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Stirling Engines. and Irrigation Pumping

C. D. West

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Engineering Technology Division

STIRLING ENGINES AND IRRIGATION PUMPING

C. D. West

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CONTENTS

LIST OF SYMBOLS

D	diameter of fluidyne displacer tube
f	frequency of operation
g	acceleration due to gravity
H	pumping head or lift
H_{max}	maximum possible pumping head or lift from a single stage
LD	mean length of displacer liquid column
M^4	product of pumped volume and head (Note: Units are $m^3/h \times m$. Elsewhere in this report, SI units only are used)
P_m	mean pressure of working fluid
t	thickness of thermal insulation
T_H	heater temperature (must be in degrees absolute for use in equations)
TK	Cooler temperature (must be in degrees absolute for use in equations)
V_e	displacer swept volume
V_m	mean volume of'working fluid
V_o	peak-to-peak volume change
$\dot{V}_P$	volume of liquid pumped per unit time
W_n	the West number (-0.25)
W_o	power output
ρ	density of pumped liquid
AP	peak-to-peak pressure change in working fluid
ΔP_d	peak-to-peak pressure change in working fluid due to displacer action
ΔP_t	peak-to-peak pressure change in working fluid due to volume change (tuning-line motion)

STIRLING ENGINES AND IRRIGATION PUMPING

C. D. West

ABSTRACT

This report was prepared in support of the Renewable Energy Applications and Training Project that is sponsored by the U.S. Agency for International Development for which ORNL provides technical assistance. It briefly outlines the performance that might be achievable from various kinds of Stirling-engine-driven irrigation pumps. Some emphasis is placed on the very simple liquid-piston engines that have been the subject of research in recent years and are suitable for manufacture in less well-developed countries.

In addition to the results quoted here (possible limits on M^4 and pumping head for different-size engines and various operating conditions), the method of calculation is described in sufficient detail for engineers to apply the techniques to other Stirling engine designs for comparison.

---

1. STIRLING ENGINE POWER OUTPUT

If well-designed and constructed, conventional Stirling engines (see Ref. 1 for a guide to Stirling technology) have a rather simple relation between the brake power, the piston stroke of the machine, the pressure of the working fluid, the frequency of operation, and the temperature of the heater and cooler:

$$W_o \approx W_n P_m f V_o \frac{T_H - T_K}{T_H + T_K} . \qquad (1)$$

A survey of 23 very different engines[1] indicated the average value of W_n to be 0.25, and this is the number usually employed in making rough calculations or predictions of the performance of new Stirling engines. The W_n of 0.25 is only applicable if a consistent unit set (such as SI units) is used in Eq. (1).

If the engine is used to drive a pump, then the power may be used to raise liquid (often water) against gravity:

$$\dot{V}_p H \rho g \approx W_0 \approx 0.25 P_m f V_o \frac{T_H - T_K}{T_H + T_K} . \qquad (2)$$

This relation does not take into account the loss of power in the pump itself. Actually, such an omission is quite legitimate because some of the engines surveyed to assign a value to W_n were pumping engines. Their measured power output referred to the actual pumped volume and head; that is, the pump efficiency (or inefficiency) has to some extent already been included in W_n.

2. STIRLING ENGINE M^4

For convenience, we can rearrange Eq. (2) to calculate $\dot{V}_p H$, which is the quantity sometimes called M^4. However, we also need to convert the pumping rate from m^3/s (i.e., SI units) to m^3/h (the units normally used, unfortunately, for M^4). With these changes,

$$M^4 = \dot{V}_p H = 3600 \times 0.25 \frac{P_m}{\rho g} \times fV_o \frac{T_H - T_K}{T_H + T_K} . \tag{3}$$

Now, $\rho = 10^3$ kg/m^3 (for water), $g = 9.81$ m/s^2, and substituting these numbers into Eq. (3) yields

$$M^4 \approx 0.092 \; P_m f V_o \frac{T_H - T_K}{T_H + T_K} . \tag{4}$$

Most Stirling engines operate at rather high pressure and high speed to maximize the specific power, but others, including the liquid-piston machines (fluidynes) to be described later, are inherently low-pressure, low-speed engines. Table 1 lists the relevant data for five different engines and the value of M^4 calculated from Eq. (4).

Table 1. M^4 calculated for various Stirling engines

Engine[a]	P_m (MPa)	f (Hz)	V_o (mL)	T_H (°C)	T_K (°C)	M^4 (m^4/h)
C-60	0.2	40	60	650[b]	77[b]	20
Fluidyne pump	0.1'	0.63	32,000[b]	375	50[b]	60[c]
102-C	1.2	27	67	900	15	120
GPU-3	6.8	25	'120	780	20	1,100
V-160	13.0	30	226	720	50	4,100

"Brief descriptions of these engines can be found in Tables 5.1, 5.2, and 6.2 of Ref. 1.

[b]Estimated.

'This machine was actually operated as a pump (see Table 9.2, Ref. 1) with a measured M^4 of very nearly 60, thus lending credibility to these calculations.

3. FLUIDYNE LIQUID-PISTON STIRLING ENGINES

Fluidyne is the name given to a class of Stirling engines in which the pistons are actually columns of liquid (usually water) moving up and down in a set of U-tubes. The appendix, a reprint of a 1984 conference paper, describes the principles and practice of fluidynes, together with a brief history.

Because the working parts of the fluidyne are water and the power output is available in the form of either pulsating pressure or movement of liquid in a tube, the most obvious application is as a pumping engine, for example, in irrigation systems. However, only experimental and demonstration machines have been built so far. The largest one had a throughput of more than 15 m^3/h and a lift of almost 4 m. For more information on the technology of the fluidyne , see Ref. 2.

In subsequent sections of this report, some estimates of the possible performance of fluidyne pumping systems are made.

3.1 Power Output of a U-Tube Fluidyne

Suppose the displacer is water in a U-tube of diameter D with a stroke of D — that is, a "square" engine. The uprights are separated by a thickness t of thermal insulation. To avoid excessive mixing and heat losses in the displacer liquid as it turns into the curved section of the U-tube, the liquid surface at bottom dead center of the piston movement is at a minimum height D above the curved portion of the U-tube (see Fig, 1). We assume, reasonably, that the phase angle between the liquid

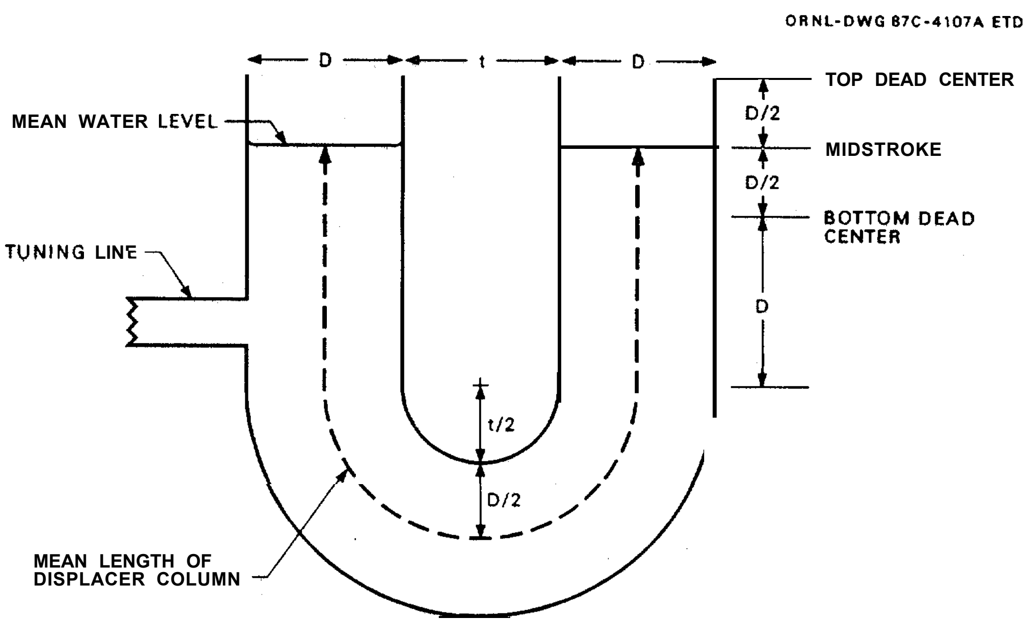

Fig. 1. Geometry of displacer U-tube.

motion in the two arms of the U-tube is 90°. Then the net volume change, which is the vector sum of the volume change at each end of the displacer column, is

$$V_o = \sqrt{2} \times \frac{\pi D^3}{4} . \tag{5}$$

The operating frequency of a fluidyne is determined almost entirely by the length of the displacer column[3]

$$f = \frac{1}{2\pi} \sqrt{\frac{2g}{L_D}} . \tag{6}$$

From the geometry of the displacer (Fig. 1), the length of the centerline along the column is calculated as

$$L_D = 3D + \pi(D/2 + t/2) . \tag{7}$$

Therefore,

$$1 = \frac{1}{2\pi} \sqrt{\frac{2g}{3D + \pi(D/2 + t/2)}} . \tag{8}$$

We can now calculate the approximate power output of the fluidyne from Eqs. (1), (5), and (8):

$$W_o \approx 0.25\ P_m f V_o \frac{T_H - T_K}{T_H + T_K}$$

$$\approx 0.25 \times P_m \times \frac{1}{2\pi} \sqrt{\frac{2g}{3D + \pi(D/2 + t/2)}} \times \frac{\sqrt{2}\pi D^3}{4} \times \frac{T_H - T_K}{T_H + T_K} . \tag{9}$$

It is not convenient to operate this kind of fluidyne at a mean pressure above atmospheric (0.1 MPa) because a higher pressure would expel the liquid from-the tuning line. Substituting $P_m = 10^5$ Pa and g = 9.81 m/s* into Eq. (9) yields

$$W_o \approx \frac{19{,}600\ D^3}{\sqrt{3D + \pi(D/2 + t/2)}} \times \frac{T_H - T_K}{T_H + T_K} . \tag{10}$$

Because of the large surface area of a high-power, atmospheric-pressure engine, good thermal insulation is essential to high efficiency. The insulation thickness should probably be at least equal to the U-tube diameter, that is, $t = D$. Then Eq. (10) becomes

$$W_o \approx 7900\ D^{5/2} \times \frac{T_H - T_K}{T_H + T_K} . \qquad (11)$$

Notice that Eq. (8) is not very sensitive to the exact value of t, so that choice of an insulation thickness somewhat greater or less than the value assumed here would not significantly affect the conclusions.

Equation (11) could be applied approximately to a concentric-cylinder fluidyne by defining D as the diameter of the innermost concentric cylinder.

3.2 Pumping Water with a U-Tube Fluidyne

A quantity of interest to many designers and sponsors of irrigation systems is the so-called M^4 of a system:

$$M^4 = \text{the volume of water pumped} \times \text{the lift} = \dot{V}_p \times H . \qquad (12)$$

For irrigation purposes, a mixed unit set of metres and hours is usually adopted. Now,

$$\dot{V}_p H\rho g = \text{energy added to the water/unit time.}$$

In SI units, for a U-tube fluidyne, Eqs. (11) and (12) indicate that

$$\dot{V}_p H\rho g = W_o = 7900\ D^{5/2}\ \frac{T_H - T_K}{T_H + T_K} . \qquad (13)$$

Changing the time units from seconds to hours gives

$$\dot{V}_p H\rho g = 3600 \times 7900 \times D^{5/2} \times \frac{T_H - T_K}{T_H + T_K} ; \qquad (14)$$

$$M^4 = \dot{V}_p H = \frac{3600 \times 7900}{\rho g} \times D^{5/2} \times \frac{T_H - T_K}{T_H + T_K}$$

$$= \frac{3600 \times 7900}{10^3 \times 9.81} \times D^{5/2} \times \frac{T_H - T_K}{T_H + T_K}$$

$$\approx 2900\ D^{5/2}\ \frac{T_H - T_K}{T_H + T_K} . \qquad (15)$$

Assume $T_K = 30°C$ (86°F). Then M^4 can be calculated as a function of the displacer diameter and the hot-end temperature. A heater temperature of 350°C is practical with the simplest of materials and joining techniques (including adhesives). With more care paid to the choice of materials and construction methods, but with no exotic technology, 550°C is easily reached with safety.

Table 2 and Fig. 2 illustrate the rapid increase of M^4 as the displacer diameter is increased.

Table 2. M^4 for various displacer diameters and heater temperatures

D (mm)	M^4 (m^4/h)	
	T_H = 350°C	T_H = 550°C
100	3.2	4.2
150	8.7.	11.7
200	17.9	24.0
250	31.3	41.9
300	49.4	66.0
350	72.6	97.1
400	101.4	135.5
450	136.1	181.9

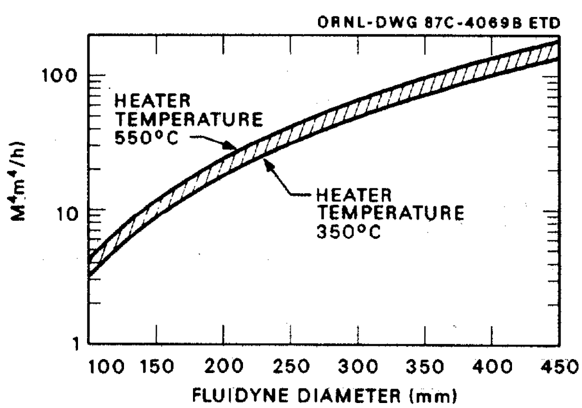

Fig. 2. Fluidyne pumps: M^4 as function of displacer diameter and heater temperature.

3.3 Pumping Head Available from a U-Tube Fluidyne

Staging the pumps by driving several pump arms from the same displacer[2] can lift water to any desired height. However, each stage can lift the water no more than a certain maximum incremental height because the maximum pressure available to drive the pump in an atmospheric-pressure engine is quite limited.

An approximate calculation of the maximum head available is fairly straightforward. When pumping at close to the maximum available head, the volume pumped will be low with little change of volume in the engine during the pumping stroke. Therefore, the main sources of pressure variations in the working fluid of the engine will be the displacer action and the tuning-line action. According to **Eq.** (2.3) of Ref. 1, the **peak-to-**peak pressure change due to displacer action is given by

$$\Delta P_d \approx P_m \frac{V_e}{V_m} \frac{2(T_H - T_K)}{(T_H + T_K)} \; . \tag{16}$$

The mean volume of working fluid V_m in the engine includes the heat exchangers and connecting ducts, as well as the volume in the displacer V_e. Typically in modern designs, the heat exchanger and duct volume (i.e., the unswept or dead volume) is such that $V_m \approx 2.5\ V_e$; see Table 2.1 of Ref. 1.

The peak-to-peak pressure variation due to the volume change V_o is simply given by the ratio of the volume change to the mean volume:

$$\Delta P \approx P \frac{o}{-} \; . \tag{17}$$

As seen earlier, $V_o \approx \sqrt{2}\ V_e$, and therefore

$$\Delta P \approx \sqrt{2}\ P \frac{e}{-} \; . \tag{18}$$

The two arms of the displacer were assumed to be moving with equal strokes and a 90" phase difference (which is why $V_o = \sqrt{2}\ V_e$), and the phase angle between the displacer action and the tuning-line or **volume-**changing action is therefore **45°.**

The pressure change due to the displacer action is in phase with the displacer movement, but the pressure variations due to the volume change are 180° out-of-phase with the change (i.e., when the volume is minimum, the pressure is maximum and vice-versa). Therefore, the phase angle between ΔP_d and ΔP_t is $180° - 45" = 135°$.

The total pressure variation AP is the sum, taking account of the phase angle between them, of ΔP_d and ΔP_t:

$$\Delta P = \sqrt{\Delta P_d^2 + \Delta P_t^2 - 2\Delta P_d\ \Delta P_t \cos 135°} \; . \tag{19}$$

Substitute for ΔP_d and ΔP_t from Eqs. (16) and (18) recall that

$V_m \approx 2.5\ V_e$, and simplify:

$$AP \approx \frac{2P_m}{2.5}\left[\left(\frac{T_H - T_K}{T_H + T_K}\right)^2 \frac{1}{2} + \left(\frac{T_H - T_K}{T_H + T_K}\right)\right]^{1/2} . \tag{20}$$

Only if the peak-to-peak pressure variation exceeds the hydrostatic pressure from the pumping head can the valves open and the water begin to flow. Therefore, the maximum possible lift (excluding any possible enhancement by dynamic effects) in a single stage is H_{max} where

$$H_{max}\ \rho g = \Delta P$$

or

$$H_{max} = \frac{2P_m}{2.5\ \rho g}\left[\left(\frac{T_H - T_K}{T_H + T_K}\right)^2 + \left(\frac{T_H - T_K}{T_H + T_K}\right) + \frac{1}{2}\right]^{1/2} . \tag{21}$$

For an atmospheric engine and water pump, $P_m = 10^5$ Pa, $\rho = 10^3\ kg/m^3$, and $g = 9.81\ m/s^2$. Using Eq. (21), the maximum possible lift for a single-stage fluidyne with a hot-end temperature of 350°C is 8.0 m. However, remember that the pumping rate falls toward zero as the maximum possible lift is approached, so that the practical limit is much lower than the theoretical one.

With a hot-end temperature of 550°C, the theoretical maximum head in a single stage is 8.8 m.

As an example, Fig. 3 combines the results of Eqs. (15) and (21) to show the potential performance limits of a 300-mm (12-in.) displacer-diameter fluidyne operating at 350 and 550°C. The results are shown for a single-stage and a two-stage pumping system.

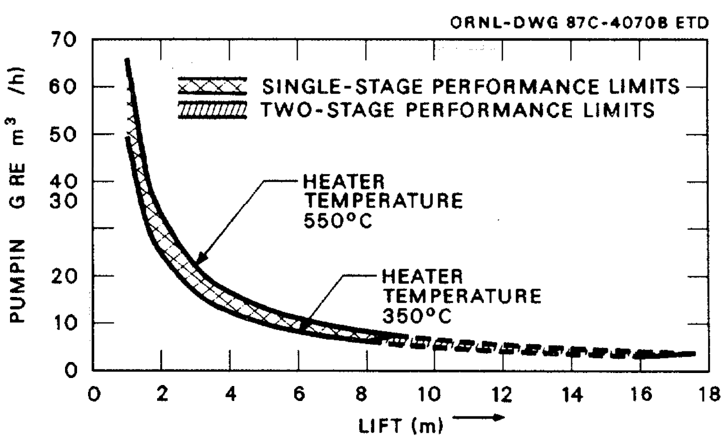

Fig. 3. Maximum pumping rate vs lift for 300-mm bore fluidyne.

4. SUMMARY

An existing correlation for the performance of Stirling engines can be used to make estimates of M^4 for Stirling-powered irrigation pumps. Five engines, representing a wide range of different designs, have been evaluated on the basis of the correlation; the M^4 values range from 20 m^4/h for the smallest engine surveyed to 4000 m^4/h for the largest.

One particular type of Stirling machine, the fluidyne liquid-piston pump, was examined in greater detail. Depending on the size of the engine, M^4 values in the range of 10 to 200 seem to be practical. The maximum theoretical pumping head is about 8 or 9 m (depending upon the operating temperature) for a single-stage design, or 16 m for a two-stage system. The flow rate falls rapidly as the maximum head is approached.

REFERENCES

1. C. D. West, ***Principles and Applications of Stirling Engines***, Van Nostrand Reinhold, New York, 1986.

2. C. D. West, *Liquid* ***Piston Stirling Engines***, Van Nostrand Reinhold, New York, 1983.

3. C. D. West, "Dynamic Analysis of the Fluidyne," Paper 839126, ***Proc.*** *18th IECEC,* August 1983.

Appendix

LIQUID'PISTON STIRLING MACHINES

1—10

2 ND INTERNATIONAL CONFERENCE ON STIRLING ENGINES

JUNE 21-24, 1984

SHANGHAI,, CHINA

LIQUID-PISTON STIRLING MACHINES

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The Chinese Society of Naval Architecture
and Marine Engineering
and
Chinese Society of Engineering Thermophysics

LIQUID-PISTON STIRLING MACHINES

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ABSTRACT

Since the Invention of the Fluidyne engine in 1969, several research groups have explored and described the potential of liquid-piston Stirling machine designs for a wide variety of applications, Including water pumping from solar beat, simple and long-lived fossil-fuel-fired irrigation pumps, and heat-powered heat pumps. A substantial amount of theoretical work has been published, along with experimental results from a number of very different machines and design data for the construction of experimental engines. This paper describes the progress that has been made and the performance of existing systems, identifies outstanding research needs, and outlines some of the potential for further progress.

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BACKGROUND

The Fluidyne liquid-piston Stirling angina was invented at the Harwell Laboratory of the United kingdom Atomic Energy Authority in 1969, snd the first machines were operated there In 1970. Two internal reports wsre written by the Inventor [1,2], and a British pstent was filed [3] coveting the basic invention and some improvements. The Patent Specification was published in 1973, describing the basic theory and some early experimental results.

In 1974, a psper describing the Fluidyne concept was presented to a meeting of the International Solar Energy Society [4] , and the' attractive simplicity of the liquid-piston mschine with its potential for low-cost reliable water pumping was recognized; at that time, tha Metal Box group of companies hsd already expressed an interest in collaborating with Harwell 'on the development *of* Fluidyne irrigation pumps. In 1975, tharefore, several small-scale research efforts were under way; over the next few years, research results confirmed the potential of the liquid-piston engine, while identifying some problems that lie in the way and offering possible solutions. Some of the major results of this research are reviewed in this paper.

PRINCIPLES

One of the simplest versions of the Fluidyne to construct and operate is the liquid-feedback mschine shown diagramatically in Fig. 1. Oscillation of liquid in the displacer U-tube unaccompanied by any movement of liquid in the tuning or output column U-tube would represent, In terms of a conventional Stirling engine, pure displacement; gas would be displaced between the hot end cold spaces but with no net change in gas volume. Movement of liquid in the tuning line &es result in a net change of gas volume, as &es the power piston of a conventional Stirling machine.

.For the mschine to function as an engine, the phasing must be such that the liquid level in the open end of the tuning line is falling during the time that the liquid level in the hot side of the U-tube is higher than that in the cold side : this compresses the gas when most of it Is in the cold space. Conversely, when the liquid level in the hot side is lower than in the cold, the level in the open end of the tuning line must be rising, thus expanding the gas. It is apparent that with such a phasing, the Fluidyne will operate as a Stirling-like engine in the alpha or Rider configuration.

For a Fluidyne of the type shown in Fig. 1, the positioning of the junction between the tuning line and the displacer U-tube is crucial. Usually the junction must be closer to the hot then to the cold end of the displacer for successful operation as an engine, although exceptions to this rule have been noted [5]. With the arrangement shown in Fig. 1, the two sections of the displacer-liquid column (between the junction and each free surf ace) are subjected to the same pressure difference, if the pressure drop in the gas flow across the regenerator/connecting tube is neglected. However, the liquid mass between the junction and the hot surface is less, because of its shorter length, than that between the junction and the cold surface. As a

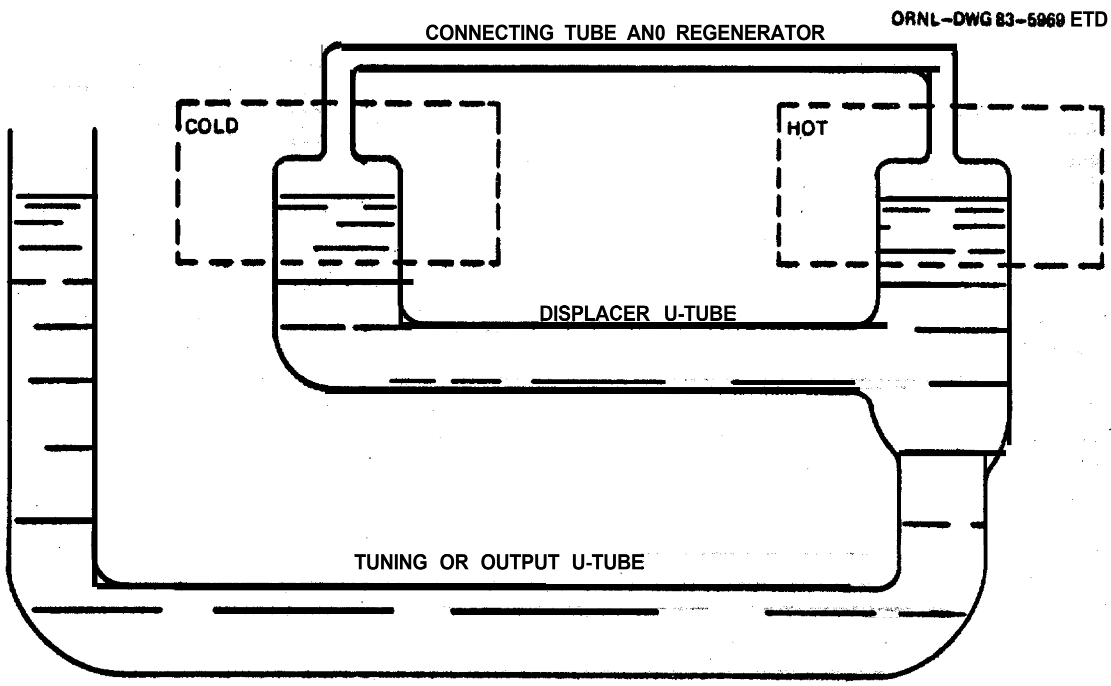

Fig. 1. Liquid-feedback Fluidyne

consequence, the hot side of the displacer-liquid column responds to the pressure difference more readily than the cold side, and its movement will be more advanced in phase. This is exactly the relationship needed for a Stirling engine, in which the hot expansion-space volume variation must lead the phase of the cold compression-epace volume.

If the tuning line is driven externally, there will still, according to the above argument, be a positive phase difference between the resulting motions of the liquid surface In the short and long legs of the displacer U-tube. Consequently, heat will still be moved between the two cylinder volumes, and one will have a Liquid-piston refrigerator or heat pump.

The description given above of the liquid-feedback mechanism is greatly oversimplified. The liquid-feedback system was first proposed, on the basis of intuition, by Cooke-Yarborough; not until 1974 did theoretical explanation for Its operation become available when Elrod saw a description of the liquid-feedback Fluidyne and devised an elegantly simplif led analysis *of* its principles [6]. Elrod 's analysis vindicated Cooke-Yarborough's intuition. The theory has been subsequently extended to take account of loss and loading effects [7,8]. Several computer analyses have also been carried out and are included in the bibliography.

Other feedback systems have also been used or proposed, including systems in which the displacer is given, by one of several possible means, a rocking motion to maintain the amplitude of oscillation of the displacer liquid. Different configurations for the liquid columns have also been used, including a multicylinder arrangement of the Siemens type and a concentric machine in which one leg of the displacer U-tube forms an annulus around the other. Further details of these and other variants are given in Ref. [9].

Although the Fluidyne is essentialby a Stirling engine (or, at, least, has the same kind of volume variations), the use of liquid pistons gives an added freedom of design beyond that available with more conventional Stirling machines. The potential advantage is obvious in cost, simplicity, and maintenance requirements offered by pistons that always fit their cylinder's exactly, regardless of wear or manufacturing tolerances.

It must be recognized, however, that the liquid pistons create or exaggerate effect8 that are absent 'or negligible in solid-piston engines. These include *the* effect (generally undesirable) of oscillating flow on viscous losses and on thermal leakage; the relative ease with which a desirable isothermalization of the cold cylinder can be introduced; the possibility (desirable or otherwise) of substantial evaporation in the hot cylinder; the undesirable limitation on stroke and frequency imposed by gravity-controlled oscillation and by the Rayleigh-Taylor instability of the surface; and the need to keep a more or less constant orientation of the engine so that gravity can hold the liquid in place. The major factors will be discussed in the next few sections.

Oscillating Flow Effects

The liquid flow rate in the displacer and tuning lines is not constant or even unidirectional; it varies approximately sinusoidally with two reversals in each cycle. For channels or tubes of the size normally used in a Fluidyne, the behavior of the liquid, and especially the viscous drag, is greatly off ected by this oscillation. In effect, the boundary layer never has time to develop fully before the flow reverses* Even in laminar flow conditions, the effect of the oscillatory nature of *the* flow is to confine the velocity gradient in the liquid to a narrow region close to the wall [10]. The thickness of this boundary layer is of the order $\sqrt{2\nu/\omega}$, where ν is the kinematic viscosity of the liquid and ω the angular, frequency of the oscillation. For room temperature water oscillating at 1 Hz, thie thickness is of the order of 1 mm.

There are two major consequences. First, the confinement of the velocity gradient to a *narrow* region means that within that region, the gradient will be higher than it would be if the shear stress were spread somewhat uniformly across the full width of the tube, as it is in normal Poiseuille flow. Therefore, the viscous flow losses are increased, On the other hand, the characteristic length for the Reynolds number of the oscillating flow is not the actual diameter of the tube, but the thickness of the boundary layer within which the shear effects are concentrated: this raises the flow velocity at which the onset of turbulence may be expected [9]. Although the effect of turbulence on viscous flow losses in oscillating flow is not established, It is possible that once turbulence has set in, the distinction between oscillating and unidirectional flow may be much reduced.

Although the influence of the oscillations on the kinetic *or* "minor pipe" losses is likewise unknown, it is reasonable to assume (in the absence of. published experimental evidence to the contrary) that the correlations for. unidirectional turbulent flow may be used. These correlations are established for conditions (turbulent flow) in which the time-averaged velocity profile in a straight tube section also shows a characteristic narrow boundary layer near the wall and a fairly uniform velocity over the rest of the tube diameter.

Second, the oscillatory motion of the liquid also enhances, of ten by a very large factor [9], the conduction of heat along the liquid column. The magnitude *of* this undesirable effect depends, inter alla, on the ratio of thermal conductivity to the square root of kinematic viscosity. This 'ratio is rather large for water (almost 1000, in SI units, at WC, compared with only 10 for a typical oil), which is unfortunate because water is of ten the liquid of choice for a Fluidyne. The effect may be a major source of heat loss from the hot cylinder; although it can be greatly reduced by use of an insulating float on the water surface, such a float (if solid) makes it impractical to introduce extended surface area devices (such as fins or tubes) into the cylinder to isothermalize the gas behavior.

<u>Isothermalization and Transient Heat Transfer Losses</u>

In an ideal Stirling cycle, all processes are isothermal. In practice, the gas in the cylinders, and sometimes in the connecting ducts as well, does not have time to exchange much heat with the walls during the course of a single cycle — that is, it behaves almost adiabatically. Consequently, the expansion and compression processes tend to lower and raise the gas temperature in these spaces (except in the regions very close to the wall, where there is a thermal boundary layer). Three major efficiency-loss effects can be attributed to this cause, although it should be realized that there are important interactions among the three.

First, heat flows across the" temperature difference between the gas and the wall, which is an irreversible process. Although the net heat flow over a cycle will be zero, once the equilibrium condition has 'been reached, heat is lost from the gas to the wall during the part of the cycle when the gas temperature is above the wall temperature and is returned to the gas &ring the part of the cycle when its temperature is lower. Giving up heat at a high temperature and regaining it at a lower temperature is obviously an inef- f icient process. This loss mechanism is often called the transient heat-transfer loss, the hysteresis loss (because of its Importance in gas springs), or the cyclic heat-transfer loss. It was first recognized as a very important effect in the Fluidyne by Ryden, whose report [11] was not however made public until 1983. For a truly adiabatic cylinder, the transient bat-transfer loss will be zero because, by definition, there is no heat exchange between the gas and the wall in an adiabatic space. On the other hand, neither will an ideally isothermal cylinder suffer from transient heat-transfer losses because there is, by def inftion, no temperature difference between the gas and the wall in such a cylinder. It follows that there is a worst case somewhere in between the zero heat transfer ooeff icient and infinite heat transfer coef - f icient cases. In fact, Lee has shown [12] that the worst case will occur when the thermal boundary layer thickness is about equal to the hydraulic radius of the cylinder. The thermal boundary layer thickness is given by $\sqrt{2\alpha/\omega}$, where a is the thermal diffusivity and ω the angular frequency of the oscillations. For 1-Hz oscillations in air at room temperature and pressure, the thickness calculated from the thermal diffusivity based on' pure conduction is $\sim$3 mm. Notice that the ratio of the flow boundary layer thickness to the thermal boundary layer thickness is u/a, the Prandtl number. For gases, the Prandtl number is of the order of 1, so that gas spaces large enough to show marked oscillating flow effects will tend to behave nearly adiabatically; nearly isothermal spaces, on the other hand, will not show very marked oscillating gas flow effects. In practice, the thermal diffusivity will be enhanced by convection and turbulence that will modif y , and of ten increase, the transient heat-transfer loss.

By placing fins, tubes, or other area enhancements in the Fluidyne cylinder , provided the spacing between them is less than a few millimeters, the gas behavior can be brought into the nearly isothermal regime. The transient heat-transfer loss may still be considerable; although the temperature fluctuations and therefore the heat transferred per unit area will be greatly reduced, this is offset by the increase in area. Nevertheless, as we shall see, the reduced temperature fluctuations favorably affect some other losses, and near-isothermalization is generally worthwhile if it can be achieved without an excessive increase in complexity or in flow losses. It is obviously easier with a liquid piston, where the liquid will invest the spaces between the fins regardless of tolerances, than with a solid piston, where close matching of moving and stationary fins would be needed.

The second major effect of the pressure-induced fluctuations in the cylinder gas temperature is that the mean gas temperature over the cycle is no longer equal to the temperature of the adjacent heat exchanger. During the compression phase when the gas temperature will be raised by adiabatic compression, most of the gas is In the cold compression cylinder. During the expansion phase when the gas temperature is falling, there is relatively little gas in the compression cylinder. Therefore, the mean temperature of the gas in the compression cylinder reflects the compressive heating more than the expansive cooling, and so the mean gas temperature in the compression cylinder is higher than that in the cooler. Similarly, the mean gas temperature in the expansion space is lower than the heater temperature. As a result, the effective temperature difference between the expansion and compression phases of the cycle is less than the difference between the heater and cooler temperatures, so that efficiency calculations based on gas temperatures I a the . . . exchangers will be overestimates. Clearly, reducing the amplitude of the temperature fluctuations by partially isothermalizing the cylinders will usually reduce the off iciency loss due to this shift of mean temperature, although it may not decrease the transient beat transfer loss unless the isothermalization is very effective. The use 'of liquid pistons makes it easier to install attended surface area into the cylinders for this partial isothermalization.

The third affect, which can also be reduced by even partial bothermalization, arises from the instantaneous temperature difference between the heater (or cooler) and the gas in the adjacent cylinder. With ideal components, gas leaving the heater and entering the expansion space till do so at a constant temperature — the heater wall temperature. The gas already within the cylinder will be at a different and usually lower temperature due to the pressure-induced temperature variations. There will therefore be a mixing of gas at two different temperatures — an kreveroible process. Similarly, during the part of the cycle when gas is leaving the expansion space and entering the beater, it will generally do so at 8 temperature that is dif- f erent from and, for most of the cycle, lower than the heater. Once again, heat is transmitted irreversibly across a finite temperature difference, leading to a loes of eff fciency .

Effect of Evaporation on Engine Performance

Unless the liquid in the expansion cylinder has a low vapor pressure at the operating temperature or is separated from the hot gas by Some kind of insulating float, enough evaporation will take place from the liquid surf ace and wetted cylinder walls to modify substantially the composition and pressure of the working gas. Because the evaporation will naturally tend to take place mainly during the downstroke of the expansion piston when hot gas is entering the cylinder, it will raise the pressure during the expansion phase and thereby increase the indicated power. Calculation8 have indicated that a several-fold increase may be attainable, and this is presumably the reason that small Fluidynes (which suf fer from the large transient heat-transfer looses associated with the Increased surface-to-volume ratio of small cylinders and the large flow losses associated with small-diameter tubes) will not work In the absence of evaporatloa [13].

Much greater beat input is needed for the evaporative ("wet") cycle than for one in which evaporation is suppressed ("dry" Fluidyne). A conventional regenerator cannot recover such of this extra heat because the saturation temperature of the fluid is higher during the high pressure of the compression phase (when much of the hear stored in the regenerator should be returned to the working fluid) than &ring the expansion phase, when the beat was stored in the regenerator. Without regeneration, an earlier study suggested that the brake efficiency of the evaporative cycle in a Simple Fluidyne may be limited to around 1% or 2% [14]. However, with more careful control of the quantity and timing of the evaporation an Indicated efficiency in the range of 3 to 9% at low temperatures (110 to 130°C) may be achievable even without regeneration [15]; this may open the way to higher brake efficiency, Moreover, a most Ingenious proposal by Renfroe [16] that uses a hydrated-salt atorage medium in the regenerator has opened up the possibility of achieving at least some degree of effective regeneration and hence still higher efficiency in evaporative Fluidynes.

At present, the general conclusions ue that the theoretical eff iciency advantage of the dry engine is likely to be realized in practice only for larger machines (with a cylinder diameter perhaps >50 mm) and that for small engines evaporation is essential for successful operation. Similarly, for operation across a small temperature difference — arch as might be available from simple solar collectors — the greater indicated power output of the evaporative cycle may be needed. If the highest possible efficiency is the goal, then a large dry engine is needed.

As indicated earlier, transient beat transf er is a major source of loss in the Fluidyne engine; if the expansion cylinder is filled with tubes or fins to isothermalize the space, then the liquid piston in that cylinder will be approximately the same temperature as the gas. Evaporation would then be an important effect unless a lower vapor pressure liquid were used in place of water or unless very low temperature6 are used, but as already stated, it is difficult to achieve efficient low-temperature operetion without evaporation.

Although the discussion here has centered around machines in which the working fluid is either a permanent gas (usually air, although lower molecular weight gases have been used with good results) or a gas/vapor mixture, one could construct a Fluidyne type of engine in which permanent gas is excluded from the working space so that the cycle depends entirely on steam or some other vapor [17]. Apparently, no experimental results from such a machine have been published.

Frequency and Stroke Limitations

For the type of machine shown in Fig. 1, the frequency of opeation is determined largely by the natural frequency of oscillation of the liquid in the displacer tube under the restraining force of gravity. This natural frequency is equal to $\sqrt{2g/L}$ rad/s, where g is the acceleration due to gravity and L is the length of the liquid column. Some small demonstration models have been built with a displacer column as short a6 100 mm, corresponding to a frequency of about 2 Hz, but for larger engines the minimum length is limited by the diameter of the displacer tubing and the need to accommodate the stroke in each upright of the U-tube. For an engine with a 150-mm diam displacer and a bore-to-stroke ratio of 1, it would be diff icult to construct a displacer U-tube of much less than l-m length (Fig. 2), corresponding to a frequency of only 0.7 Hz; in fact, both the engines of this size described in the literature [18,19] have a displacer somewhat larger than 1 m and operate at a frequency of ~0.55 Hz.

There are way6 to circumvent these limitations to a certain extent, such as using a displacer tube of nonuniform cross section or using a rocking-beam configuration in which the frequency is determined primarily by the motion of the displacer tube itself rather than the Liquid column within it [9]. A more fundamental approach is to increase the restoring force beyond that provided by gravity alone. In the multicylinder engine shown in Fig. 3, each liquid column 16 subject to the restoring forces arising from the compression or expansion of gas in the adjacent cylinders a6 well as from gravity. Usually, these gas spring forces are much larger than the gravitation81 ones, and so the resonant frequency is much higher than it would be for the same column oscillating under gravity alone.

In a stationary liquid with a free surface, the direction of the acceleration due to gravity is from the dense fluid (liquid) toward the less dense

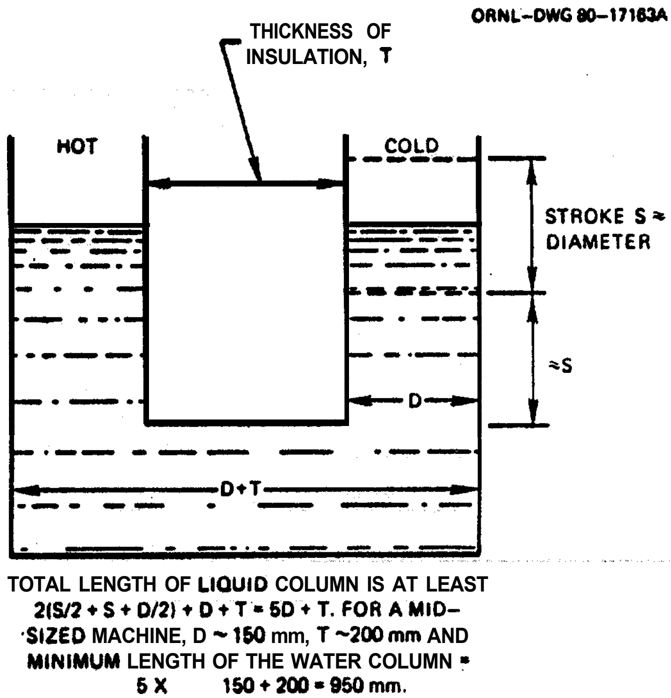

Fig. 2. Minimum length of displacer-liquid column

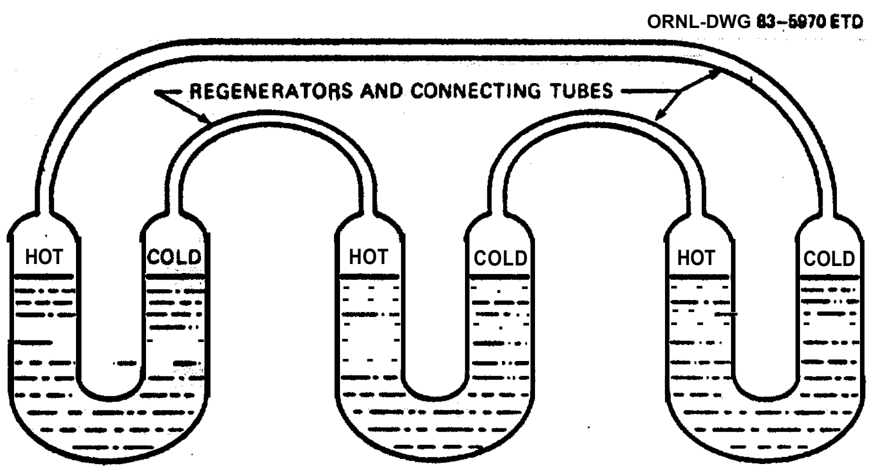

Fig. 3. Double-acting multicylinder Fluidyne engine

fluid (gas), and the interface is stable — as in a glass of water. In the last century, Lord Rayleigh showed that if the acceleration is in the opposite direction (e.g., if the glass is rapidly accelerated downwards, 'or if it is turned upside down), the Interface is unstable and will break up — that is, small departures from a smooth flat surface will tend to grow. According to legend, the theory was reinvented by Taylor in the course of investigations into the problems encountered during WW 11 while trying to obtain a smooth and spherically symmetrical implosion of the core of an atomic bomb. In any case, it is now known as the Rayleigh-Taylor instability. For a sinusoidal movement of the liquid, the peak acceleration of the surface is $s\omega^2/2$, where s is the etroke and ω the angular frequency. If this exceeds the acceleration due to gravity, then the liquid surface will be subject to the Rayleigh-Taylor instability during the downstroke. For a 100-mm stroke, this will be the case for all frequencies higher than 2.2 Hz. This could pose a severe design constraint, especially on a high-frequency multicylinder engine and on the output column of a liquid-feedback engine. (The output column is usually made from narrower tubing than the displacer to minimize Its length, and it has a higher stroke so that the output column will reach the stability limit before the displacer column.) However, even beyond the limits of the Rayleigh-Taylor instability, the surface disturbances have a finite rate of grqwth, and the stability boundary can be crossed without problems for at least a short time. Both the stability limit and the rate of growth are affected by surface tension and other effects that may be particularly Important in narrow tubes; this may explain why Martini's multicylinder beat pump Fluidyne [20] has successfully operated with a frequency/stroke combination exceeding gravitational acceleration. This question has not been very deeply explored, especially not in a quantitative manner, although it is clear that some sort of limit must exist; it is, after all, well known that water will generally fall from a glass held upside down.

PERFORMANCE DATA

Having described some of the interesting theoretical aspects of the Fluidyne engine, one may turn to the measured performance of some of the machines that have been built.

Table 1 lists the characterfetics, in relation to the previous discussion, of several different engines along with the engine performance. This table includes only engines used for pumping. The wet machines operate successfully even in small sizes, but no results have been published showing the performance of a large engine with evaporation. The three larger engines listed in the table all operate on a dry cycle, and although they show a higher efficiency , a higher operating temperature is needed. The effective cylinder temperature of the vet machines is not recorded, but it is presumably no greater than 100°C or so because there is usually no boiling during operation. All these machines had air as the permanent gas fraction of the working fluid.

Little has been published about the largest machine, designed and built by R. Pandey as prt of a research program at the Metal Box Company of India; the program was ultimately aimed at developing coal-fired irrigation pumps. The engine is of concentric cylinder design with the tuning line constructed as a spiral on the base of the displacer (Fig. 4 shows one possible layout of this type). The water pump is driven by the gas pressure variations in the

Table 1. Characteristics of some Fluidyne water pumps

Reference	Displacer diameter (mm)	Frequency (Hz)	Cycle type	Input W	Maximum flow (L/h)	Maximum head (m)	Notes
21	16	~1.0	Wet	10	22	0.3	Electrically heated, no isothermalization, and no hot end float
22	48	0.8	Wet	60	92	1.0	Electrically heated, no isothermalization, and no hot end float
2	44	0.8	wet	530	390	1.6	Electrically heated, no isothermalization, and no hot end float
19	150	0,561	Dry	200	740	1.3	Electrically heated, cold cylinder isothermalized, heater temperature 350°C, and hot end float
18	150	0.554	Dry	300	1,700	3.0	Electrically heated, cold cylinder isothermalized, heater temperature 360°C, and hot end float
23	300[a]	0.625	Dry	4,400	16,000	3.7	Gas fired, cold cylinder isothermalized, heater temperature 375°C, and hot end float

[a] Estimated diameter of inner cylinder in a concentric cylinder engine geometry.

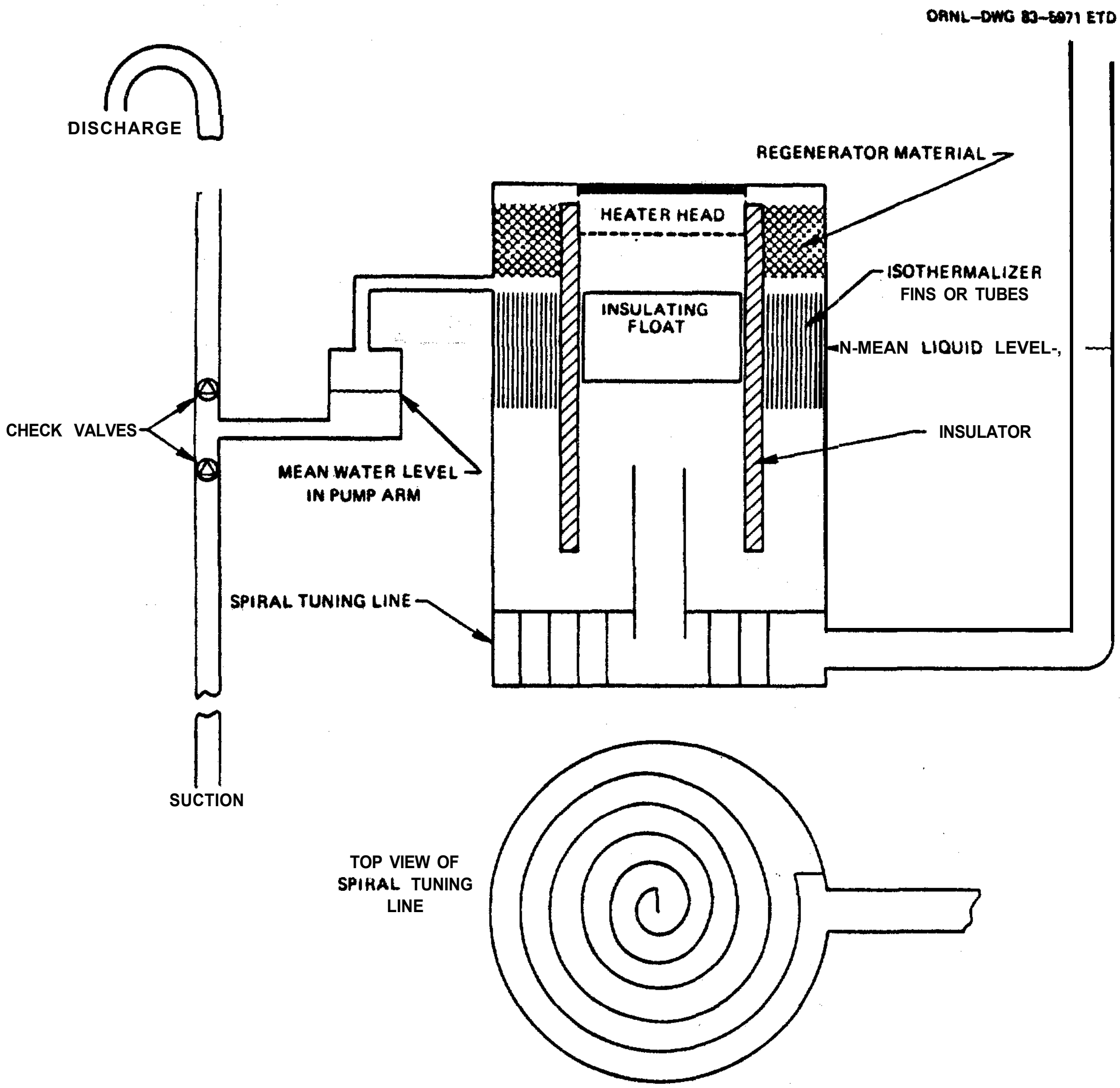

Fig. 4. A concentric cylinder machine with spiral tuning line and gas-coupled pump

working space, a system known as gas-coupling. The Pandey engine operate6 at a mean gas pressure equal to atmospheric and at a fairly low heater temperature (375°C). Qnsequently , the specific output is low, even though the Beale number is 0.008, which is quite respectable for such a low-temperature machine. A large, but simple; engine is therefore needed to generate the pumping power, which for a throughput of 4,400 U.S. gal/h through a head of 12 f t is almost 1/4 hp. No details of the heater bead design are available, and very little is known even about the basic dimensions of the engine. However, it appears that the expansion cylinder, equipped with a float to minimize evaporation, may have a diameter of about 12 in. and the tuning line an effective diameter of 8 in. The operating frequency is 0.625 Hz, Implying a displacer-liquid column 'length of slightly more than 4 ft.

This uthor prepared an outline design, Including all ma jot dimensions, for a pump intended to have a performance (80 gal/min through a head of 10 f t) quite similar to Pandey 's machine. The design was prepared for the Riggs & Stratton Corporation, but it was never built; the company has kindly given their permission for its publication. Pull details are available (24), but briefly the design proposes a U-cube displacer with an Internal diameter of 13-1/2 in. and a tuning line internal diameter of 7-3/4 in. The heater is an annulus placed above the expansion cylinder and concentric with It. The regenerator matrix is 3-mm pitch aluminum honeycomb, and the cold cylinder is filled with 5-mm pitch honeycomb for isothermalization. Evaporation and heat conduction from the expansion cylinder are minimized by means of a short insulation-f illed float on the vater surf ace.

APPLICATIONS

The most obvious application for a liquid-piston engine is as a liquid pump, and most of the Fluidyne research and development has been carried out with this in mind. Many people have thought of the use of solar heat or waste heat with this kind of engine, particularly for Irrigation pumping. Small engines (15 mm-diam or smaller displacer cylinders) have in fact been successfully operated from sunlight focused with an inexpensive plastic Resnel lens of perhaps 300-cm^2 area. In the quest for simplicity and reduced maintenance requirements, various Fluidyne pumps of 10- to 100-mm displacer diameter have been successfully operated with fluidic valves, In 'a system having no solid moving parts (except, for a hot cylinder, float in the 100-mm engine). For irrigation pumping, the simplest type of Fluidyne has a limitation; by choosing a mean working gas pressure equal to atmospheric pressure, construction and operation are much simplified compared with a pressurized engine, but the pumping bead is limited by the available pressure swing to no more than perhaps 15 ft. This Is, of course, adequate for irrigation pumping In many areas of the world, but not in all. There, are ways to increase the pumping head by staging or by pressurizing the working fluid [9], but these methods have so far been little. explored.

Other pumping applications that have been proposed include water circulation in gas-fired hot water central heating systems (to reduce the dependence of gas-fired systems on electricity supplies), drainage pumping, domestic water supply in remote locations without benefit of utility electricity, combustion engine cooling with engine waste heat as the power source, and failsafe cooling of nuclear reactors after shutdown.

The multicylinder configuration can, in principle, be operated as 8 heat-powered heat pump, with some of the expansion spaces being heated externally and generating the power to sat the system into oscillation. The remaining expansion cylinder(s) then act as the input side of a Stirling heat pump [9]. For several years, there was a series of conflicting arguments about this scheme: some argued, on apparently impeccable grounds, that it could work; some argued, on apparently equally impeccable grounds, that it could not. The immediate question was resolved by Martini, who built a small machine that worked [20]. Martini's small proof-of-principle model does not give any guide to the ultimately available performance, cost, or efficiency of these systems, but it does offer reassurance that the efficiency can be greater than zero.

As mted earlier, a Fluidyne driven by external gas preasure pulsations would operate as a refrigerator, but it appears that no calculations, much less experiments, have been carried out on such a system.

Finally, it may be remarked thet the Fluidyne incorporates, in a working heat engine simple enough to be put together in the smallest laboratory or workshop, many of the Important basic lessons of classical physics — including single- and ewe-phase thermodynamics, kinematics, the behavior of tuned oscillators and fluid flow. It is therefore a good teaching and learning example for high school and college students; many successful student projects on the Fluldyne are known to the author, and presumably there are others that have not been communicated.

CONCLUSION

The liquid-piston Fluidyne is a form of Stirling engine sharing many of she characteristics of conventional kinematic and free-piston Stirling machines. The use of liquid pistons, however , gives it some unique advantages as well as certain problems that are not encountered or are not important in engines with solid pistons. Because the output is naturally available in the form of an oscillating liquid flow or a fluctuating pressure, the Fluldyne is well suited to liquid pumping, but other applications have also been considered.

REFERENCES

1. C. D. West, *Hydraulic Heat Engines*, AERE-R 6522, 1970,

2. C. D. West, *The Fluidyne Heat Engine*, AERE-R 6775, 1971.

3. C. D. West, J. C. H. Geisow, and E. H. Cooke-Yarborough, "Improvements In Or Relating To Stirling Cycle Heat Engines," British Patent — 1 329 567, (filed 1970).

4. C. D. West, "The Fluidyne Heat Engine," *Proceedings of International Solar Energy Society Conference at Brighton, England*, 1974.

5. L. F. Goldberg, "A Computer Simulation and Experimental Development of Liquid Piston Stirling Cycle Engines," MSc Thesis, Dept. of Mechanical Engineering, University of the Witwatersrand (1979).

6. H. G. Elrod, *The Fluidyne Heat Engine: How To Build One — How It Works*, ONR London Report R-14-74, NTIS No. AD/A-006-367 (1974).

7. C. W. Stammers, "The Operation of the Fluidyne Heat Engine at Low Differential Temperatures," J. Sound *Vib.*, 63, (4) 507-516 (1979).

8. C. D. West, "Dynamic Analysis of the Fluidyne," Paper No. 839126, *Proceedings of the 18th IECEC*, 1983.

9. C. D. West, *Liquid Piston Stirling Engines*, Van' Nostrand Reinhold, 1983.

10. D. J. Ryden, *Viscous Energy Losses in Oscillating Fluidyne Liquid Columns*, AERE-R 8290 (1976).

11. D. J. Ryden, *Energy Losses Due to Irreversible Gas Process in Fluidyne Pumps*, AERE-R 8291 (1976).

12. K. P. Lee, "A Simplistic Model of Cyclic Heat Transfer Phenomena in Closed Spaces," Paper No. 839116, *Proceedings of the 18th IECEC*, 1983.

13. D. J. Ryden, "Evidence That Fluidyne Pumps Work on a Steam Cycle and not a Stirling Cycle," AERE-M 2779 (1976).

14. C. D. West, "Performance Characteristics of West and Dry Fluidynes," Paper No. 829292, *Proceedings of the 17th IECEC*, 1982.

15. C. D. West, "Stirling Engines with Qntrolled Evaporation of a Two-Phase Tim-Component Working Fluid," Paper No. 839149, *Proceedings of the 18th ZECEC*, 1983.

16. D. A. Renfroe, "Effects of Using Hydrated Salts as Regenerator Material in a Stirling Engine Utilizing a Two-Phase Two-Component Working Fluid," to be published in *Proceedings of the 19th IECEC*, 1984.

17. E. Franklin, "Improvements in or Relating to Pumps," British Patent 20172278, (filed 1979).

18. C. D. West and R. B. Pandey, "A Laboratory Prototype Fluidyne Water Pump," Paper No. 819787, *Proceedings of the 16th IECEC*, 1981.

19. P. M. Storey, "Experimentation on a Fluidyne," University of Calgary, Faculty of Engineering, Project Formal Report (1983).

20. W. R. Martini, "Tests on a 4 U Tube Heat Operated Heat Pump," Paper No. 839142, Proceedings of the 18th IECEC (1983).

21. D. C. Mosby, "The Fluidyne Heat Engine," M.Sc. Thesis, Naval Postgraduate School, Monterey, California (1978).

22. G. T. Reader, M. A. Clarke and D. R. Taylor, "Some Experiences with Fluldyne," Paper No. C20/82, Proceedings of the Institute of Mechanical Engineers', Conference 'Stirling Engines," Reading, England (1982).

23. Private Communication, R. B. Pandey to C. D. West (April 13, 1981).

24. C. D. West, "Re-Prototype Fluidyne Water Pump," Westware Co. report to Briggs & Stratton Corporation, available from Martini Engineering, 2303 Harris, Richland, WA 99352, USA (1981).

BIBLIOGRAPHY

ANALYTICAL STUDY OF LIQUID PISTON BEAT PUMP TECHNOLOGY, ANTI Interim Report to Argonne National Laboratory under contract No. 31-109-38-6506 (1982).

DYNAMICS OF A PISTONLESS ENGINE CAPABLE OF UTILIZING SOLAR ENERGY, S. C. Bhargava et al., *J. Appl. Phys.* 49(6), 3521-3526 (1978).

PASSIVE SOLAR WATER PUMP, G. C. Bell, Independent Project Arch 589, [University of New Mexico] 1979.

SOLAR POWERED LIQUID PISTON STIRLING CYCLE IRRIGATION PUMP, G. C. Bell, [Sandia Laboratory] SAN-1894/l, 1979.

THE FLUIDYNE ENGINE, A. J. Bell and L. F. Goldberg, Final-year laboratory report, [University of the Witwatersrand] 1976.

TEST REPORT OF TWO FLUID PISTON HEAT ENGINES, D. Cutler and C. Hanke, [Chicago Bridge and Iron Co.], CBI Res. contract R-0268, 1979.

AN INITIAL MODEL FOR THE FINITE DISPLACEMENT RESPONSE CHARACTERISTICS OF A FLUIDYNE PUMP, T. M. Drzewiecki, WL-TR-1868, 1979.

A STUDY OF A FLUIDYNE HEAT ENGINE, R. Nndlay and P. Hook, Thesis, Department of Mechanical Engineering, [McGill University] Montreal, Canada, 1977.

THE ONSET OF OSCILLATIONS IN A LOSSLESS FLUIDYNE, D. J. Ryden, AERE-M 2840, 1976.

THE MATHEMATICAL MODELLING OF A SET-STREAM FLUIDYNE, P. F. Gill, RNEC-SERF-F1-80, 1980.

A PROTOTYPE LIQUID-PISTON FREE-DISPLACER STIRLING ENGINE, L. F. Goldberg and C. J. Rallis, Paper No. 799239, *Proceedings of the 14th IECEC*, 1979.

SOME EXPERIMENTAL RESULTS ON LABORATORY MODEL FLUIDYNE ENGINES, L. F. Goldberg et al., Paper No. 779255, *Proceedings of the 12th IECEC*, 1977.

ANALOG SIMULATION OF A FLUIDYNE ENGINE, M. Gosling and D. Boast, BSc Project Report 382 [University of Bath] 1976.

ASSESSMENT OF A HARD-COUPLED FLUIDYNE PUMP, T. W. Hensman, RNEC-SERF-F2-1980, 1980.

THE DESIGN, MANUFACTURE, OPERATION AND PRELIMINARY TESTING OF A LIQUID PISTON, FREE DISPLACER, BACK-TO-BACK, GAMMA TYPE STIRLING ENGINE, S. W. Leigh, P. C. Lohrman, and R. Swedlow, Final-year laboratory project [University of the Witwatersrand] 1979.

OPERATION OF A JET-STREAM FEEDBACK FLUIDYNE, P. D. Lewis, RNEC-TR-78008, 1978.

THE PUMPING CHARACTERISTICS OF A JET-STREAM FEEDBACK FLUIDYNE PUMP, S. J. I-- Lloyd and T. W. Hensman, RNEC-SERG-4, 1979.

EXPERIMENTAL AND COMPUTATIONAL EVALUATIONS OF ISOTHERMALIZED STIRLING ENGINES; W. R. Martini, S. G. Hauser, acid M. W. Martini, Paper No;' 779250, *Proceedings of the 12th IECEC*, 1977.

DESIGN STUDY OF ISOTHERMALIZED STIRLING ENGINES, W. R Martini, Final draft report to Argonne National Laboratory under ANL contract 31-109-38-5304, 1980.

REVIEW OF LIQUID PISTON PUMPS MD THEIR OPERATION WITH SOLAR ENERGY, C. L. Murphy, ASME-79-Sol-4, pp. 2-8, 1979.

THE RUIDYNE - A NEW CLASS OF HEAT ENGINE, G. T. Reader, Paper No. 19, ' Polytechnic Symp. Thermodyn. Heat Transfer, Leicester, 'England, 1979.

MODELLING TM JET-STREAM FLUIDYNE, G. T. Reader et al., Paper No. 819792, *Proceedings of the 16th ZECEC*, 1981.

MODES OF OPERATION OF A JET-STREAM FLUIDYNE, G. T. 'deader and P. D. Lewis, Paper No. 799238, *Proceedings of the 14th ZECEC*, 1979.

THE FLUIDYNE - A WATER IN GLASS HEAT ENGINE, G. T. Reader and P. D. Lewis, J.N.S., 5, (4) 240-245, 1979.

STIRLING ENGINES, G. T. Reader and C. Hooper, Spon, 1983.

INVESTIGATIONS OF A PHYSICAL SYSTEM CAPABLE OF USING SOLAR ENERGY, R. Rup, *J. Physics D - Applied Physics*, 11, (18) L208-L209 (1978).

THE FLUID MECHANICS OF THE JET-STREAM FLUIDYNE, J. R. Singleton, RNEC-SERG-S-79, 1979.

DEVELOPMENT OF THE INSTRUMENTATION TO ENABLE AN EXPERIMENTAL ANALYSIS TO BE MADE OF A JET-STREAM FEEDBACK FLUIDYNE, G. C. Thwaites, RNEC-SERG-5-79, 1979.

PATENTS

C. D. West, E. H. Cooke-Yarborough and J. C. H. Geisow, "Improvements in o r Relating to Stirling Cycle Heat Engines," British Patent I 329 567, f iled 1970 (describes basic Fluidyne concept and numerous different conf igurations; also gives some experimental results).

C. D. West, "Improvement6 in or Relating to Stirling Cycle Heat Engines," British Patent 1 487 332, filed 1974 (describes rocking beam fluidyne-driven by bellows from working gas pressure variations).

C. D. West, *Improvements in or Relating to Stirling Cycle Beat Engines," British Patent 1 507 678, filed 1974 (describes nulticylinder Fluidyne engines and refrigerators; applications also filed in India, Egypt and Kuwait).'

C. D. West and J. C. H. Geisow, "Improvements in or Relating to Stirling Cycle Engines," British Patent 1 568 057, filed 1975 (deecribee isothermalizer techniques, cooling and heating through the liquid medium, use of a float to reduce evaporation in the hot cylinder, and design of an internal pump for cooling; applications also filed in India, Egypt and Kuwait).

C. D. West, J. C. H. Geisow end R. B. Pandey, "Improvements in or Relating to Stirling Cycle Beat Engines," British Patent 1 581 748, filed 1976 (describes the concentric cylinder layout, folded tuning line, multicylinder concentric layout, and application of nulticylinder liquid piston configuration to beat activated beat pumping; applications also filed in India and Kuwait).

C. D. West, J. C. H. Geisow and R. B. Pandey, "Improvements in or Relating to Stirling Cycle Heat Engines," British Patent 1 581 749, filed 1977 (describes means for pressurizing the working fluid in a Ruidyne pump; Applications also filed in India, Egypt, Kuwait, and Nigeria).

J. Gerstmann and Y. Friedman, "Liquid Piston Beat Actuated Beat Pump and Methods of Operating Same," U.S. Patent 4,148,195, filed 1977 (describes multicylinder, liquid-piston heat-actuated heat pump, and methods of switching between summer and winter operation).

E. Franklin, "Improvements in or Relating to Pumps," British Patent 21072278, filed 1979 (describes a Fluidyne in which the working fluid is vapor only, with no permanent gas content).

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Internal Distribution

1. N. C. J. Chen
2. D. M. Eissenberg
3. W. Fulkerson
4. J. A. Getsi
5. F. P. Griffin
6. M. A. Kuliasha
7. F. R. Mynatt
8. H. E. Trammell
9-11. C. D. West
12-13. T. J. Wilbanks
14. S. B. Wright
15. ORNL Patent Office
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