of two ways (Hetz, 1967). The eccentricity of the crank can be modified, thus changing the effective stroke. Or a constant amount of the eccentricity can be diverted from the plunger thus modifying the stroke length by what is called the "lost motion" technique.

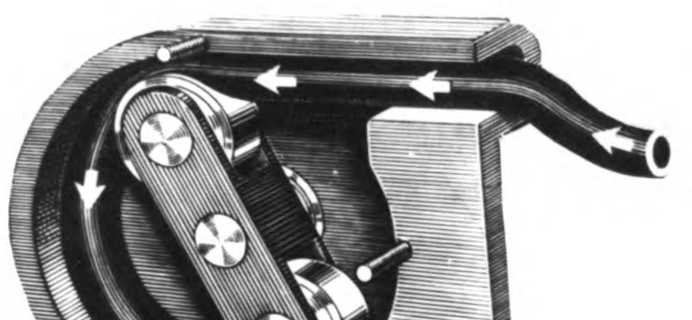
Diaphragm metering pumps, however, do have some disadvantages. Like piston pumps they are generally quite costly, are limited in the types of materials for construction, and still produce a pulsating flow. Diaphragm pumps must also be protected against overpressures which might cause diaphragm failures (AIChE,1960).

Peristaltic metering pumps can handle a wide range of liquids and slurries. The pumping action involves the squeezing of a flexible container or hose such as cam-operated fingers (Figure 4.1.5) pressing the tube in succession or a rotor with rollers which sqeezes a tube in a curved cradle (Figure 4.1.6). This process closely resembles some body functions and, therefore, this type of pump has wide use in medical and biochemical technology. The metering is very accurate and can be



Source: Marton 1963

Figure 4.1.5. Peristaltic Pump With Cam-Operated Fingers.



Source: The Randolph Company, 1970.

Figure 4.1.6. Revolving Roller Peristaltic Pump.

very small (down to 1 drop per hour) and the fluids do not contact the pump, thus preventing contamination of the pump or the fluid. These pumps are self-priming and require no valves or seals (Marton, 1963).

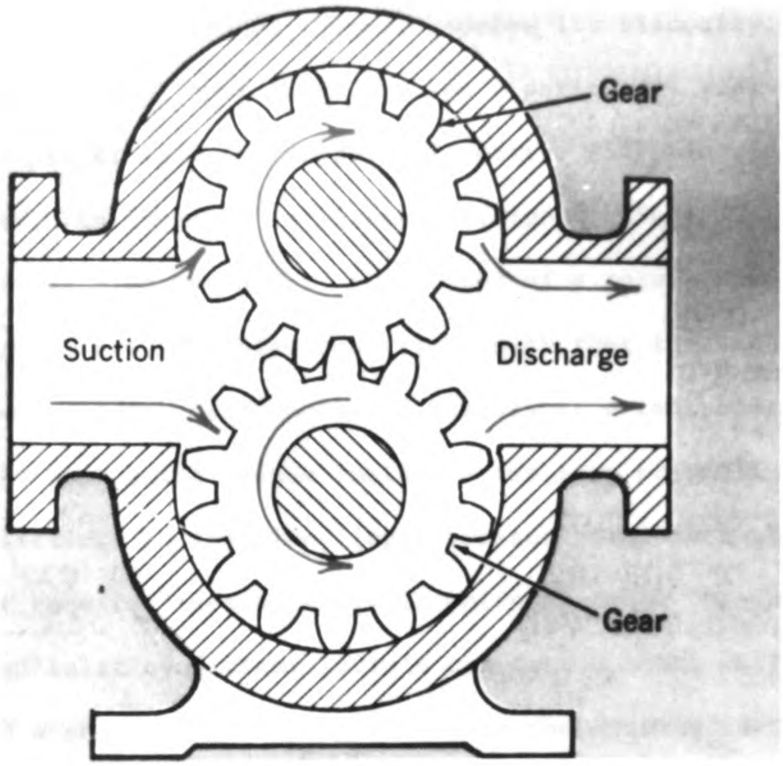
Rotary metering pumps furnish nearly a pulse-free flow under pressures up to 1500 psi and can handle high viscosity fluids. They can have a very high volumetric efficiency. The pumping action is accomplished in a variety of ways including eccentric rotors with sliding vanes, rolling vanes and other impellors, and meshing spur gears. The external gear pump shown in Figure 4.1.7 is one type of rotary pump frequently used for metering.

Both peristaltic and rotary pumps show promise for metering systems for ULV sprayers. The peristaltic pump warrants further inspection because of its high accuracy at low flow rates (Marton,1963) and isolation of the liquid being pumped. Rotary gear pumps present the desired characteristics of high accuracy and pulsationless flow.

4.2. ROTARY PUMPS

The general working principle for rotary pumps is that as the rotor turns, it creates cavities which move from suction to discharge forcing the liquid along. A seal between suction and discharge is formed by close running clearances or rolling or sliding contact (AIChE, 1960).

The output of rotary pumps is proportional to speed except for losses due to inlet conditions and normal internal leakage from discharge to suction, called slip, which varies inversely with the viscosity of the fluid and directly with pressure differential. The inverse relationship of slip and viscosity usually does not hold for high speed and high



Source: White, 1971.

Figure 4.1.7. External Spur Gear Rotary Pump.

viscosity. Under these conditions, there is a noticeable temperature effect due to the high rate of viscous shear in the pump which heats the liquid in the clearance spaces and thereby reduces its viscosity. Slip is a minimum in an external spur gear pump for a peripheral gear speed of 34 fps and 100 psi at 1500 sec Saybolt viscosity. Slip increases from this point with an increase in viscosity (Pigott, 1944).

Inlet conditions also can affect the output of a rotary pump. It is common practice to provide inlet conditions such that the vacuum at the pump inlet does not exceed 5 inches of mercury at normal operating speeds. Causes of high inlet vacuum include a too long or small inlet line, bends and fittings in the line, large pressure drop through a strainer, or lift required from a reservoir which is below the pump. The results of bad inlet conditions include cavitation which decreases the flow rate and a shortening of pump life. As the suction increases, the pump may become noisy and begin to vibrate (Horn, 1968). Theoretically, flow is independent of the pressure differential for rotary positive displacement pumps. In reality, there is a slight increase in slip with increased pressure resulting in decreased flow. Power requirements vary with pressure and speed, or, if both are constant, with the viscosity of the fluid.

Rotary pumps are used for pumping a wide variety of materials including chemicals,oils, gasoline, solvents, tar and varnish. They can handle pressures up to 1000 psi with non-lubricating fluids and even higher with lubricants. Rotary pumps can also produce low flows over a wide range of pressures. They have a relatively low initial cost and require little space. However, the close clearances associated with rotary pumps limit somewhat the choice of materials of construction and also the ability to pump liquids containing suspended particles. Also, being positive displacement pumps they must be protected from excessive pressure with a relieving device.

A rotary pump is a positive displacement pump which consists of a set of gears, cams, screws, vanes, or other elements in a casing which are actuated by a drive shaft. Rotary pumps are characterized by their lack of inlet or outlet valves, and their close running clearances. Two general classes of rotary pumps are, single rotor and multiple rotor (Hydraulic Institute, 1965). Relationships between types and classes of rotary pumps are shown in Appendix C. There are four basic single rotor pumps (Hydraulic Institute, 1965). They are (1) vane pumps which have a vane or vanes that take the shape of blades, buckets, rollers, or slippers and operate with a cam to draw fluid into and force it from the pump cavity; (2) rotary piston pumps which have reciprocating pistons mounted in a rotor; (3) flexible rotor pumps containing a flexible tube, vane, or liner; and (4) screw pumps. Screw pumps, which can be either single or multi-rotor, displace the fluid axially through the rotor screw threads. Two other types of multiple rotor pumps, lobe (Figure 4.2.1) and circumferential piston, carry the fluid in the spaces between the lobes or pistons from inlet to discharge. Both pumps lack torque transfer contacts between rotor surfaces and, therefore, must be externally timed.

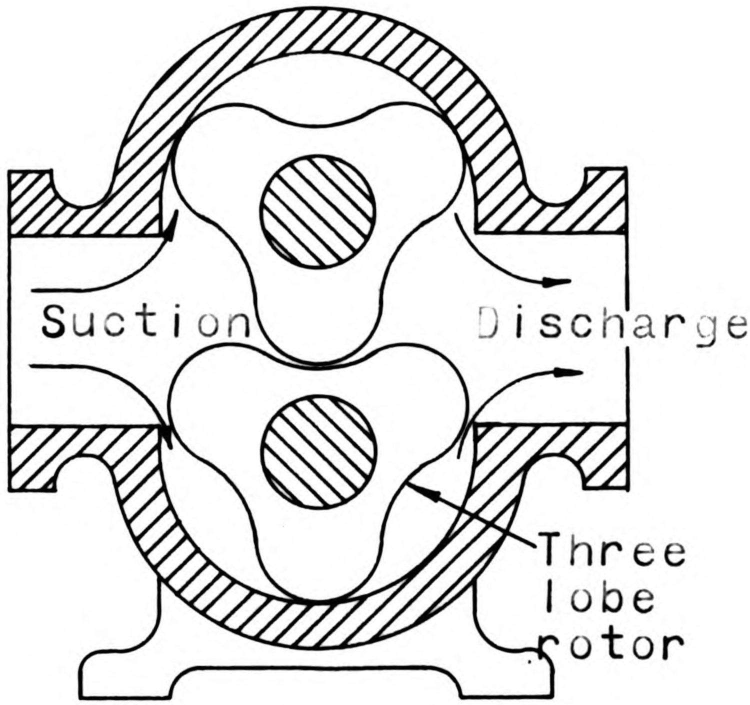
The gear pump is one of the most common multiple rotor rotary pumps. Internal gear pumps have one rotor with internally-cut gear teeth that meshes with a rotor having externally-cut teeth. Some internal gear pumps are made with a crescent-shaped partition (Figure 4.2.2) separating inlet and discharge. Other internal gear pumps have gears with teeth of a size, shape, and number so that a separator is not needed. External gear pumps (Figure 4.1.7) have externally-cut gear teeth on all rotors and these teeth may be spur, helical or herringbone in shape.

External spur-gear pumps are intended for slow-speed operation up to about 600 rpm for handling most thick viscous liquids which possess lubricating properties. Most spur-gear pumps have a driver and idler gear type arrangement with the driver gear keyed to an external drive shaft. However, sometimes, the rotors are driven by timing gears for several reasons. Rotor drive contact may produce too much wear or a clearance may be desirable to avoid the necessity of lubricants in the pump chamber. Over part of the cycle, contact may be such that torque can only be transmitted in the sense opposite to that required (Hadekel, 1951).

The volume delivered by an external gear pump depends on tooth depth, so that the greatest output for a given pump size and speed is produced with the minimum number of teeth (Horn, 1968). Displacement per revolution in a gear pump where the gears mesh perfectly is equal to one-half the volume enclosed between the addendum and dedendum circles of the gear or, for a two-gear pump:

$$D = (r_a^2 - r_d^2)b \qquad (4.1)$$

where: D = Displacement per revolution
r_a = Radius of addendum circle
r_d = Radius of dedendum circle
b = Length of the gear



Source: AIChE, 1960.

Figure 4.2.1. Three-Lobe Rotary Pump.

Source: Stuart and Heldeman, 1970.

Figure 4.2.2. Internal-Gear Rotary Pump With Crescent-Shaped Partition.

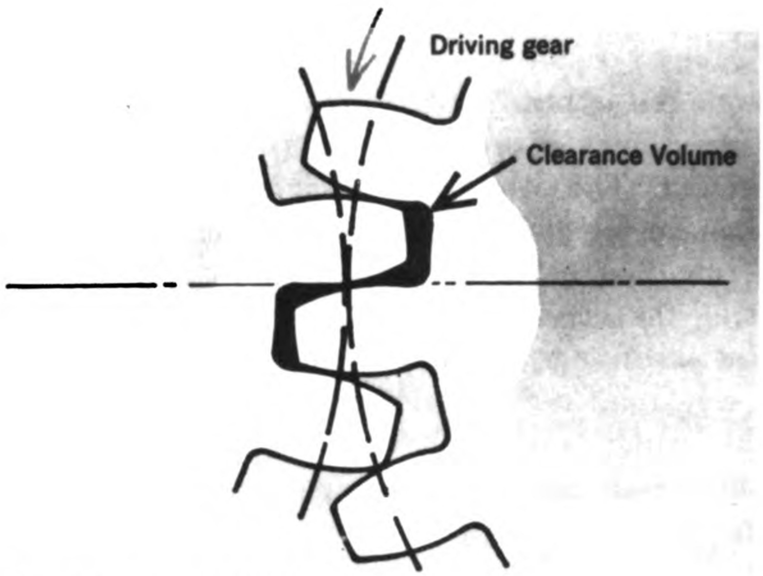
Volume delivered is equal to displacement per revolution multiplied by the speed of the pump (Wilson, 1950). This, of course, is a simplified and idealized treatment of displacement calculation. An experiment to determine actual displacement per revolution is discussed by Pigott (1944) and a rigorous mathematical calculation of delivery per radian of rotation considering such parameters as number of teeth per gear, gear-tooth pressure angles, etc. is performed by Hadekel (1951).

One problem associated with spur-gear pumps, is liquid which is trapped between meshing gears (Figure 4.2.3) and the resulting pressures which produce unbalanced bearing loads. There are several methods used to relieve this pressure. One method is to drill radial holes in the tips and roots of the gear teeth allowing an escape route for the trapped fluid (Kristal, 1953). Small grooves can be cut in the side plates allowing the trapped liquid to pass from tooth to tooth (Pumping Manual, 1962). Another method used for low pressure applications is to design a small amount of backlash into the gears or choose the dimensions of the gears in such a way that the trapped space is <u>increasing</u> during the entire overlap period (Hadakel, 1951).

Another unbalanced pressure bearing load is caused by the basic design of the spur-gear pump. The pressure on the inlet side of the gears is essentially atmospheric while the pressure on the outlet side is whatever the discharge pressure happens to be thus placing an unbalanced load on the bearings. This load is called a differential pressure load and is equal to the outlet pressure multiplied by the projected area of the gear blank (Stuart & Holdeman, 1970). This pressure differential affects notonly the bearing loads, but also, as mentioned before, the amount of leakage or slip of the pump. Some pumps have been designed to eliminate this unbalanced load but are seldom used. The three spur-gear pump (Figure 4.2.4) eliminates the bearing load unbalance on the driver gear, and has the added advantage of supplying practically double the outflow for a relatively small increase in size.

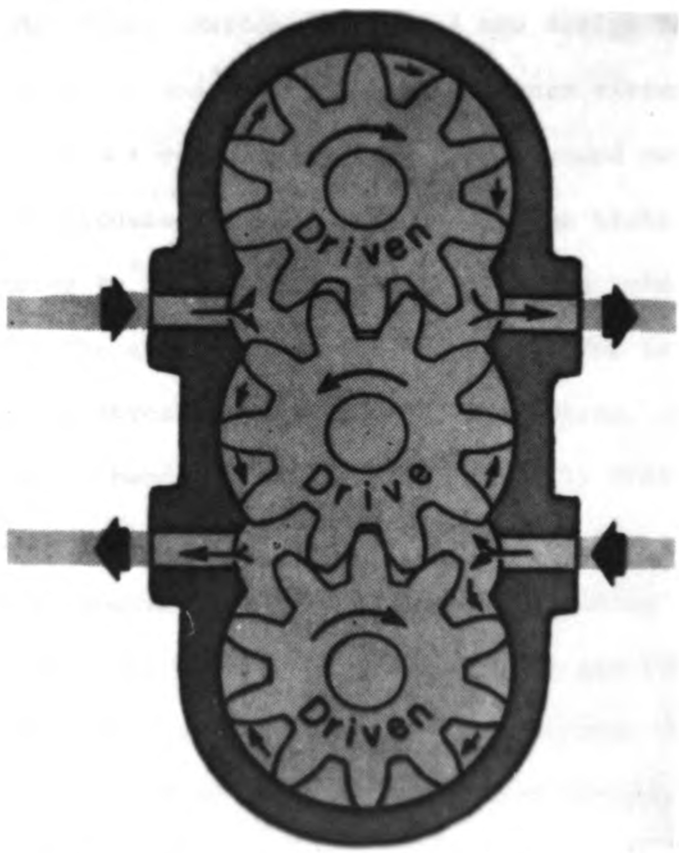
4.3. PERISTALTIC PUMPS

Peristaltic motion is defined as "the rhythmic, wavelike motion of the walls of the alimentary canal and certain other hollow organs consisting of alternate contractions and dilations of transverse and longitudinal muscles that move the contents of the tube onward" (Guralnik, 1970). Pumps that operate with this type of motion, with no glands or valves, are being used in increasing numbers in medical applications, research and industry. Peristaltic pumps are capable of handling fluids which cannot tolerate atmospheric contamination or, conversely,must not be allowed to corrode pump parts or contaminate the atmosphere. They can handle many types of fluids including slurries, near-solids, liquids, and gases, and are self-priming (Neal, 1965). Since the fluid remains within flexible tubing, there are no cleaning problems and sanitation can be insured. Peristaltic pumps are essentially low-pressure pumps and are not used for continuous heavy-duty pumping. The flexible tubing is subject to repeated stresses and its service life is dependent upon the speed of the pump, the pressure differential, and the nature of the fluid being pumped.



Source: White, 1971.

Figure 4.2.3. Liquid Trapped Between Gear Teeth.



Source: Stuart and Heldeman, 1970.

Figure 4.2.4. Three Spur Gear Rotary Pump.

The "muscles" that contract the tubing take several shapes, the most common of which is a series of rollers mounted on the periphery of a rotor. The tubing is mounted in a curved cradle so that as the rollers pass over it, the tubing is progressively squeezed, pushing the fluid forward in front of the advancing contraction toward the outlet and simultaneously creating a vacuum in the tubing behind the roller drawing in fluid from the inlet (Figure 4.1.6). Although most peristaltic pumps use this revolving roller scheme, some use a cam-operated sequence of metal fingers which progressively squeeze the tubing (Figure 4.1.5). Another method involves a rotor on an eccentric shaft which squeezes a flexible cylinder liner (Marton, 1963). A new design has been developed for biological and medical research where extremely low flow rates are needed. It is a very small elastic tube wound around a delta-shaped rotor which produces kinks in the tube. The kinks travel around as the rotor rotates to force the contents along the tube (Neal, 1965).

The design of the cradle and tube-holding devices is an important factor affecting the stresses, and life, of the tubing. Most peristaltic pumps are designed to handle almost any commercially available tubing that is appropriate in size and material properties for a given application. Some manufacturers, however, do provide preshaped tubing for special applications. Some pumps have a fixed-speed drive and their output is controlled by tubing size or degree of tubing occlusion (Howitt,1970). In others output is varied by changing pump speed through mechanical variable gearing or through an electronic speed unit. In the most sophisticated pumps, the pump speed, thus the flow rate, can be controlled by a pneumatic or electrical control signal generated by some process variable (Marton, 1963). Because of the similarity of peristaltic pumping action to body functions, there has been much research (Latham, 1966; Jaffrin and Shapiro, 1971) with the specific objective of modeling the human ureter. The purpose of this modeling was to attempt to discover the mechanism of fluid mixing or motion in the tube which might explain disease transmission from the bladder in a reverse direction to the kidneys. Research has shown that under many conditions of operation, the net time-mean flow in peristalsis is the algebraic difference between a forward time-mean flow in the core of the tube and a backward (reflux) time-mean flow near the periphery (Shapiro et al, 1969). This would seem to be the explanation for the reverse disease transmission, but it has been shown that reflux conditions are not present in a healthy ureter (Weinberg et al, 1971).

The amount of backward motion of reflux of the fluid can be quite significant. It is affected by the degree of pinch or occlusion of the tube and also by the pressure differential from the inlet to the outlet (Latham,1966). With a constant degree of tube occlusion and constant pump speed, the amount of reflux increases as pressure head increases. If pressure and speed are held constant and the degree of occlusion is increased, reflux decreases and the output of the pump increases. In peristaltic metering pumps, the tube is completely or almost completely occluded. The pump is essentially positive displacement in nature and outflow is independent of pressure head within certain finite limits (Jaffrin and Shapiro, 1971).

The fields of application for peristaltic pumps are continually growing. In industry, they are used for handling corrosive and radioactive fluids, cement (both wet and dry), printing ink, and foodstuffs. In the chemical processing industry, they are used for continuous sampling and analysis, for effluent control, and boiler feedwater dosing. In the medical field, peristaltic pumps are used on artificial kidney machines, and for controlled dosing in both surgical application and in research (Neal, 1965). Peristaltic pumps are increasingly taking over many of the duties of other types of pumps because of their simplicity, cleanliness, and versatility.

V. CHARACTERISTICS OF SPRAY MATERIALS

Before experimenting with a metering system, the characteristics of the materials to be handled should be known. Important characteristics influencing the development of a metering system are density and viscosity of the materials. Also of interest are the specific rheological characteristics if the metered fluids have non-Newtonian behavior because the performance of the metering system may be affected by resulting changes in apparent viscosity.

5.1. SPECIFIC GRAVITY MEASUREMENT

Specific gravity was measured using a beaker of material, a thermometer, and a precision balance (Figure 5.1.1). The procedure was to (1) record the material temperature, (2) weigh the beaker of material and subtract the weight of an empty beaker,and (3) calculate the weight of an equal volume of water at the same temperature. Specific gravity was then obtained using the following formula:

$$\text{Specific Gravity} = \frac{W_m}{W_w} \qquad (5.1)$$

where:

W_m = Weight of volume of material at a fixed temperature

W_w = Weight of equal volume of water at the same temperature.

The results of this test and manufacturers' specifications are given in Table 2. The results show that most of the chemicals tested were similar in density to water. Thus, density is not expected to seriously affect the flow rate from the metering system.

5.2. VISCOSITY MEASUREMENT

Fluid viscosity influences pump output for a given speed and also power requirements for a given output. A Brookfield Synchro-Lectric viscometer (Figure 5.2.1) was used for the measurement of the viscosities of selected chemicals. This viscometer rotates a spindle in the fluid and measures the torque necessary to overcome the viscous resistance to induced movement. Four spindle speeds and four sizes of cylindrical spindles are provided for measuring a wide range of viscosities. Non-Newtonian behavior related to the rate of shear can be studied simply by changing speeds with a given spindle size. Cross-like spindles and a special Helipath stand are included with the equipment for measuring apparent viscosity of non-Newtonian fluids for which viscosity at a fixed rate of shear changes with time.

Viscosity measurements were made with a 3-inch diameter beaker of fluid which is greater than the 2.25-inch minimum size required for the viscometer. The procedure followed for each measurement was to use the viscometer factor charts and an estimate of the viscosity to select the spindle to be used for that test. The dial reading was recorded for each of the four spindle speeds. The viscometer was run for several minutes for each test to determine if viscosity changed with time. If a change with time was observed, the special cross-like spindles and Helipath stand were used for measuring the apparent viscosity. Viscosities were calculated by multiplying the dial reading by the viscometer factor for the spindle.

All substances were evaluated at three different temperatures to observe the effect of temperature on viscosity. The three temperatures chosen were 49°F, 72°F, and 85°F. The samples were brought to 49°F by

Figure 5.1.1. Specific Gravity Test Apparatus.

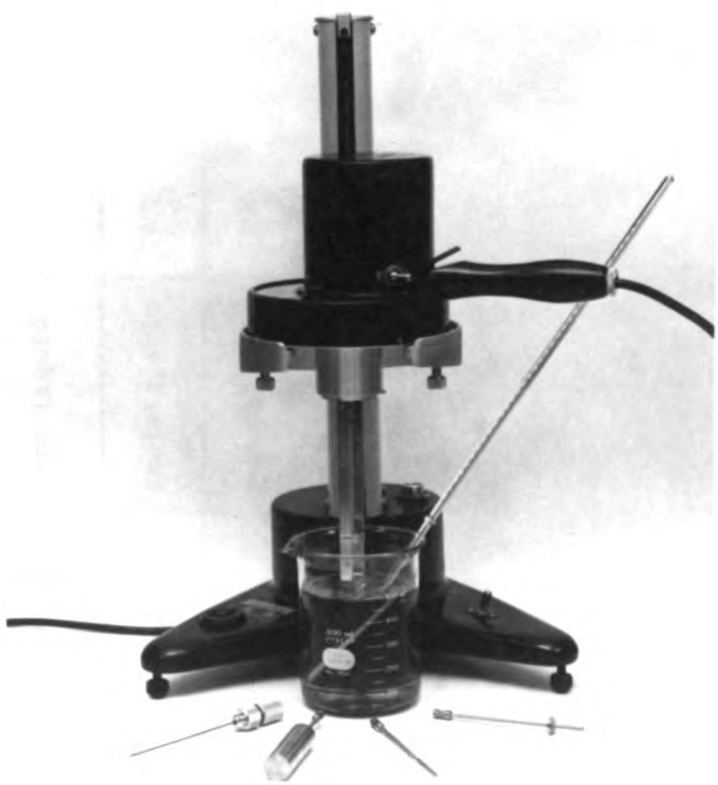


Figure 5.2.1. Viscosity Measurement Apparatus.

TABLE 2.--Characteristics of Common Spray Materials

Material	Classification	Composition	Measured Specific Gravity	Manufacturers Specific Gravity #	Measured Viscosity (cpse)	Manufacturers Viscosity (cpse) #	Comments
Cythion	Insecticide	Pure Liquid	1.175	1.2315	27 at 77°F	36.78 at 77°F	Newtonian
Perthane 4EC	Insecticide	Emulsifiable Concentrate	.972	1.022	≈ 5 at 72°F	*	Newtonian
Dioctyl-Pthalate (DOP)	Carrier Liquid	Pure Liquid	.97	.9861	45 at 71°F	*	Newtonian
Sevin mole 4	Insecticide	Flowable	1.13	1.2	340-160 at 80°F	190-350 at 80°F	Non-Newtonian Thixotropic and Pseudoplastic
Sevin 4LS	Insecticide	Flowable	.98	1.03	705 at 80°F	650 at 80°F	Non-Newtonian Thixotropic and Psuedoplastic
Omite 6E	Acaricide	Emulsifi-able Concentrate	1.01	1.04-1.08	47 at 72°F	44 at *	Newtonian

* Information not available.

Information obtained through personal correspondence with chemical companies' representatives.

placing them in a refrigerator overnight. The 72°F fluid temperature was obtained by letting the samples stand at room temperature for a day. And the 85°F fluid temperature was obtained by heating all samples in an oven. All samples were sealed with thin plastic to prevent evaporation of carrier liquids with low vapor pressures.

Viscosity decreased with increased temperature for each of the six materials tested. Graphs showing the effect of temperature on viscosity are presented in Appendix D. The measured viscosities are given in Table 2 together with specifications obtained from chemical manufacturers. The measured viscosities were within approximately 20 percent of the manufacturers' specifications. The viscometer calibration curve is given in Figure 5.2.2. One problem encountered in some of the tests was that the material adhered to the spindle. Another possible source of error was the inability to keep the samples and spindles at a constant temperature for the entire set of measurements at that temperature.

Most of the materials, particularly pure chemical liquids and emulsifiable concentrates showed very-nearly Newtonian behavior with a range of viscosities from about 5 cpse at 85°F to 650 cpse at 49°F. The flowable materials exhibited a high apparent viscosity because of a high volumetric content of discrete particles. The flowable formulations exhibited a decrease in viscosity with time, characteristic of thixotropic fluids. Viscosity of the flowable materials decreased with increased shear rate indicating a pseudoplastic behavior. According to Carter (1969), decreases in viscosity due either to thixotropic or pseudoplastic behavior can cause increased slip and a

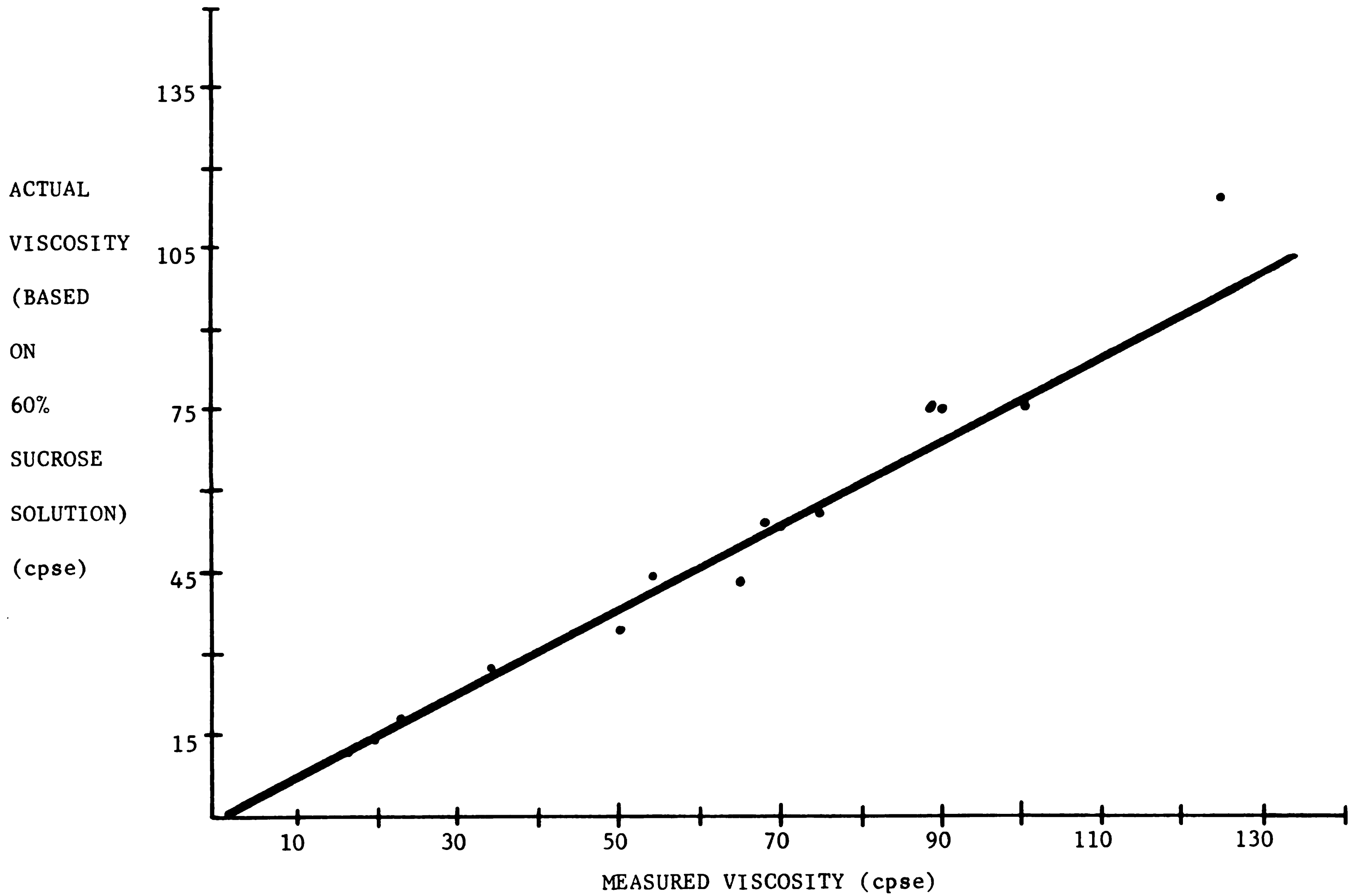


Figure 5.2.2. Viscometer Calibration Curve.

significant reduction in output.

5.3. SELECTION OF TEST MATERIALS

Upon conclusion of the viscosity tests, two materials representative of the two major types of liquids to be metered were chosen for use in the metering system evaluation. Cythion was chosen as representative of the pure liquid formulations. It has a specific gravity of 1.2 and a viscosity of 36.78 cpse at 77°F. It is Newtonian in behavior and one of the most widely used low volume insecticides.

Sevinmole 4 was selected to represent the flowable formulations. It has a viscosity range of 190 to 350 cpse and a specific gravity of 1.2 at 80°F. It is a pseudoplastic and also slightly thixotropic.

VI. PERISTALTIC PUMP SYSTEM

6.1 INTRODUCTION

The most important requirement for the metering system was a constant output delivery for a given pump speed even with variations in pressure head caused by ground contour in the orchard. The net outflow of a peristaltic pump for a given viscosity and pump speed is dependent upon the amount of backward flow or reflux in the tubing. The amount of reflux is affected by the degree of tube occlusion and also by pressure differential (Latham, 1966). For positive displacement, required for a metering pump, tube occlusion should be as complete as possible in order to minimize reflux. In the Sigmamotor T6S peristaltic pump (Figure 6.1.1) the degree of tube occlusion is dependent upon the position of a flat steel plate with respect to the cam-operated fingers. The plate provides a backing against which the fingers press the tubing. If the plate is adjusted sufficiently close to the operating fingers, the tubing will be pinched until one wall contacts the other and the tube is completely occluded (Figure 4.1.5). If the plate is set back from the fingers, only partial occlusion takes place. Adjusting this plate for complete occlusion virtually eliminates reflux. To minimize the influence of reflux on pump output, the backing plate adjustment method of output modification was abandoned, leaving tubing size and pump speed as alternate output control methods. Changing tubing size at constant pump speed gives a change in output. The number of different output values, however, is limited to the number of tubing sizes the pump can accommodate. A more practical method of output control is to change the pumping speed through the use of a variable-speed drive.

The variable-speed transmission used was a disc and ball bearing type in which the angle of contact between a pair of rotating discs was adjusted with a screw providing an infinitely variable output speed between zero and 645 rpm for an input speed of 1725 rpm. A small electric generator-type tachometer with an accuracy within 2 percent of full scale reading was used to measure the pump shaft speed. Flexible ¼-inch ID Tygon clear plastic tubing was used throughout the system except for a six inch piece of ¼-inch ID Viton chemical-resistant tubing on which the pump fingers operated. The length of the inlet line was 2.5 feet. Coupled in the output line about a foot past the pump was a Gilmont flowmeter which was used to indicate flow uniformity. The output line from the flowmeter was about 1.5 feet in length. Figure 6.1.2 shows the general arrangement of the test setup.

The theoretical displacement per revolution of the Sigmamotor peristaltic pump was estimated in the following way. Figure 4.1.5 indicated that the tubing trapped between the two extreme fingers is pinched into a half-sine wave shape. The average diameter perpendicular to the backing plates, therefore, equalled 0.707 times the maximum diameter or 0.177 inches for ¼-inch diameter tubing (Figure 6.1.3). As the diameter perpendicular to the backing plate becomes smaller due to pinching by the fingers, the diameter parallel to the face of the

Figure 6.1.1. Sigmamotor Pump Showing Backing Plate and Cam-Operated Fingers.

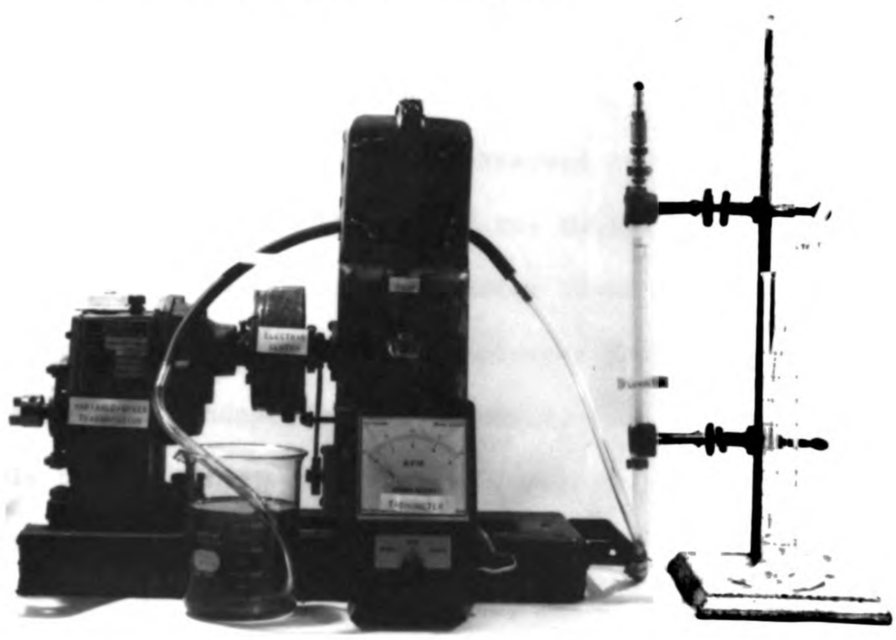


Figure 6.1.2. Output Versus Pump Speed Test Setup for the Peristaltic Pump.

backing plate gets larger for geometric compatibility which assumes the perimeter of the inside of the tube remains constant (Figure 6.1.4). The unequal diameters became the major and minor axes of an ellipse. The theoretical lengths of these axes were calculated as 0.314 inches and 0.177 inches. Using these dimensions and the distance between pinch-off points, the volume of the theoretical elliptical cylinder was calculated. This value was the theoretical displacement per revolution. For ¼-inch tubing, the displacement per revolution is 1.964 ml/rev.

6.2. EFFECT OF PUMP SPEED

The procedure followed for each test was to set the pump speed and let the system operate for a few minutes to eliminate the initial effect of the dry, uncoated tubing. Then the total output was collected in a graduated cylinder for a one minute interval. The behavior of the output from the pump was visually observed in two ways. The evenness of flow was observed at the end of the tubing as the sample was collected and oscillations of the flowmeter float were also noted.

Comparing Figures 6.2.1 and E.1 indicates that pump output at low speed was relatively independent of viscosity for pure liquids and emulsifiable concentrates. As pump speed was increased, there was a slight decline in output with the more viscous material. For Cythion in the range from 20 to 160 rpm the output per revolution changed from 0.97 ml/rev to 0.85 ml/rev. Figures 6.2.2 and E.4 show that outputs for the flowable formulations decrease quite rapidly with an increase in pump speed. For Sevinmole 4 the output per revolution changed from 1 ml/rev to 0.43 ml/rev for the 20 to 160 rpm speed range.

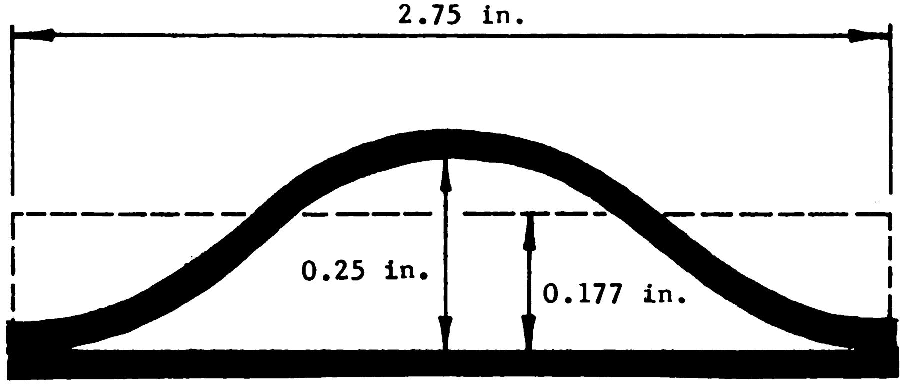
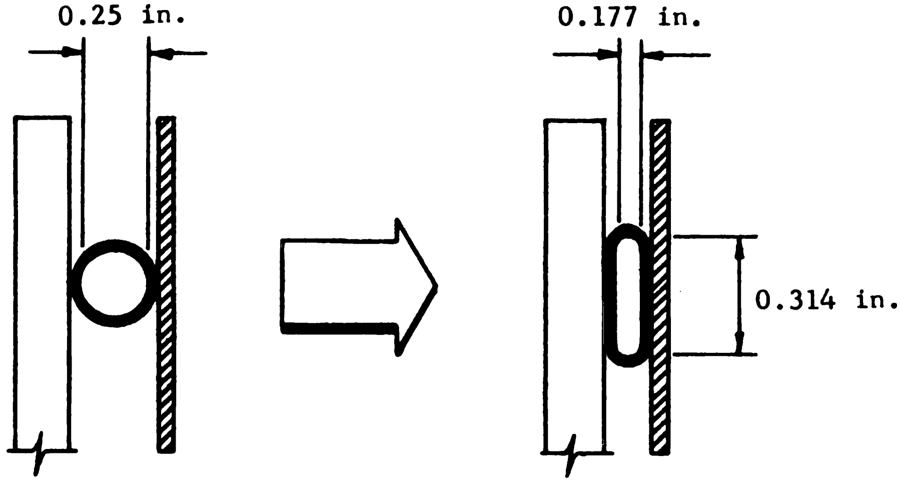


Figure 6.1.3. Average Tubing Diameter Perpendicular to Backing Plate.



Perimeter $= \pi d$

$= 0.785$ in.

Area $= \dfrac{\pi d^2}{4}$

$= 0.048$ in.^{2}

Perimeter $\approx \dfrac{\pi (a + b)}{4} [3(1 + \lambda) + 1/(1 - \lambda)$; $\lambda = \left[\dfrac{a - b}{2(a + b)}\right]^2$

$= 0.785$ in.

Area $= \pi a b$

$= 0.044$ in.^{2}

Figure 6.1.4. Change in Shape of Tubing Cross-section as Tubing is Pinched.

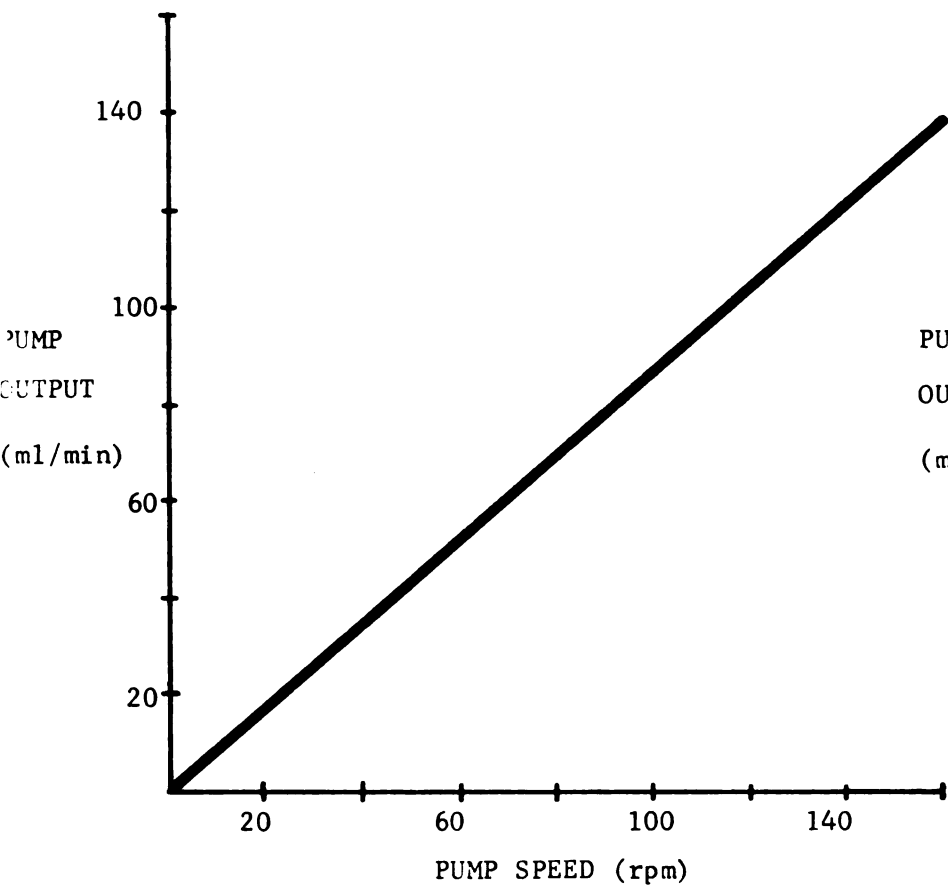


Figure 6.2.1. Output versus Pump Speed Using Cythion--Peristaltic Pump.

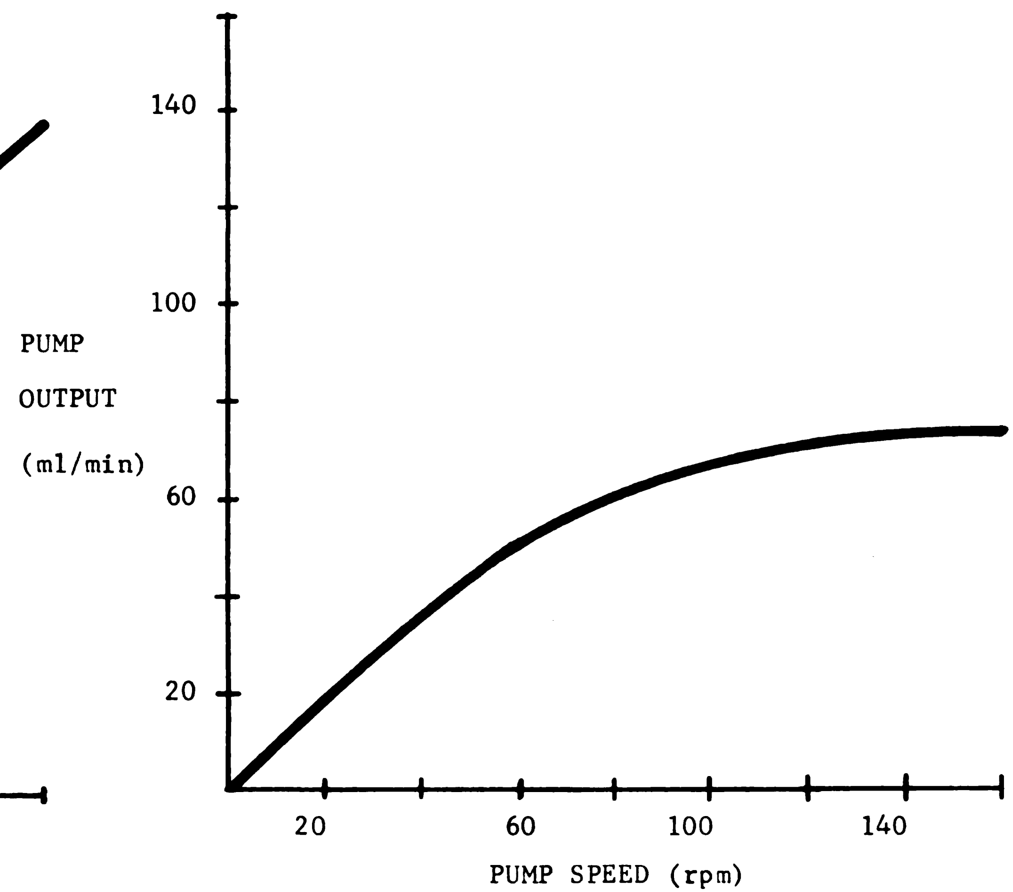


Figure 6.2.2. Output versus Pump Speed Using Sevinmole 4--Peristaltic Pump.

There was quite a substantial difference between the theoretical displacement per revolution (1.964 ml/rev) and the actual displacement per revolution (1 ml/rev to 0.43 ml/rev). The difference can be partially explained by the adverse flow properties of some of the materials especially the friction drag forces caused by the particle content of flowables which restricted inlet capacity at higher pump speeds. Another factor based on visual observation was a permanent change in hose shape from round to slightly elliptical after several tests. This permanent deformation would affect hose capacity and, consequently, displacement per revolution. Another possible cause was improper pump sealing, i.e. the forward-most pinching finger allowing line pressure into the pumping segment of the hose before the inlet side is completely sealed off, thus causing a small amount of reflux.

An assumption made for the theoretical displacement calculation was that the tube sustained pinching force from the fingers only in a direction perpendicular to the tube wall. Any strain or movement in the axial direction was disregarded. In reality, the fingers may also have stretched the tubing longitudinally further reducing the effective displacement volume. Furthermore, the deformed shape of the tubing was taken as elliptical for ease of calculation when the actual shape has two parallel sides and two rounded ends (Figure 6.1.4). This shape has the same perimeter as an ellipse but not as much area, thus a further reduction in displacement volume.

It was observed that a sudden raising or lowering of the output line made only instantaneous changes in output from the tube and the system quickly stabilized. Observation of the output and flowmeter float showed a continual pulsation of the flow at all pumping speeds. At low speeds this pulsation produced a flow-no flow condition in the output which is not acceptable for a sprayer metering system.

This experiment showed that for most spray chemicals except the flowable formulations, a direct proportionality existed between pump speed and output. The relationship was good enough to preclude the use of calibration charts for obtaining an output rate within 10% of that desired at any speed. Flowable formulations required exact calibration curves because of their nonlinear output versus pump speed characteristics. Also indicated by this experiment was the need for some form of desurger to correct the flow-no flow conditions created by the pulsating output.

6.3. EFFECT OF PRESSURE HEAD

Adjusting the backing plates (Figure 6.1.1) of the pump so that the fingers completely occlude the tube should minimize reflux, thus making the pump positive displacement in nature. In a positive displacement pump, the output is dependent upon the speed of the pump, except for normal losses from internal leakage, or slip. In this case, slip will be due to incomplete tube occlusion. To evaluate slip in the pump, a needle valve and pressure gauge were installed in the tubing between the pump and flowmeter. Then, for a constant pump speed, the pressure was set by adjusting the needle valve. The pressure was varied from 0 to 25 psi in increments of 5 psi for the tests. Figures 6.3.1 and 6.3.2 show the results of this test for Cythion and Sevinmole 4. Because of pressure fluctuations of up to 1 psi caused by the pulsating output, it was impossible to obtain a steady reading on the pressure gauge. Therefore, an average reading was used. The fluctuations were all within

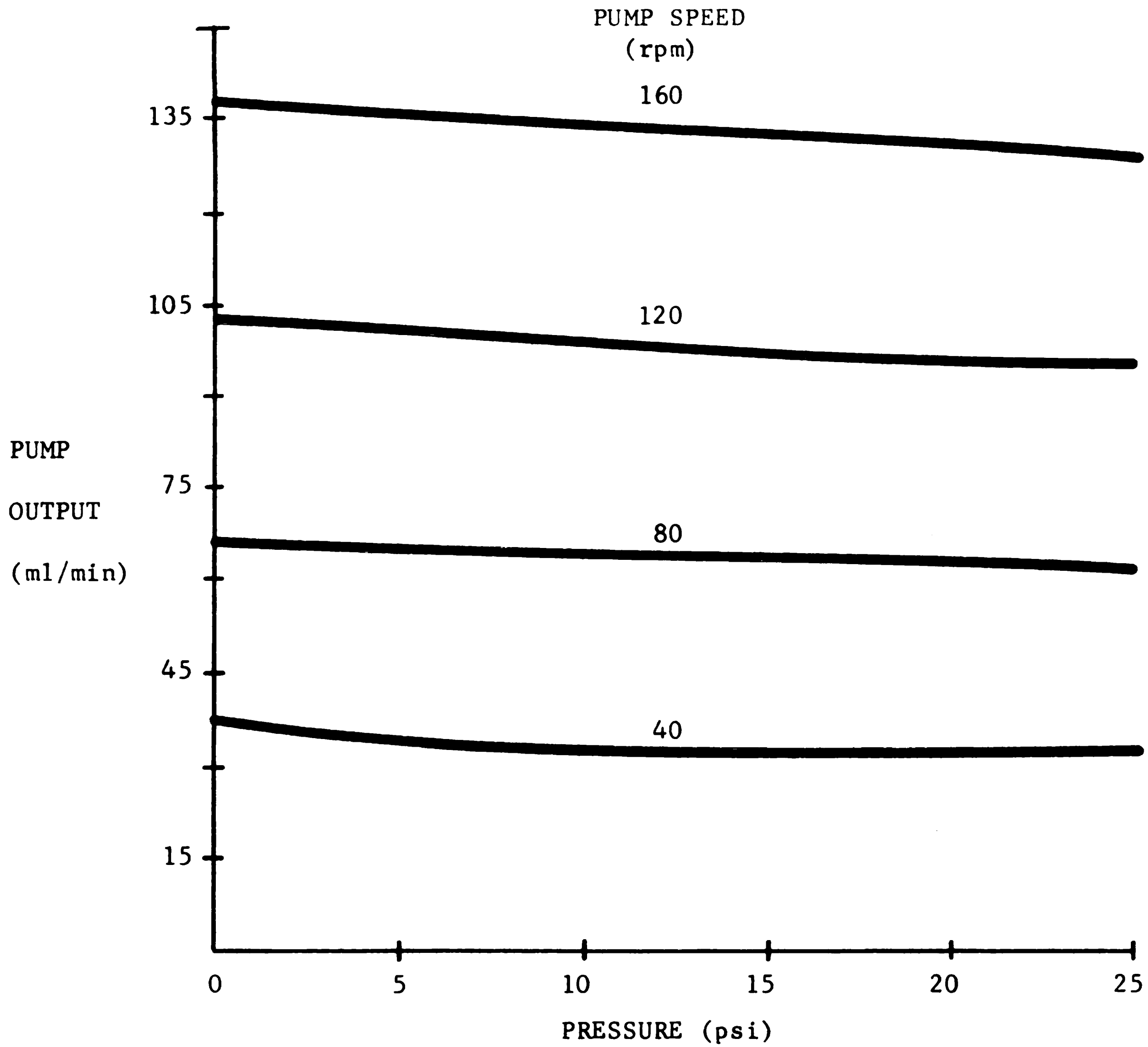


Figure 6.3.1. Output versus Pressure Head Using Cythion--Peristaltic Pump

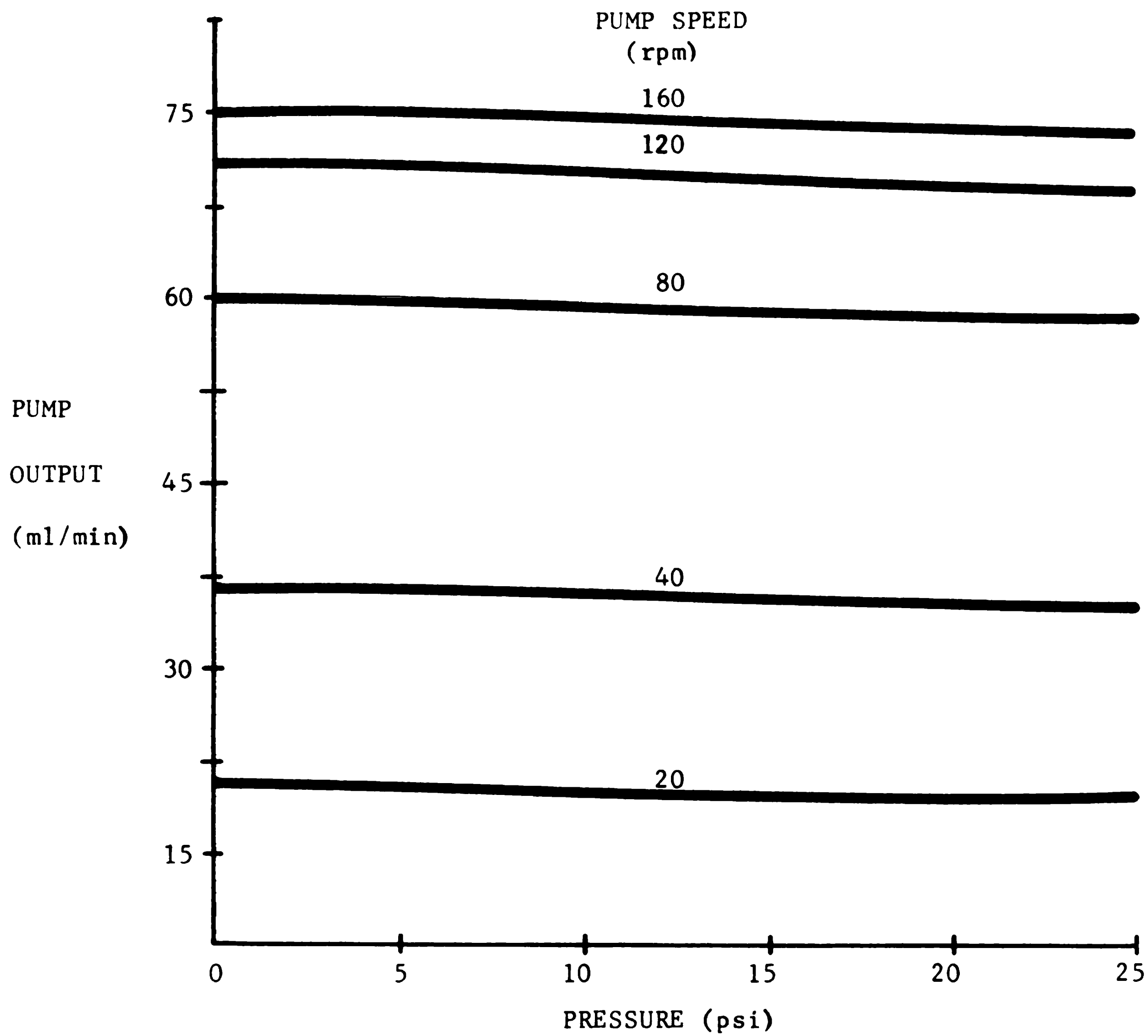


Figure 6.3.2. Output versus Pressure Head Using Sevinmole 4--Peristaltic Pump

$\pm\frac{1}{2}$ psi of the selected evaluation pressures. Output remained fairly constant for both materials at all pressure settings at low pump speeds. At higher pump speeds, a small amount of slip was observed at the higher pressures. The largest output loss observed was only 6.8% and occurred at the highest pressure. Some minor difficulties were encountered in maintaining a prescribed pressure reading with the Sevinmole 4 due to the impingement of discrete particles on the surface of the needle valve. An interesting observation made during this test was that the needle valve restriction in the line produced a nearly pulsation-free output for all materials tested.

6.4. DESURGING THE OUTPUT

The restriction caused by the needle valve in the previous experiment caused the conversion of velocity pulse, ΔV to a pressure pulse, ΔP. This pressure pulse took the form of a pressure wave which disturbed the system between the pump and the needle valve at a frequency equal to the pump speed. The pressure wave was absorbed by the elasticity of the tubing, friction forces in the tubing, and, to a small degree, fluid compressibility. The elasticity of the plastic tubing was the most significant factor in the pressure dissipation, as it visibly expanded and contracted with each pulse. To examine this tubing characteristic, another pressure gauge and an additional 50 feet of $\frac{1}{4}$-inch Tygon tubing were added to the system between the pump and the needle valve (Figure 6.4.1). Repeating the procedure used in the output versus pressure head investigations and observation of the flowmeter float and pressure gauges indicated that as much as 10 psi of pressure wave could be dissipated by 50 feet of $\frac{1}{4}$-inch Tygon tubing.

Another method for dispersing the pressure wave was attempted by installing an air chamber in the line between the needle valve and the pump (Figure 6.4.2). The air chamber consisted of a vertical 6-inch length of 3/8-inch pipe capped at the top and open to the output line at the bottom. The change in air volume in the chamber for a given pressure was calculated using the relationship.

$$P_1V_1 = P_2V_2 \tag{6.4a}$$

since the changes are essentially isothermal. The pressure pulse ΔP produced a change in air volume ΔV that is found by substituting $P_2 = P_1 + \Delta P$ and $V_2 = V_1 - \Delta V$ into Equation 6.4a yielding

$$\Delta V = V_1 - \frac{V_1P_1}{P_1 + \Delta P} \tag{6.4b}$$

The between-pulse pressure drop was eliminated as the compressed air expanded to equalize liquid and air pressure. The volume of air in the chamber at atmospheric pressure using inside pipe dimensions was 27.4 milliliters. Applying a pressure of 5 psi to the chamber theoretically reduced the volume to 20.5 milliliters. Applying a 1 psi pressure pulse, caused by the pulsation of the pump output, to the chamber which was under a constant 5 psi load produced an instantaneous theoretical air volume change of about 1 milliliter which is sufficient to accommodate the displacement of the pump for one revolution thus allowing an even distribution of the output over the entire pumping cycle. A check of the flowmeter float and the output line indicated the output was pulsation-free.

The essentially fixed-orifice flow impediment supplied by the needle valve was unsatisfactory because the pressure increased on the

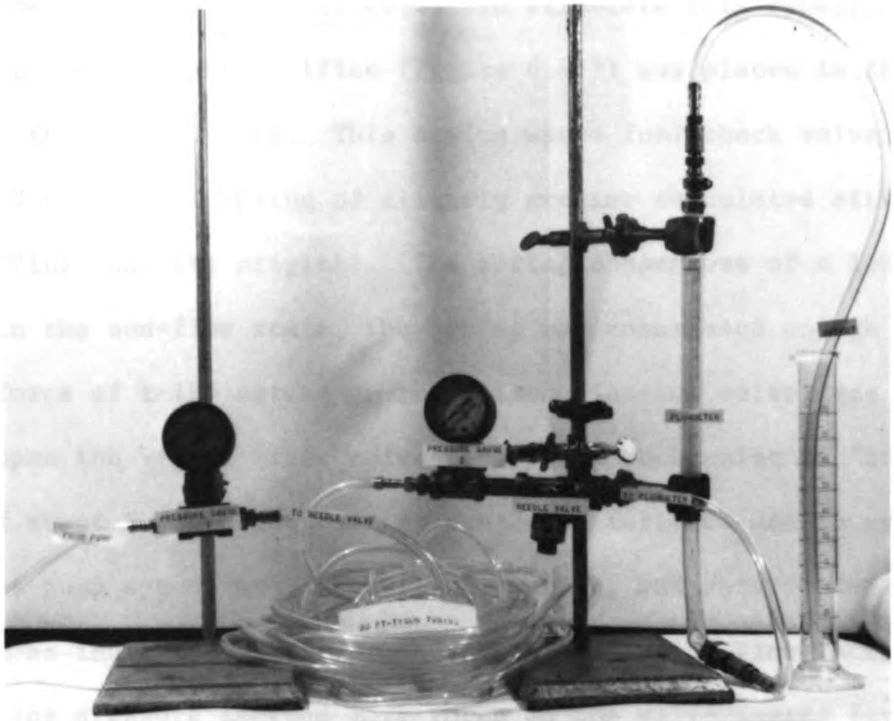


Figure 6.4.1. Pressure Dissipation Test Setup.

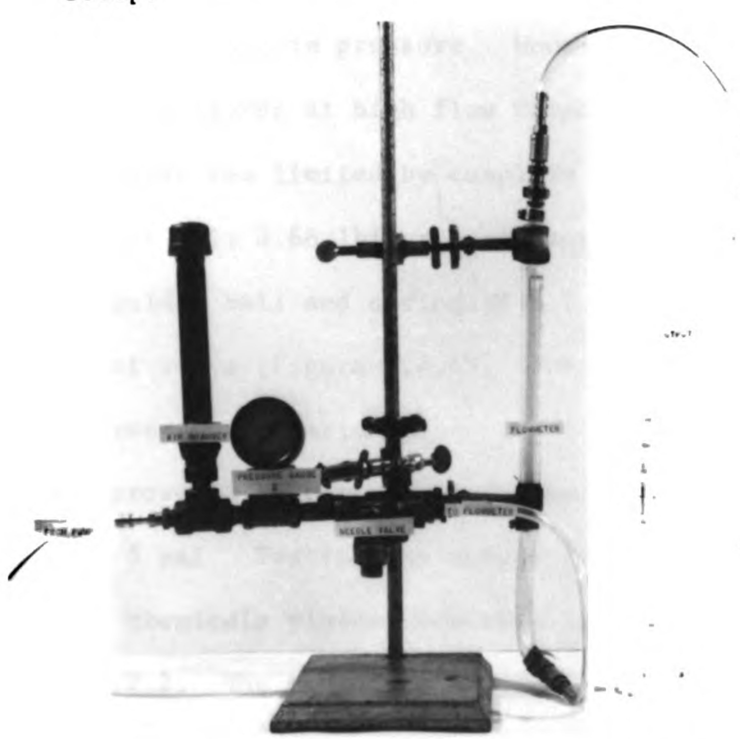


Figure 6.4.2. **Pressure Dissipation Test Setup Using an Air Chamber.**

system as the flow rate was increased. To alleviate this condition, a pressure-accuated variable orifice (Figure 6.4.3) was placed in the line in place of the needle valve. This device was a fuel check valve which was modified by using a spring of slightly greater calculated stiffness (K = 3.3lbf/in) than the original. The spring chosen was of a length such that in the non-flow state, the spring was compressed enough to require a force of ¼-lbf acting on the ¼-inch diameter valve face to initially open the valve. This corresponded to an opening or "cracking" pressure of about 5 psi. The testing procedure followed was to gradually increase the pump speed, and thereby, the output, and observe the change in pressure as indicated by the pressure gauge. As the flow increased, the increasing pressure exerted more force on the valve spring forcing the valve open further. This system allowed the flow rate to be increased with a relatively small increase in pressure. However, this system still produced unsatisfactory pressures at high flow rates since the flow area in the modified check valve was limited by complete spring compression which required a force of only 0.66 lbf or a pressure of 13.5 psi. A valve consisting of a guided ball and spring (K = 1.42 lbf/in) was installed in place of the original valve (Figure 6.4.4). The testing procedure was the same used in the preceding experiment.

Because of the improved flow channel in the ball valve the maximum line pressure was only 5 psi. Testing the output versus pump speed for the representative chemicals yielded essentially the same results as Figures 6.2.1 and 6.2.2. The maximum 5 psi line pressure caused only about a 3 percent output reduction at high pump speeds. Figure 6.4.5 shows the final peristaltic pump system configuration.

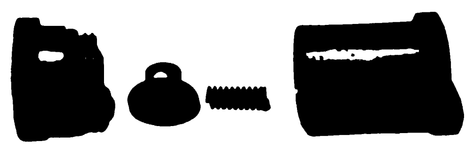


Figure 6.4.3. Modified Fuel Line Check Valve.

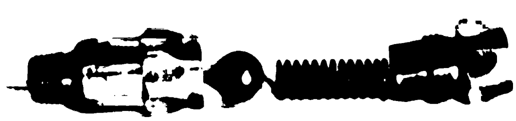


Figure 6.4.4. Ball Check Valve.

Several conclusions were drawn from these tests. The additional length of tubing did dissipate pressure, but to use this method for a sprayer would have required enough extra tubing for at least four lines or about 200 feet of tubing.

Also the expansion caused by the pressure wave caused an accelerated deterioration of the tubing. This was due to the exposure of the strained tubing fibers to the xylene base of most of the spray chemicals. Xylene is a mild solvent for Tygon tubing but does not seriously affect the tubing in an unstrained state. The original variable orifice used did not perform well with Sevinmole 4 at low flow rates. Its construction (Figure 6.4.3) was that of a cylinder within a cylinder which created a large contact surface with close clearance. Discrete particles lodged on the surface disrupting normal orifice operation.

6.5. FIELD TESTS

The peristaltic metering system was mounted on a Sprayall concentrate sprayer (Figure 6.5.1) for field testing on experimental plots maintained by the Michigan State University Departments of Entomology and Plant Pathology. Both registered and experimental chemicals were applied to a variety of fruit crops. Metering system tests were made only when applying a chemical whose effectiveness for a given pest had already been proven. Tests were conducted only on days when spraying conditions were ideal so that the metering system could be evaluated without consideration of factors pertaining to the rest of the delivery system, i.e. the fan and nozzles. The procedure followed was to calibrate the metering system to the prescribed dosage by catching the total output for a one minute interval in a graduated cylinder

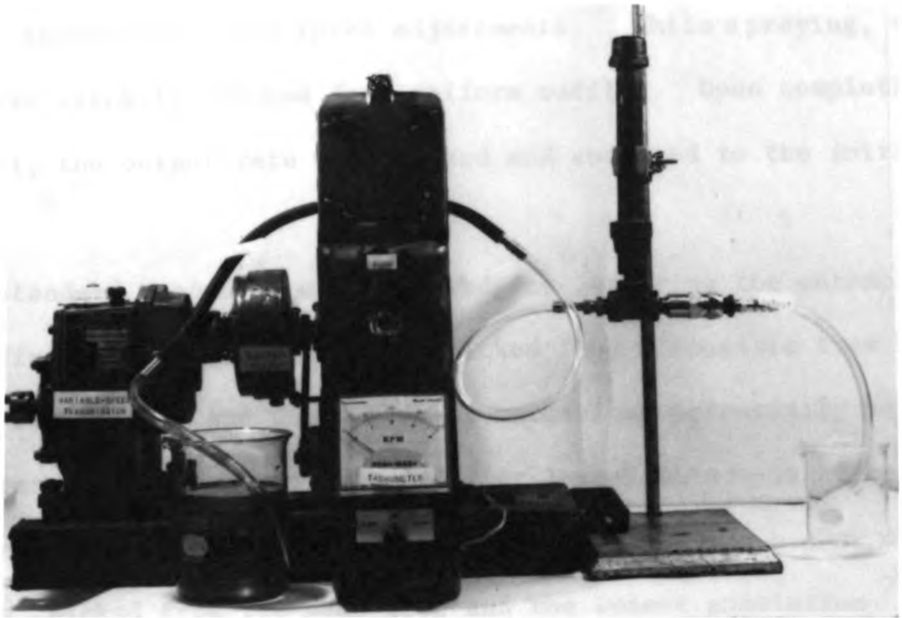


Figure 6.4.5. Complete Peristaltic Pump System.

Figure 6.5.1. **Peristaltic Pump System Mounted** on Sprayall Sprayer.

and making appropriate pump speed adjustments. While spraying, the nozzles were visually checked for uniform outflow. Upon completion of the plot, the output rate was checked and compared to the initial output.

The standard procedure was followed for gathering the entomological data. A fixed number of leaves was picked from a specific tree before the spray was applied and the number of harmful or potentially harmful insects present was counted. At 3-day or 1-week intervals after application, depending on the chemical applied, an equal number of leaves was picked from the same tree and the insect population tabulated. These consequent insect counts were compared with the pre-spray counts, showing the effectiveness or ineffectiveness of the spray application.

The results of the field testing were very encouraging showing excellent uniform protection of cherries against cherry leaf spot, a fungus disease, with a rate of chemical application of only 8 ounces per acre of a pure liquid fungicide formulation. This particular rate reduces to a metered output of 6 ml/min for the given spray conditions. The initial output was 6.2 ml/min and after spraying for 20 minutes an output check showed 6 ml/min or a reduction of only 3.3 percent. Another application for the control of a small insect called psylla on pears required a metered output of 72 ml/min. Initial output was 72 ml/min and the after-spray check was 70 ml/min or a decrease of 2.8 percent. The nozzles showed a uniform outflow during the entire spray period. Entomological data (Table 3) indicated a substantial reduction in the psylla population.

TABLE 3.--Results of Field Testing

Experiment: 71-21
Location: Trevor Nichols Farm, Fennville, MI
Crop: Bartlett Pears
Methods: One acre, non-replicated plots employed. A total of 10 spurs with 4-6 leaves per spur were sampled from each plot. The psylla nymphs were counted under a binocularscope. The materials were sprayed with Beecomist dispensers fitted with metal sleeves with a controlled porosity of 20μ. The dispensers were mounted in a spray-all airblast sprayer modified for ULV purposes.
Date of Treatment: September 4, 1971

Material	Rate of Formulation per Acre	Eggs and Nymphs per Spur on Dates Listed					
		9/2		9/13		9/20	
		*e	**n	e	n	e	n
Perthane 4 EC	3 qts.	284	61	30	12	19	8
Perthane 4 EC	1.5 qts.	240	107	9	31	42	23
Perthane 4 EC	0.75 qts.	96	90	57	117	58	83
CONTROL		77	101	63	57	48	72

* e = eggs

** n = nymphs

Source: Michigan State University, Dept. of Entomology, 1971

VII. GEAR PUMP SYSTEM

7.1. INTRODUCTION

A Zenith model 4391-B external spur gear pump (Figure 7.1.1) was also evaluated as a possible metering system. Like the peristaltic pump, the gear pump is a very accurate metering pump whose output can be directly related to pump speed. The theoretical displacement of the Zenith pump calculated with Equation (4.1) is 1.18 ml/rev. The specifications of the model 4391-B pump give the displacement per revolution as 1.168 ml/rev at 100 psi. Factors that account for this difference between theoretical and specified displacement include imperfect gear mesh, clearance volume, and pressure relief grooves between inlet and discharge.

7.2. EFFECT OF PUMP SPEED

To evaluate the actual output versus pump speed, an apparatus similar to that used for the peristaltic pump output test was used. The pump speed was controlled with a variable speed transmission. One-quarter inch Tygon tubing was used for both inlet and output lines (Figure 7.2.1). A flowmeter was installed in the output line as an indicator of the flow uniformity. The procedure was the same as for the peristaltic pump speed test. The pump speed was set and the system was allowed to stabilize. The total output was collected in a graduated cylinder for a one minute interval. The flowmeter float and the end of the output tube were observed to determine the uniformity of the output.

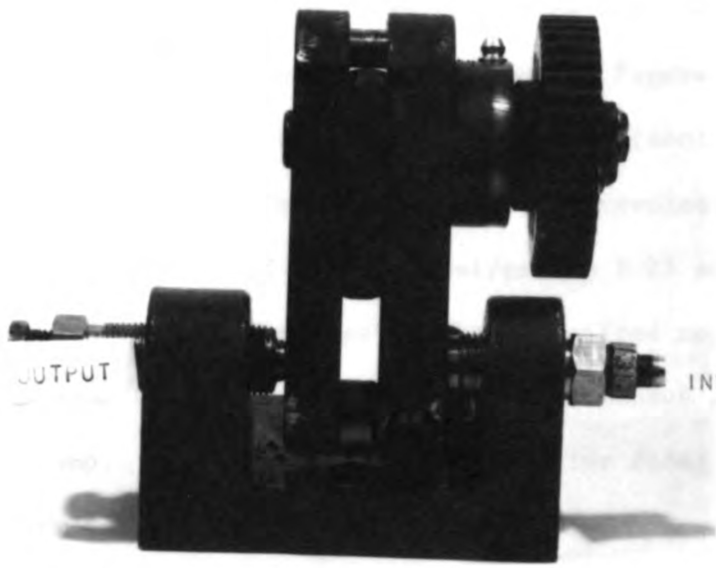


Figure 7.1.1. Zenith External Spur Gear Pump.

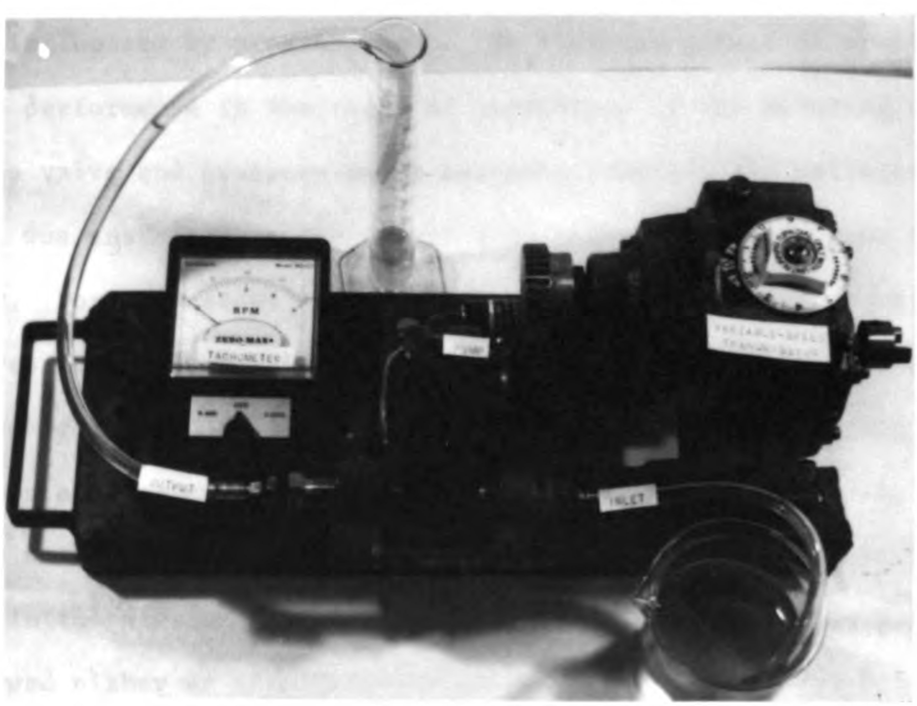


Figure 7.2.1. **Output versus Speed Test Setup for the Gear Pump.**

The output versus pump speed characteristics for the Zenith gear pump and the representative chemicals are shown in Figures 7.2.2 and 7.2.3. The output characteristics were essentially identical for both pure liquids and flowables. The displacement per revolution for both Cythion and Sevinmole 4 varied from 1.11 ml/rev to 1.23 ml/rev in a 15 to 120 rpm speed range. The greater than specified capacity is possibly due to pump speeds in excess of the recommended pumping speeds for this model pump. Observation of the flow meter float and tube end indicated uniform output for all materials.

7.3. EFFECT OF PRESSURE HEAD

A gear pump is considered a positive displacement pump and, therefore, output should depend upon pump speed except for leakage losses, or slip, which is influenced by pressure head. To test the effect of pressure on gear pump performance in the range of conditions of the metering system, the needle valve and pressure gauge assembly used for the peristaltic pump test was installed in the output line between the gear pump and the flowmeter. For a constant pump speed, the needle valve was adjusted to give a prescribed pressure and the output was measured. The pressure was varied from 0 to 25 psi in 5 psi increments. Results of this test for Cythion and Sevinmole 4 are shown in Figures 7.3.1 and 7.3.2. There was no measured pressure effect on output for either class of material in the pressure range tested. No pulsation of the output was observed either at the flowmeter or the output tube. The laboratory tests of the gear pump system were completed too late in the season for a field test of the system on experimental plots.

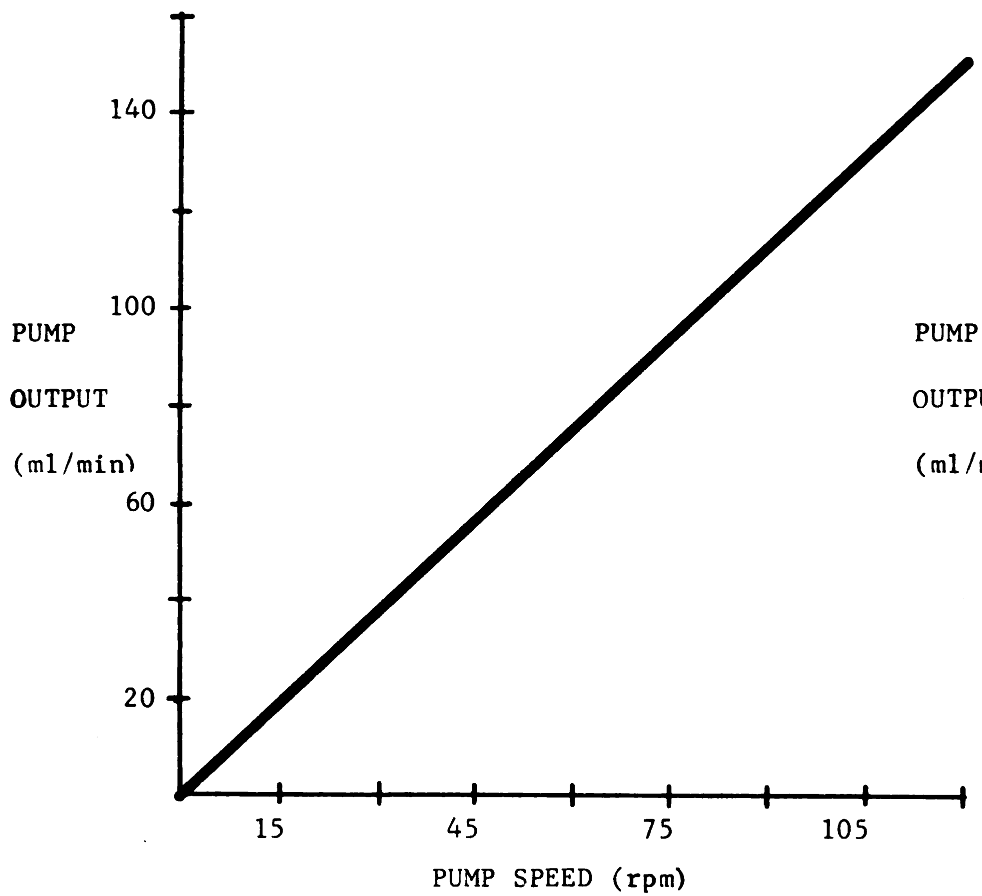


Figure 7.2.2. Output versus Pump Speed Using Cythion--Gear Pump.

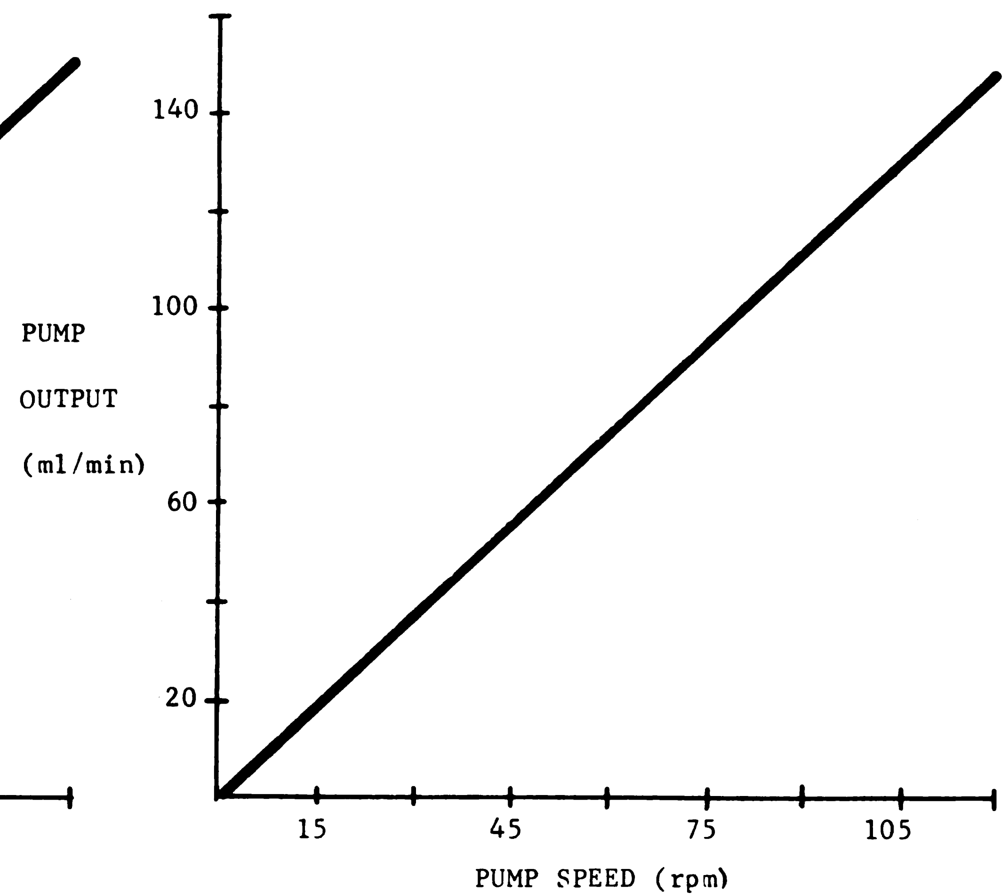


Figure 7.2.3. Output versus Pump Speed Using Sevinmole 4--Gear Pump.

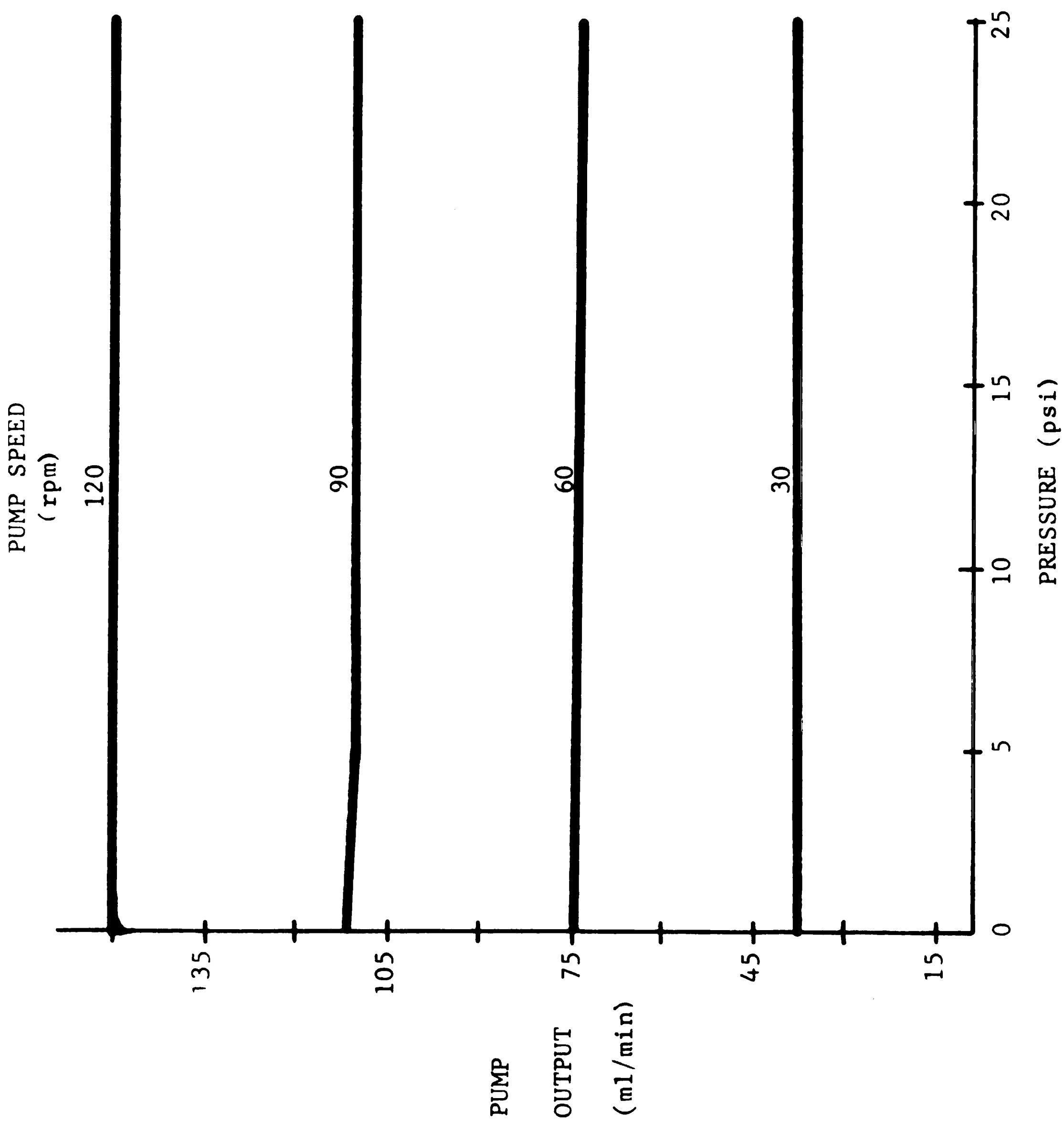


Figure 7.3.1. Output versus Pressure Head Using Cythion--Gear Pump.

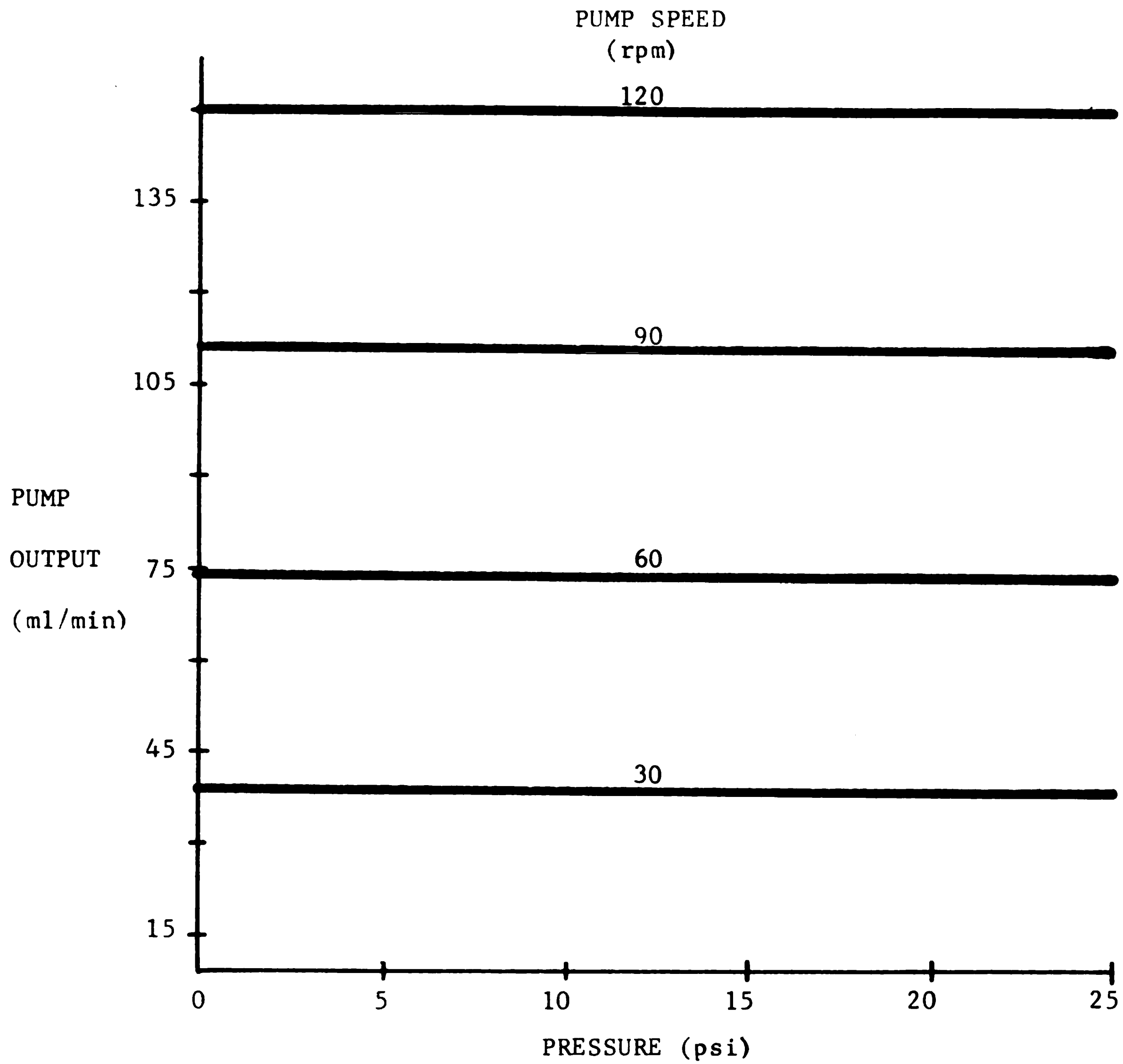
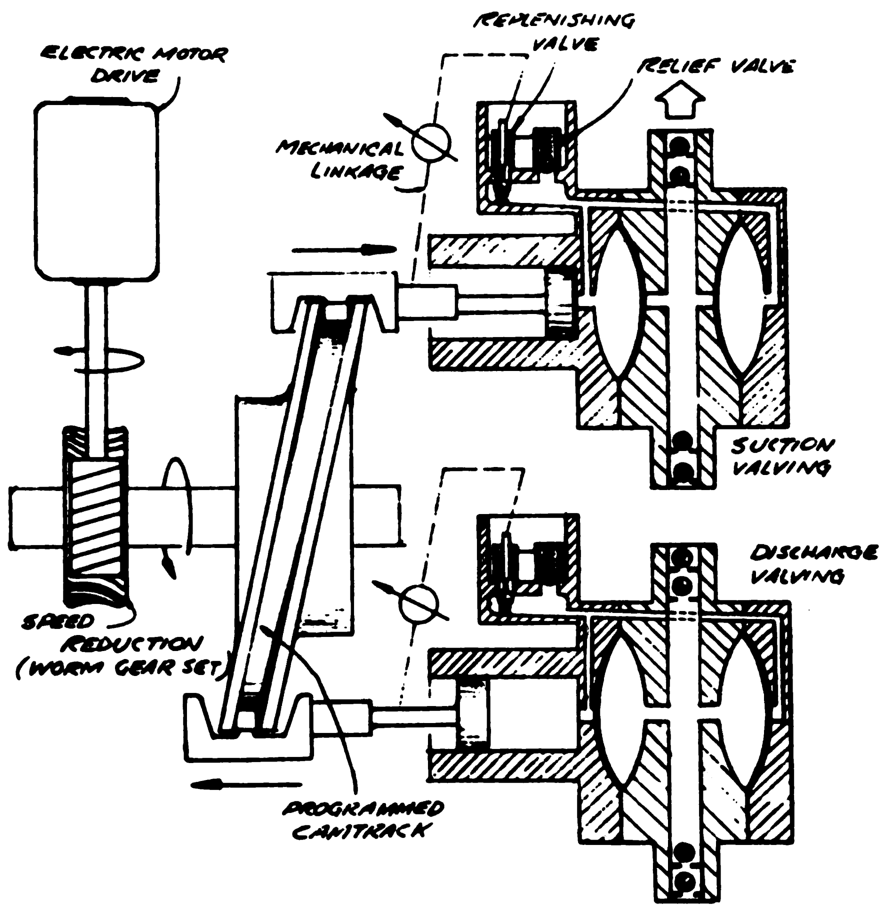


Figure 7.3.2. Output versus Pressure Head Using Sevinmole 4--Gear Pump.

VIII. OTHER PUMP SYSTEMS

Two other pump types were investigated as possible metering systems. A Jaco mechanically-actuated diaphragm pump(Figure 4.1.2) was considered. At the constant speed of 40 spm, the output could be varied from 0 to 63 ml/min. Output was controlled by "lost motion" type stroke adjustments. A Milton Roy "Minipump" reciprocating piston pump(Figure 4.1.1) was also considered. At a constant speed of 96 spm, the output was varied from 0 to 50 ml/min again using the "lost motion" stroke adjustment principle. Both of these pumps exhibited a flow-no flow output condition for all stroke adjustments thus necessitating desurging equipment. An interesting solution to this problem was proposed by Cleary and Bauer(1968). Using the full stroke, a programmed cam was used to operate two pump units ganged together to produce a pulsationless output(Figure 8.1.1). Output quantity was controlled by varying the speed of cam rotation. The programmed cam provided output over 220° of the pumping cycle,compared to the traditional 180°, which allowed overlap of the outputs of the two pumps. Although this system provided ideal output characteristics, it was economically unfeasible. Since two pumps were needed per output line, a minimum of eight pumps would be needed to outfit a sprayer which made the system cost-prohibitive. Use of the desurging equipment developed for the peristaltic pump system and a variable speed transmission with the consequent reduction to four pumps per sprayer was still too costly.



Source: Cleary and Bauer, 1968.

Figure 8.1.1. Pulsationless Flow Drive.

IX. DISCUSSION AND CONCLUSIONS

All of the original objectives of the project have been deomonstrated for both the gear pump and peristaltic pump systems except mechanical simplicity and economic compatibility. Maintenance of the peristaltic system included periodic replacement of the small section of chemical-resistant Viton tubing which was subjected to constant mechanical manipulation by the steel fingers. The pressure-accuated variable orifice should be periodically disassembled and cleaned to insure against deposits which may build up on the ball or its seat. Minor maintenance included lubrication of moving parts. The gear pump required no lubrication except for the drive gear shaft since the liquid it pumped provided sufficient internal gear lubrication for the pressures used. However, it is imperative that the pump be flushed with a clear liquid before any prolonged idle period since particulate material that might dry on the interior surfaces of the pump could cause pump failure because of the very close clearances.

The peristaltic pump system would have a lower initial cost than the gear pump system. A Sigmamotor peristaltic pump of the type evaluated here has a provision for up to six output lines at once which makes one pump unit sufficient for a sprayer. The gear pump ($80.00) is less expensive than the peristaltic pump ($120.00) but can only supply one output line. Thus, at least four gear pump units are needed to outfit a sprayer making the initial cost of the gear pump system comparatively high. However, maintenance of the desuring equipment and periodic replacement of the very expensive chemical-resistant Viton tubing ($3.00 per foot) over the lifetime of the peristaltic pump tends to equalize the two systems economically in the long run.

The gear pump showed superior performance to the peristaltic pump in the output tests. For a speed range of 15 to 120 rpm, the gear pump displacement per revolution varied from 1.11 ml/rev to 1.23 ml/rev for both Cythion and Sevinmole 4, while the peristaltic pump for the comparable speed range of 20 to 160 rpm showed a variation of 0.97 ml/rev to 0.85 ml/rev for Cythion and dropped from 1 ml/rev to 0.43 ml/rev with Sevinmole 4.

Pressure head had a slight effect on the output of the peristaltic pump at high speeds and pressure. However the maximum reduction in output was only 6.8 percent. Pressure head effects on the gear pump output were negligible, the maximum observed reduction being 2.0 percent.

On the basis of these observations, the gear pump system should be considered seriously as an alternative to the peristaltic pump system. Therefore, a program of field testing for the gear pump system is recommended. Field test of the peristaltic pump system verified the results obtained in laboratory tests and a similar outcome is expected for the gear pump system.

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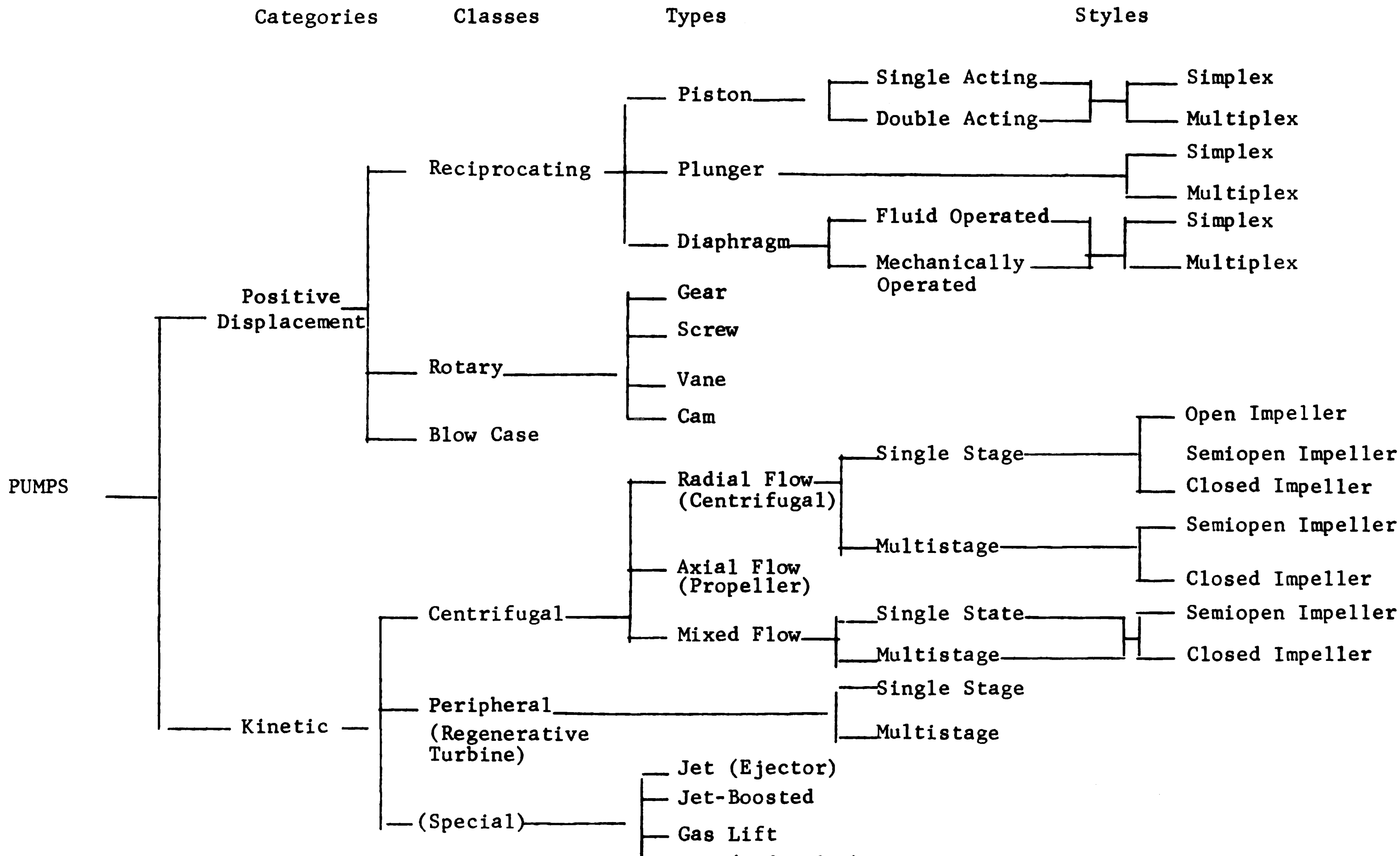
APPENDIX

APPENDIX A CLASSIFICATIONS OF PUMPS



Source: A.I.Ch.E. 1960.

APPENDIX B METERING PUMP MANUFACTURERS

Manufacturer	Type	Range	Max Pressure (Psig)
Afton Engineering	Reciprocating piston	0.06-12.6 to 4-800 gph	10,000
American Instrument Co.	Reciprocating piston	0-0.97 to 0-31.21 gph	60,000
American Meter Controls	Reciprocating piston	0-0.65 to 0-812 gph	10,000
B-I-F Industries	Reciprocating piston	0-1 ml/hr to 0-2652 gph	15,000
	Diaphragm	0-0.5 m./hr to 0-33 gph	125
Beckman/Spid	Reciprocating piston	0-2 to 0-20 ml/min	
Beckman/Spinco	Reciprocating piston	3-180 to 3-700 ml/hr	1,000
Bendix Corp.	Rotary, spur gear	0.5-1.5 to 0.5-15 gph	1,000
Utica Div.	Rotary, roller-vane	0-1 to 0-15 gph	1,000
Blackmer Pump Co.	Rotary,sliding vane	0-19 to 0-450 gpm	125
Blue White Industries	Diaphragm	0-1 gph	100
Davies Brothers	Rotary, annular piston	0-44 to 0-700 gph	1,500
Denison Engineering Div.	Reciprocating piston	0-3.3. to 0-917 gpm	3,500
Electro-Mechanical	Peristaltic	0-3 ml/min to 1 ml/24 hrs	15
Fischer & Porter Co.	Diaphragm	Up to 0-4.6 gph	125
Fluid Metering,Inc.	Reciprocating and rotating piston with milled slot; diaphragm	0-1 ml/stroke to 0-960 gph	3,000
Greiner Scientific Corp.	Peristaltic	0-5 liters/min	
Hanson-Van Winkle	Reciprocating piston	0-5 gph	
Harvard Apparatus Co.	Peristaltic	0.0042-800 ml/min	6
Hills-McCanna Co.	Reciprocating piston	0-0.8 to 0-980 gph	5,000
	Diaphragm	0-6 ml/min to 0-6 gph	2,500
International Medical	Peristaltic	0.005-10,000 ml/min	
Kenelco,Inc.	Peristaltic	0-8 to 0-160 ml/hr	
LKB Instruments	Reciprocating piston	0-4 to 0-187 ml/hr	60
Lapp Insulator Co.	Reciprocating piston with diaphragm	0-3 ml/hr to 0-12 gpm	5,000
Madden Corp.	Reciprocating piston with diaphragm	0-2 to 0-60 gph	1,250
Manzel Div.	Reciprocating piston	0-60 ml/hr to 0-60 gph	5,000
Houdaille Industries	Diaphragm	0-1.5 to 0-15 gph	125

APPENDIX B (CONT.)

Marton Equipment Co.	Reciprocating piston Air-driven diaphragm	0-100 ml/hr to 0-100 gph 0-50 ml/hr to 0-500 gph	2,400
McFarland Engrg.	Reciprocating piston	0-0.48 to 0-535 gph	25,000
Milton Roy Co.	Reciprocating piston	0-0.1 to 0-938 gph	7,500
National Instrument Co.	Reciprocating piston	drops/stroke to 0-43 gpm	
Neptune Chemical	Reciprocating piston	0-0.5 to 0-120 gph	1,500
Noble & Wood Machine Co.	Reciprocating piston	0-0.16 to 0-2.4 gph	13
Process & Instruments Corp	Peristaltic	0.07-1.4 to 1.5-45 ml/min	
Research Appliance Co.	Bellows	0-15 to 0-25,000 ml/hr	100
Ruska Instrument Corp.	Single-shot piston (supplies constant flow rate when driven by precision lead screw)	0-1 ml/hr 0-1200 ml/hr	50,000
Sage Instruments	Single-shot piston actuated by internally generated gas Reciprocating pistons (3)	0.05 to 50 ml/hr 0-0.3 to 0-102 ml/min	
Science Sales International	Peristaltic	0 to 6 liters/min (90 gph)	18
Sigmamotor, Inc.	Peristaltic (fingers) Peristaltic (plate)	0.005 ml/min to 250 gph 0.025 to 150 ml/min	20
Stewart-Warner Corp.	Reciprocating piston	0-16 ounces/stroke	430
Tuthill Pump Co.	Rotary, spur gear	0-28 to 0-144 gph	100
Vanton Pump	Peristaltic	0-40 gpm	60
Wallace & Tiernan Inc.	Reciprocating piston Diaphragm	0.4 to 800 gph 1.3 to 41.6 gph	4,000 125
Whitey Research	Diaphragm	0-0.56 to 0-10 liters/hr	5,000
Zenith Products Co.	Rotary, spur gear	0.297 to 2.92 ml/rev.	1,000

Source: Marton, 1963.

APPENDIX C CLASSES OF ROTARY PUMPS

Classes — Types — Styles

- Rotary Pumps
  - Single Rotor
    - Vane
      - Blade, Bucket, Roller, or Slipper
        - Vane in Rotor
          - Constant Displacement
            - Unbalanced
            - Balanced
          - Variable Displacement
        - Vane in Stator
          - Balanced
          - Unbalanced
    - Piston
      - Axial
        - Constant Displacement
        - Variable Displacement
      - Radial
        - Constant Displacement
        - Variable Displacement
          - Single Piston
          - Multi-Piston
    - Flexible Member
      - Flexible Tube
      - Flexible Vane
      - Flexible Liner
    - Screw
  - Multiple Rotor
    - Gear
      - External
        - Spur
        - Helical
        - Herringbone
          - Timed
          - Untimed
      - Internal
        - Crescent
        - No Crescent
    - Lobe
      - Single
      - Multiple
    - Circumferential Piston
      - Single
      - Multiple
    - Screw
      - Timed
      - Untimed

Source: Hydraulic Institute, 1965.

APPENDIX D TEMPERATURE INFLUENCE ON VISCOSITY

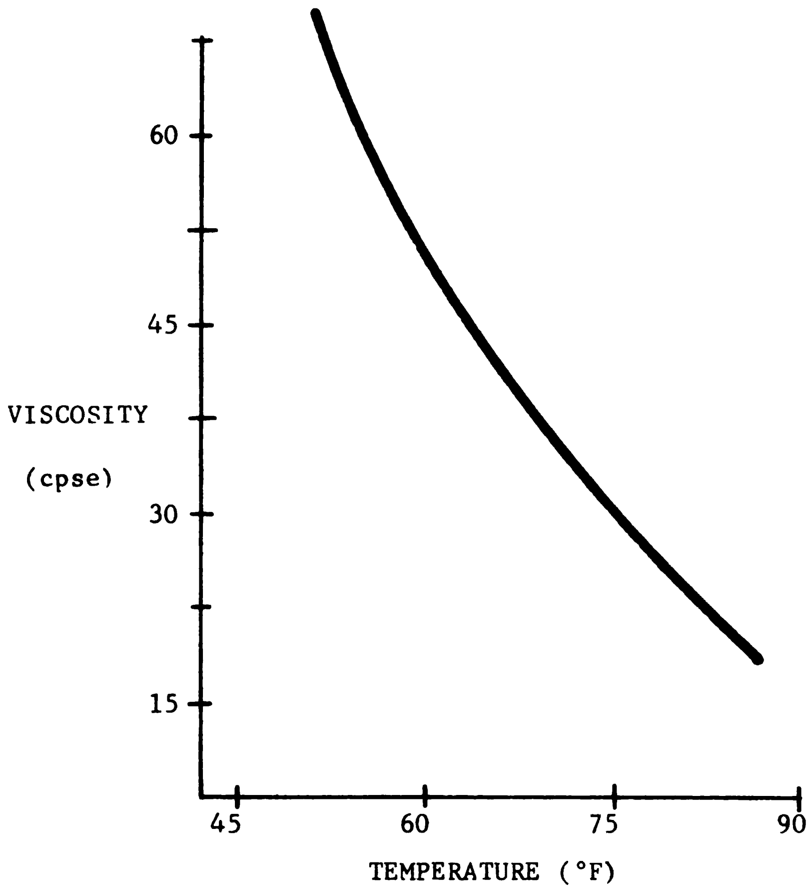


Figure D.1. Viscosity versus Temperature At Constant Shear Rate--Cythion.

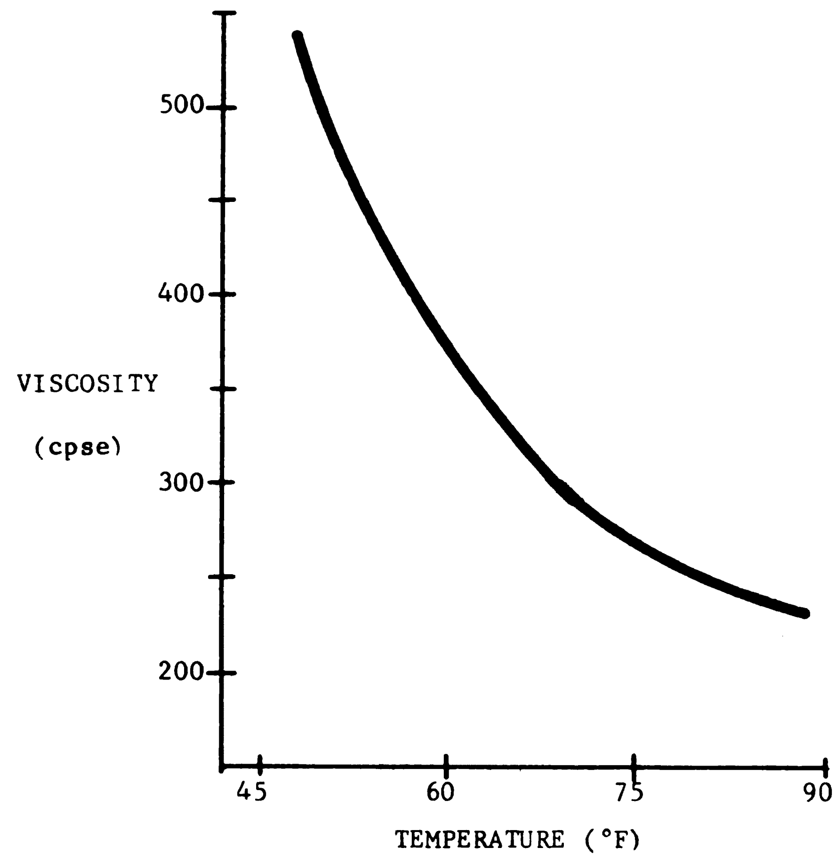


Figure D.2. Viscosity versus Temperature At Constant Shear Rate--Sevinmole 4

APPENDIX D (CONT.)

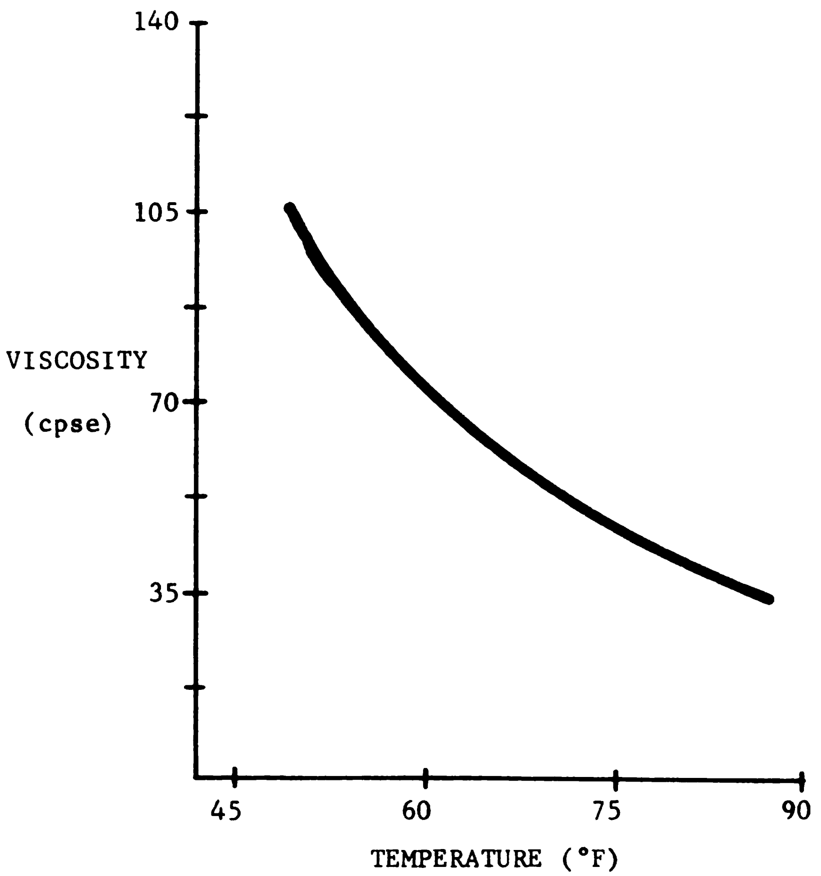


Figure D.3. Viscosity versus Temperature At Constant Shear Rate--Omite 6E.

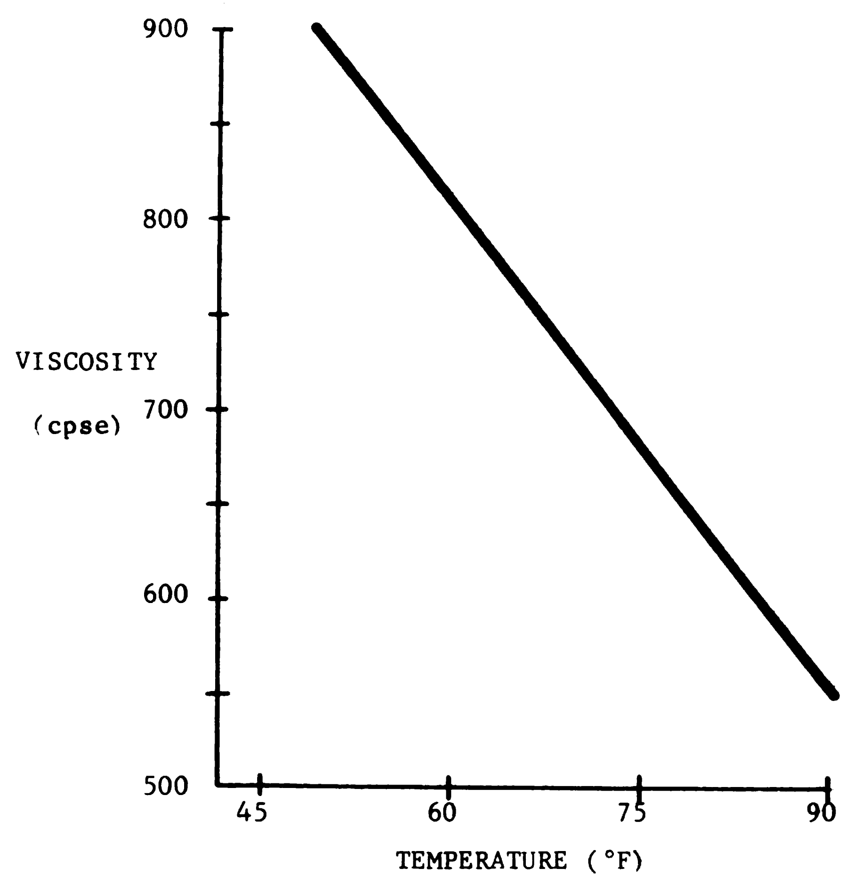


Figure D.4. Viscosity versus Temperature At Constant Shear Rate--Sevin 4LS.

APPENDIX D (CONT.)

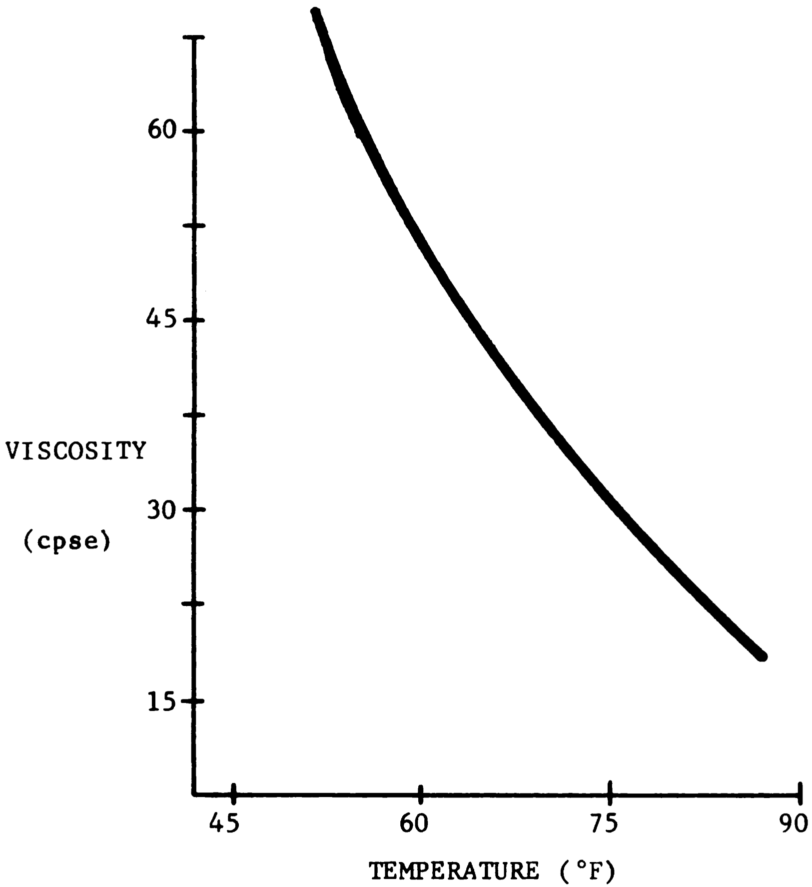


Figure D.5. Viscosity versus Temperature At Constant Rate of Shear--Dioctyl Pthalate(DOP).

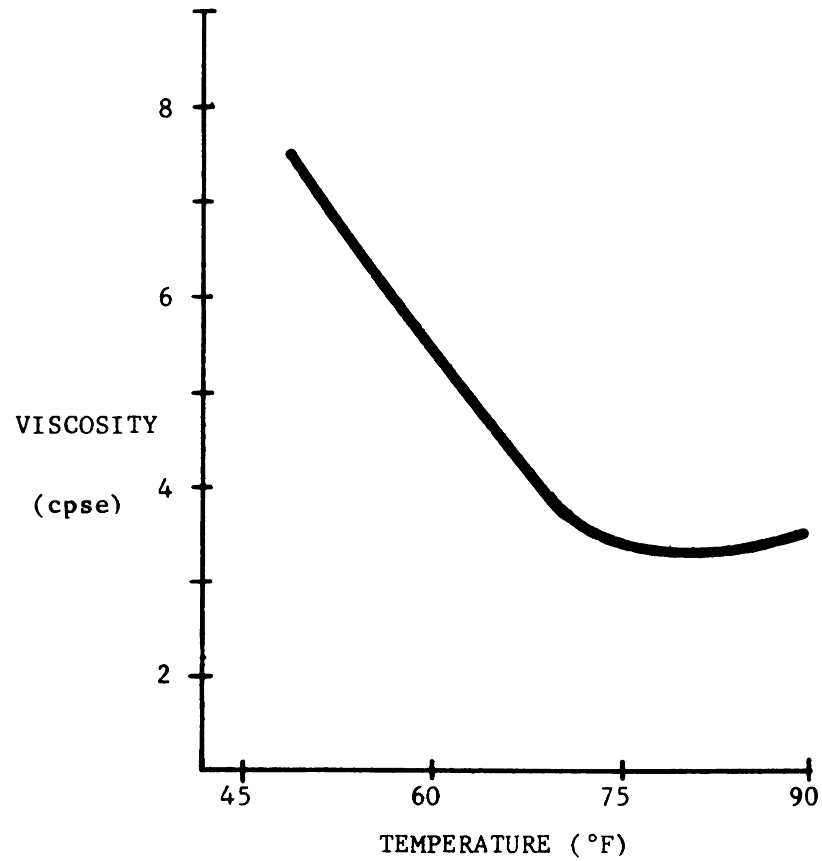


Figure D.6. Viscosity versus Temperature At Constant Rate of Shear--Perthane 4EC.

APPENDIX E OUTPUT CHARACTERISTICS--PERISTALTIC PUMP

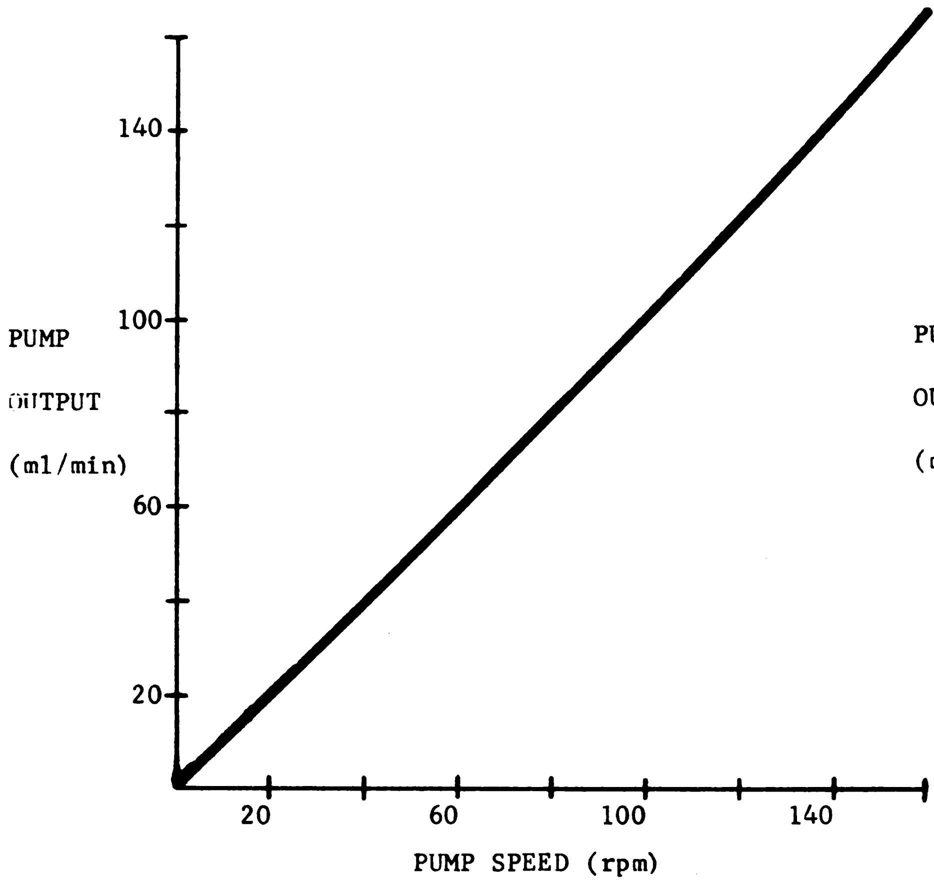


Figure E.1. Output versus Pump Speed--Perthane 4EC.

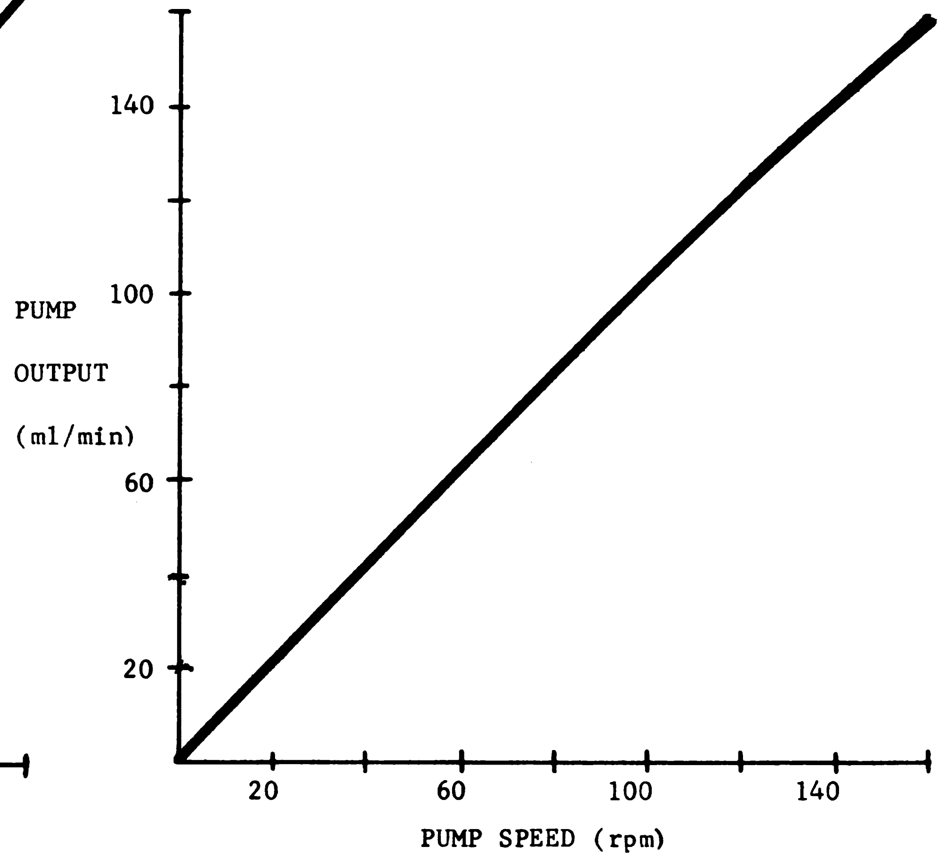


Figure E.2. Output versus Pump Speed--Dioctyl Pthalate (DOP)

APPENDIX E (CONT.)

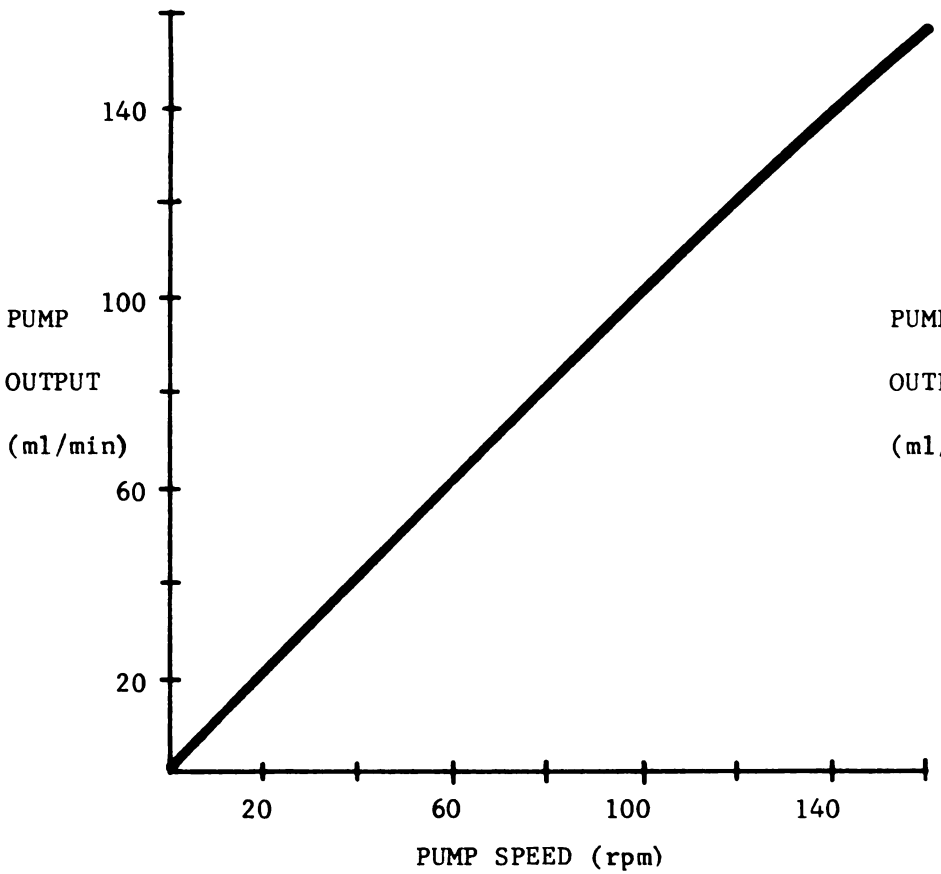


Figure E.3. Output versus Pump Speed--Omite 6E.

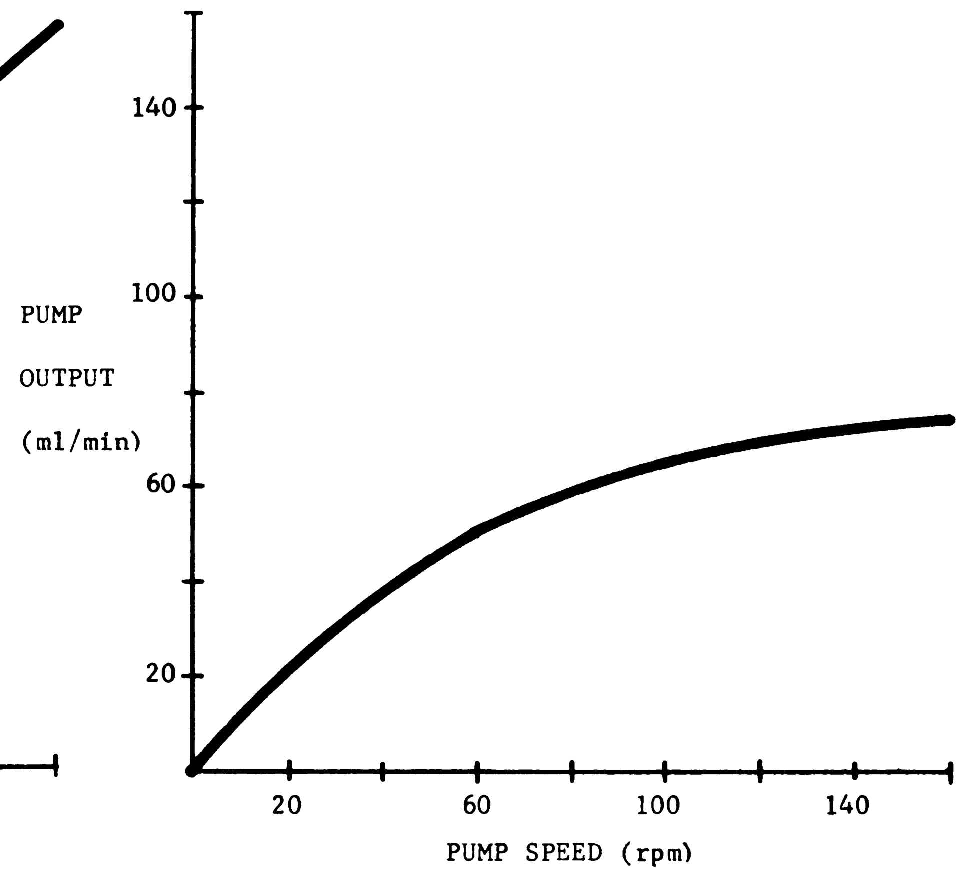


Figure E.4. Output versus Pump Speed--Sevin 4LS.

APPENDIX F OUTPUT CHARACTERISTICS--GEAR PUMP

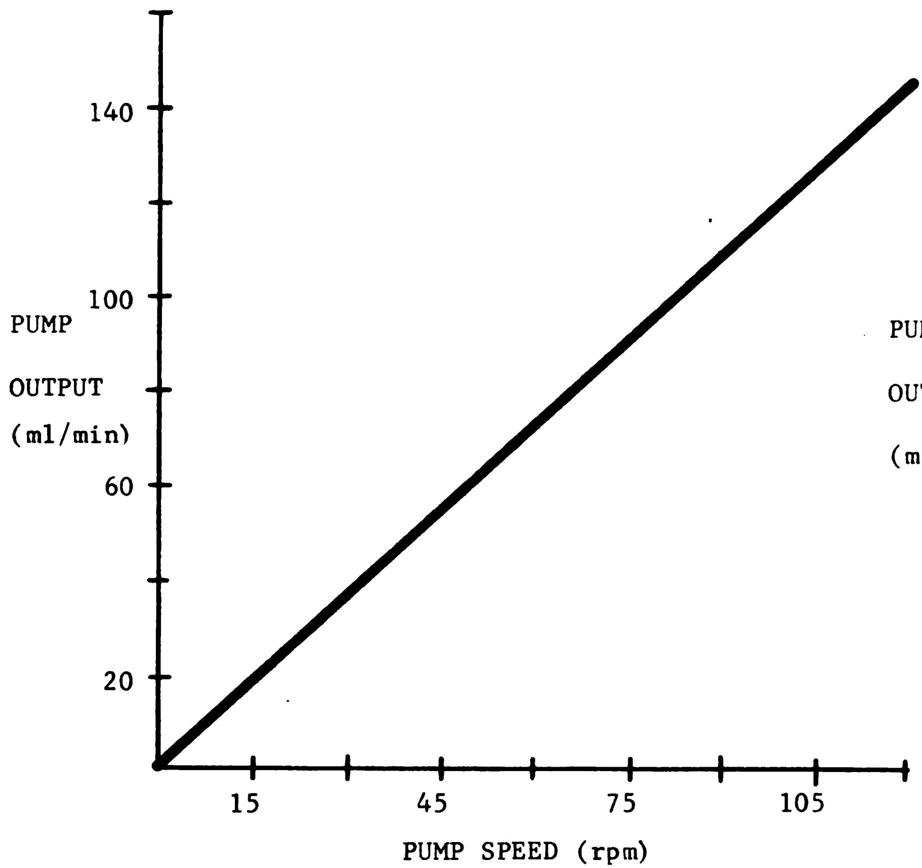


Figure F.1. Output versus Pump Speed--Perthane 4EC.

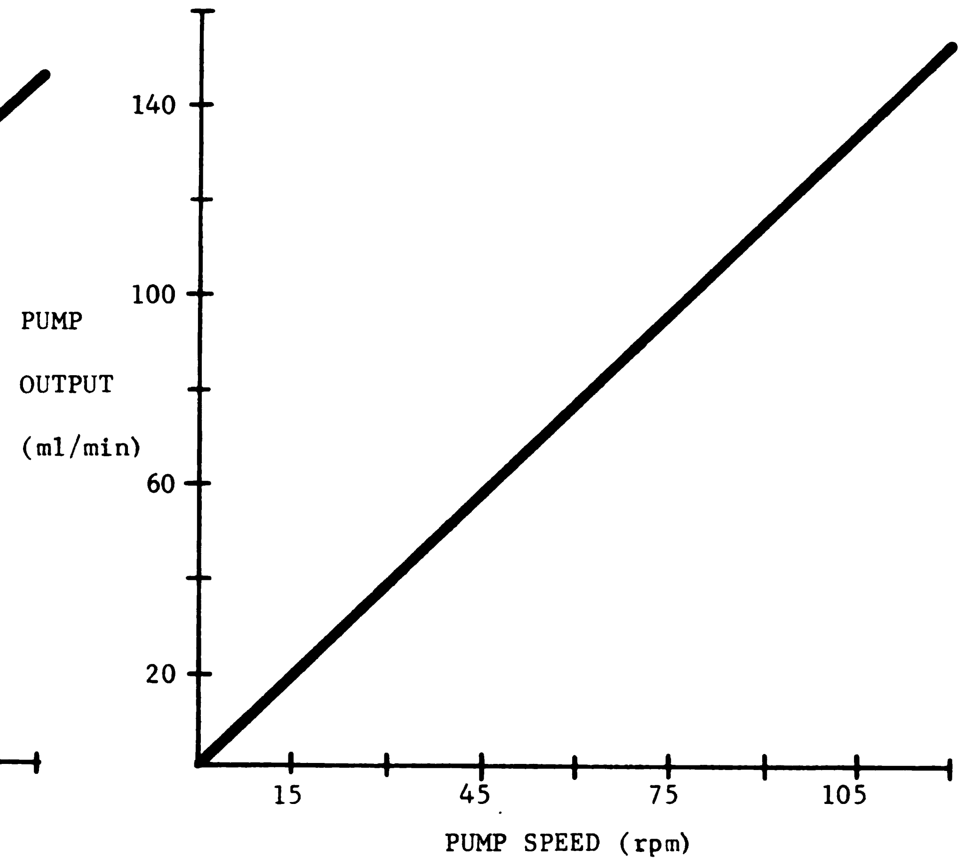


Figure F.2. Output versus Pump Speed--Dioctyl Pthalate (DOP)

APPENDIX F (CONT.)

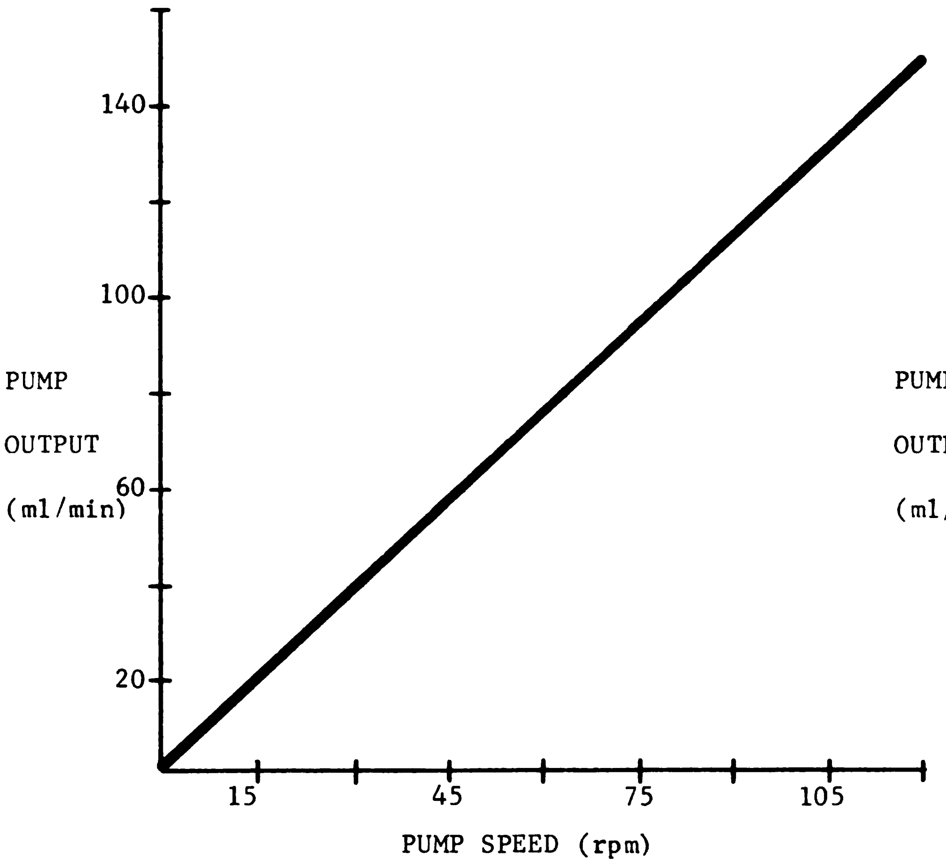


Figure F.3. Output versus Pump Speed--Omite 6E.

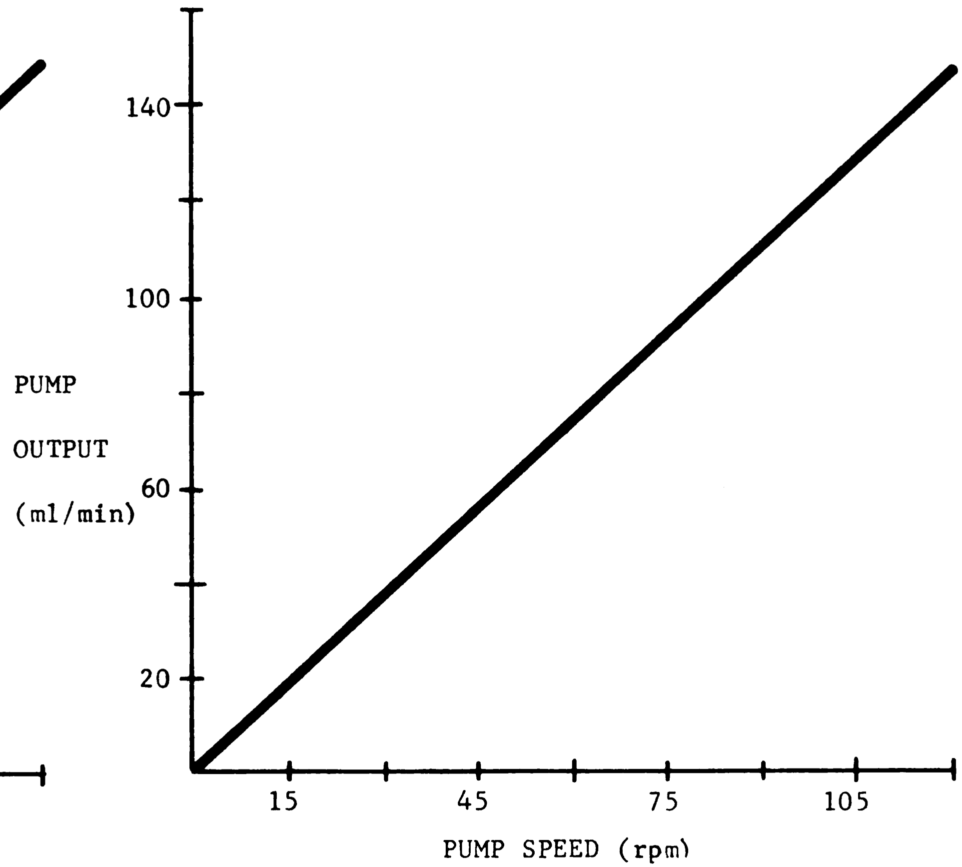


Figure F.4. Output versus Pump Speed--Sevin 4LS.