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A NEW TYPE OF PUMP TURBINE SYSTEM SPECIALLY DEVELOPED FOR REVERSE OSMOSIS APPLICATIONS

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ABSTRACT

A new type of pump has been developed specially suitable for reverse osmosis applications.

The hydrodynamic theory of the pumps is given as well as some mechanical calculations typical of these pumps. Finally, the experimental performances of a number of working models arq - shown.

A very short comment is also included on the feasibility of - utilizing a combined turbo-pump group of this type for energy recovery in reverse osmosis desalination plants.

INTRODUCTION

Several attemps have been made in the past in order to develop centrifugal pumps able to operate satisfactorily in the region of low specific speeds, specially for the case pf high pressure, low or medium flow rates and for moderate values of the rotatio. nal speed.

Although several models were developed [l] , [2] , [3.] » and so me of these unconventional centrifugal pumps are in operation, no satisfactory solution has been found of the problem, specia lly regarding to the aspect of pump efficiency.

This type of pumps would be specially suitable for reverse osmo sis application, where low specific speeds are presently needed and for which pumping requirements are specially stringent.

Operating pressures in reverse osmosis plants are high, reaching values of 1500 psi for sea water desalination, and this pressure has to be fairly constant, since pressure surges are particular- ly undiserable. Brackish and sea water are corrosive non-lubri- cant fluids; and plant economics require long-life, low maintenan ce and low cost pumps. In addition, pump efficiency has to be - high in order to reduce operating expenses.

No satisfactory pumping system presently available fulfills all this requirements, as shown, for example, in a recent report by the Office of Saline Water [4] .

A new type of centrifugal pump, specially designed for reverse osmosis applications has been developed by the Research Depart tnent of Sener, partially sponsored by the "Junta de Energia Nu clear" of Spain.

In this paper this new type of pump is described, including the hydrodynamic theory of the pumps as well as the stresses and vi^ tarati©n& calculations. In addition, the experimental results - and performances obtained with several working motlels are shown and discussed.

DESCRIPTION OF THE PUMPS

The basic principle of the Sener-STV pumps consists on the utilization of the dynamic pressure of a thin film of water *OT* any other suitable fluid rotating at high speed and mantained by centrifugal force against the inner surface of a rotary casing (Fig. 1.)

The dynamic pressure of this film of fluid is converted into static.pressure by means of a scoop tube, which enters into the film of fluid in a direction opposite to that of its motion. This scoop tube acts a,s a dif fuser, increasing its cross-section area- gradually.

The liquid is fed into the rotary casing by means of a s>stera of tubes rotating on a shaf£ coaxial with that of the rotary casing. Pressure rises within the tubes because of centrifugal force and the liquid is expanded at the end of the tubes by means of calibrated nozzles. The liquid is ejected at ambient pressure and - in the same direction as that of the tangential velocity of the rotary casing.

If the liquid enters into the rotary tubes at about ambient pressure, a simple calculation shows -that its relative ejection velqcity with respect to the nozzles it approximately equal to the tangential velocity of these nozzles. Therefore, if the rotary casing rotates at an angular velocity about twice as that of the tubes, the liquid is captured on this rotary casing at zero rela_ tive speed through a very efficient process. In this case, the rotary casing and tubes require different shafts, and the pump - may have a geometrical configuration as shown in Fig. 2 This two shafts pump will be denominated S-type pump.

For simplicity, and in order to reduce the cost of the pumps,both tubes and rotary casing may have a common shaft. Pumps are extremely simple in this case and they may be attached directly on the shaft of the power motor, and in this case the pumps do not need any bearings, although they may have smaller efficiency. Fig. 3 shows a belt driven pump of this type.

It is important that the film of the liquid in the rotary casing be thin, with a thickness approximately equal to the diameter of the inlet of the scoop tube, since in this way the hydrodynamic drag of this scoop tube is small.

This is achieved by adjusting the amount of liquid ejected from the nozzles of the rotar> tubes with respect to the maximum flow rate that the scoop tube ^an ingest.

Finally, it may be pointed out that STV pumps do not have high pressure seals and that they may not even have low pressure seals Tor several configurations. This added to their simplicity and lack of components working with friction make STV pumps extremely reliable and requiring little maintenance.

HYDRODYNAMIC THEORY OF STV PUMPS.

Rotary Tubes.- Assuming stationary state, ideal flow, one-dimensional conditions and that p. s p = p the relative ejection velocity w from the nozzles (frig. 4.a) a is given by:

$$w_t = \frac{\omega_t r_t}{\left[1 - \left(\frac{\sigma_t}{\sigma_i}\right)^2\right]^{\frac{1}{2}}} = \omega_t r_t \qquad (1)$$

since $\sigma_t \ll \sigma_i$

Introducing a friction loss coefficient $\phi_t = w_t / \omega_t r_t$ the absolute ejection velocity V is given by:

$$V_t = \omega_t r_t (1 + \phi_t) \qquad (2)$$

Disregarding aerodynamic losses, which are very small, the power consumed by the rotary tubes is given by:

$$P_t = \dot{m} (1 + \phi_t) \omega_t^2 r_t^2 \qquad (3)$$

where m is the flow rate.

Rotary Casing and Scoop Tube.- Motion of the film of liquid in the rotary casing is governed by the equations of the motion of a thin layer of fluid with a free surface at high Froude Number For the ideal case, a well-known hydraulic analogy [5] shows - that the study of this case can be reduced to that of the super sonic bidimensional motion of a compressible fluid.

However, since friction losses will have to be considered, the aforementioned treatment will not apply and a simpler method - will be utilized in which experimental coefficients will be in troduced.

Let V be the average absolute velocity of the liquid in the - film, and K_c an experimental coefficient defined as follows:

$$K_c = \bar{V}_c / \omega_c r_c \qquad (4)$$

x If h is the film thickness, Froude Number is equal to r /h.

Let D be the drag force transmitted by the scoop tube to the rotary casing and C_D the drag coefficient defined by:

$$C_D = \frac{D}{\frac{1}{2}\rho_1 \sigma_{st} \overline{V}_c^{\,2}} \tag{5}$$

in which ρ_1 is the density of the liquid.

In terms of these coefficients it is possible to express the power consumed by the rotary casing. It results:

$$P_c = \frac{1}{2}\rho_1 \sigma_{st} C_D K_c^2 r_c^2 \omega_c^3 r_{st} - \dot{m}\left[\cos\delta\, r_t \omega_t (1+\phi_t) - K_c r_c \omega_c\right] \tag{6}$$

In term of a friction loss coefficient ϕ_{st} in the scoop tube, the static pressure rise in the pump $p_o - p_a$, is given by the expression:

$$p_o - p_a = \frac{1}{2}\rho_1 \overline{V}_c^{\,2} \phi_{st}\left(1 - \frac{\xi_{st}^2}{\phi_{st}} \frac{\sigma_{st}^2}{\sigma_o^2}\right) + \frac{1}{2}\rho_1 \omega_c^2 r_c^2 K_c^2 \tag{7}$$

since $\sigma_{st}/\sigma_o \ll 1$. In this expression ξ_{st} is a scoop tube flow rate coefficient at the inlet area, defined by:

$$\xi_{st} = \sigma_o V_o / \sigma_{st} \overline{V}_c$$

Finally, hydrodynamic efficiency of the pump is given by:

$$\eta_H = \frac{\phi_t K_c^2}{\dfrac{C_D K_c}{\xi_{st}} - 2\left[(1+\phi_t)\dfrac{r_t^2}{r_c^2}\dfrac{\omega_t}{\omega_c} - K_c + (1+\phi_t)\dfrac{\omega_t^2}{\omega_c^2}\dfrac{r_t^2}{r_c^2}\right]} \tag{8}$$

Hydrodynamic efficiency is a function of speeds ratio ω_c/ω_t. Fig. 5 shows a typical curve $\eta_H = f(\omega_c/\omega_t)$. Powers P_c and P_t of the rotary casing and rotary tubes are also included.

It may be seen that efficiency is maximum for a ratio of ω_c/ω_t of the order of two.

Another case of interest is that corresponding to a ratio of » / w for which P = 0 . This is the so-called T-type pump, in which the rotary casing is drawn by the hydraulic force of the jets impinging on the rotary casing. This pump is simpler than S-type pump but it can be seen that it has smaller efficiency - and there is a considerable overflow of the liquid during starting.

STRESSES AND VIBRATIONS

Stresses.- Maximum pressure attainable by a STV pump is not limited by hydrodynamic considerations, as occurs, for example, in - conventional centrifugal pumps, in which for a given flow rate - friction losses increase as delivery pressure augments, till hydrodynamic efficiency becomes practically asero.

On the contrary, in STV pumps maximum delivery pressure is essentially limited by the stresses of the annular or cylindrical part of the rotary casing.

Assuming that stresses in that annular part (Fig. ^.b) are those existing in a cylinder of infinite lenght of equal thickness, it results for the maximum tangential stress:

$$(\sigma_\theta)_{max} = p_c \frac{1+\alpha^2}{\alpha^2-1} + \frac{3+\mu}{8} \rho_m r_c^2 \omega_c^2 \left[\frac{2(1-\mu)}{3+\mu} + 2\alpha^2\right] \qquad (9)$$

in which $\alpha = r_s/r_c$; ρ_m is the density of the material of the pump; y is the Poisson coefficient and p the pressure exerted by the liquid on the inner surface of the rotary casing. This pressure is given by:

$$p_c = \frac{\rho_1 r_c^2 \omega_c^2}{2}\left[1 - (1 - \frac{h}{r_c})^2\right] \qquad (10)$$

in which h is the film thickness.

In most cases, expression (9) gives similar results to those - obtained with the theory of thin-cylinders with constant tangential stresses.

In addition, a study was performed considering the actual case of a cylinder of finite lenght supported by a disc.

The problem was solved by assuming that both cylinder and disc - are thin, disregarding the influence of the rim and disregarding also the centrifugal force of the disc on its bending stresses.

The problem has a well-known analytical solution (see, for exampie, Ref. [6]). A numerical study of this solution showed that maximum stresses, which occurs at the rim end of the cylinder, are practically equal to thosu calculated with expression (9) - except for very narrow pumps.

Fig. 6 shows maximum delivery pressures of the pumps as function of design tangential stresses for several typical materials which might be utilized for STV pumps. Design tangential stresses have been selected by considering a safety factor of four with respect to ultimated tensile stresses.

It may be seen that very high pressures may be attained with these pumps, specially with titanium alloys and reinforced plastics.

Vibrations.- Lateral vibrations of the shaft of STV pumps have to be considered, specially in the configurations shown in Figs. 2 and 3 in which the rotary casing occupies a cantilever position with respect to the bearings.

A special feature of the pumps regarding to vibrations is that they work with a film of liquid, which has to be considered for calculating critical speeds.

It may be shown that the fundamental critical speed depends essentially on the mass M of the rotary casing, on the density _ and width L of the film of liquid, and on the elastic constant K of the shaft, which has to be calculated considering deflections of the bearings.

Pumps can be designed without difficulty with critical speeds much higher than the rotational speed of the pump.

PERFORMANCES

An experimental programme has been carried out with V-type and S-type pumps, utilizing pumps with delivery pressures up to 1500 psi, and for powers up to 100 HP.

Fig. 7.a shows typical performances curves of a STV-V pump. These curves are obtained at constant r.p.m. by throttling the outlet valve.

Flow rate is practically constant and the efficiency increases until maximum pressure is reached. If throttling is further continued, there is an overflow of the liquid over the rim of the rotary casing, but the pump may still operate under these conditions.

Fig. 7.b shows the general experimental performances curves of some typical STV-V pumps. It may be pointed out that for a given flow rate, hydrodynamic efficiency increases as pressure augments. This result is due to the fact that Reynolds number of both rotating tubes and scoop tube increases with pressure, thus decreasing fractional losses, and that no friction exists between a rotor and a stationary casing as in centrifugal pumps, which reduces the efficiency as pressure is raised.

Fig. 8 shows performances curves of STV-S pumps for two operating pressures. It may be seen that efficiency is significantly higher than in V-type pumps and that it reaches fairly good values.

Finally, it may be pointed out that mechanical losses in STV pumps are small. These losses occurs in bearings and in the trans_ mission by pulleys and belts or by a gear-box. Aerodynamic drag of the rotary casing.is very small and can be disregarded.

ENERGY RECOVERY

STV pumps can be utilized as turbines by reversing the flow and - after a number of modifications, specially on the rotary tubes.

A research programme is being carried out for the development of these turbines and combined turbo-pump groups-
Several studies have been carried out on the economics of pressu re energy recovery from turbine in reverse osmosis plants, and - very recently a study performed by Dynatech Co. for the Office - of Saline Water *k* , concludes that energy recovery is only - economical for large plants.

However, in that study pump and turbine are considered as sepa rated, and very expensive items, while STV turbd-pump groups - are less expensive and their cost is only a fraction higher than the cost of a pump.

Based on the estimated costs and performances of STV pumps and turbines, a parametric computerized study has been carried out by Sener on the economics of energy recovery in reverse osmosis plants utilizing STV turbo-pump groups.

As an example, in Fig. 9 some of the results obtained are shown The water production (or plant capacity) for which energy reco- very begins to be economical is given *as* function of the opera^ ting pressure for several values of the fresh water recovery fa^ tor and pressure drop throughout the plant.

It may be seen that by using STV groups energy recovery might be economical, even for small capacity plants.

ACKNOWLEDGEMENTS

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The collaboration given by the engineers of Sener,J. Valdes (co- inventor of the pumps), T. Velasco and P. Camarero, is fully ap- preciated.

PRINCIPAL SYMBOLS

D - hydrodynamic drag.

$\dot{m}$ - flow rate.

r - radius.

ω - angular speed.

μ - Poissons coefficient.

η - efficiency.

p - pressure.
P - power.
V - absolute velocity.
w - relative velocity.
ρ - density.
σ - area.
σ_θ - tangential stresses.
$\alpha\phi\xi$ - coefficients.

All remainder symbols are shown in Figs. 4.a. and 4.b.

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UN NOUVEAU TYPE DE TURBO-POMPE SYSTEME SPECIALEMENT DEVELOPPE POUR L'APPLICATION A L'OSMOSE INVERSE

On a développé un nouveau type de pompe pour des hautes pressions et des petits ou moyens débits spécialement appropriée pour les - plantes du dessalement par l'osmose inverse.

La principe fondamental de cettes pompes se base â l'utilisation de la pression dynamique d'un film d'eau ou de n'importe quel liquide approprié, maintenu contre l'intérieur d'un carter rotative â travers d'une force centrifuge. La pression dynamique du - liquide se transforme en pression statique dans un tube qui est introduit dans le film d'eau â une direction contraire a celle - de son mouvement. Le liquide est transporté au carter rotative - â travers d'un système de tubes radiaux qui termine avec un buse qui refoule le liquide tangentiellement. Ces tubes et le carter rotative sont coaxiaux et la relation de vélocités est optimum - lorsque la vélocité de rotation des tubes est d'un ordre de la - moitié de celle du carter.

Dans ce travail on expose la théorie hydrodynamique de cettes pom pes, ses calcula mécaniques et ses rendements expérimentaux, y - compris aussi un bref étude sur la utilisation possible de cettes pompes comme turbine pour la récupération de l'énergie dans les - plantes a osmose inverse.

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EIN NEUER PUMPEN-TURBINEN-TYP 6ES0NDERERS ENTWICKELT FUER UMKEHROSMOSE ENTSALZUNGANLAGEN

Es ist ein neuer Pumpentyp fuer hohe Druecke und kleine oder - mittlere Foerderleistungen entwickelt worden, die besonders fuer Entsalzungsanlagen durch umgekehrte Osmose geeignet ist. Das Grundprinzip diese Pumpe gruendet sich suf die Ausnutzung - des dynamischen Druckes eines duennen Wasser oder anderen geei£ neten Fluessigkeitsfilmes der durch eine Zentrifugalkraft gegen das Inners eines Drehgehaeuses aufrechterhalten wird. Der dyna mische Fluessigkeitsdruck wird in einem Rohr, das in den Flues- sigkeitafilm in umgekehrter Richtung zu seiner Fortbewegung ein getaucht wird, in statischen Druck umgewandelt, Die Fluessigkeit wird mittels eines Systems von Radialrohren, die an ihrem Ende - mit Mundstuecken versehen sind, die die Fluessigkeit tangential herausschleudern, zu dem Drehgehaeuse befoerdert. Diese Rohre - und das Gehaeuse sind coaxial, und die von diesen erreichten Ge- schwindigkeiten sind dann besonders gut, wenn die Drehgeschwindi^ gkeit der Rohre etwa die Haelfte der des Gehaeuses ist.

In der vorgelegten Arbeit werden die hydrodynamische Theoris die_ ses Pumpen, ihre mechanischen Berechnungen und ihre bei den Expe^ rimenten erzielten Leistungen dargestellt. Es wird ferner such kurk die Frage moeglichen Verwendung als Turbine fuer die Wieder_ gewinnung von Energie bei umgekehrten Osmoseanlagen gestreift.

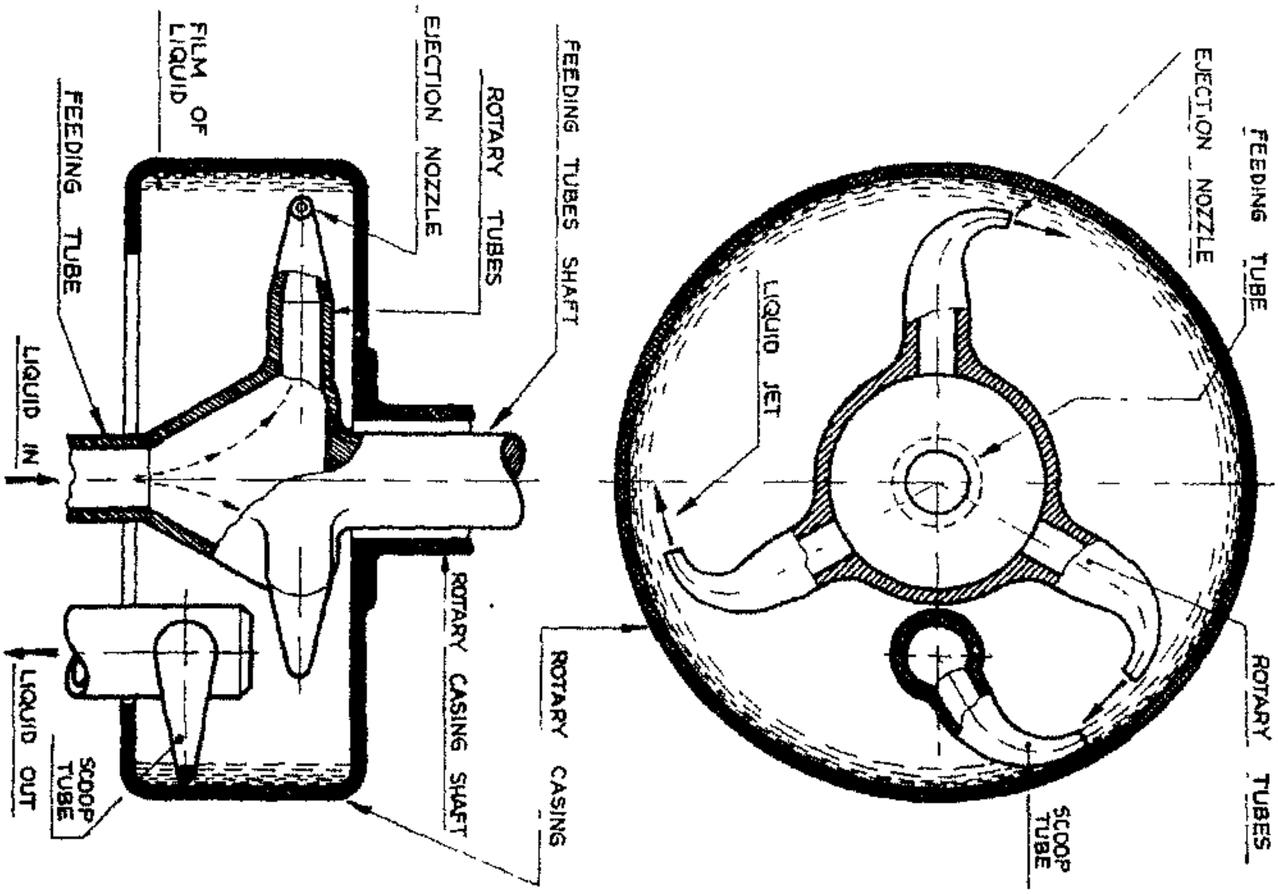


Fig. 1 Sketch of a SENER-STV Pump.

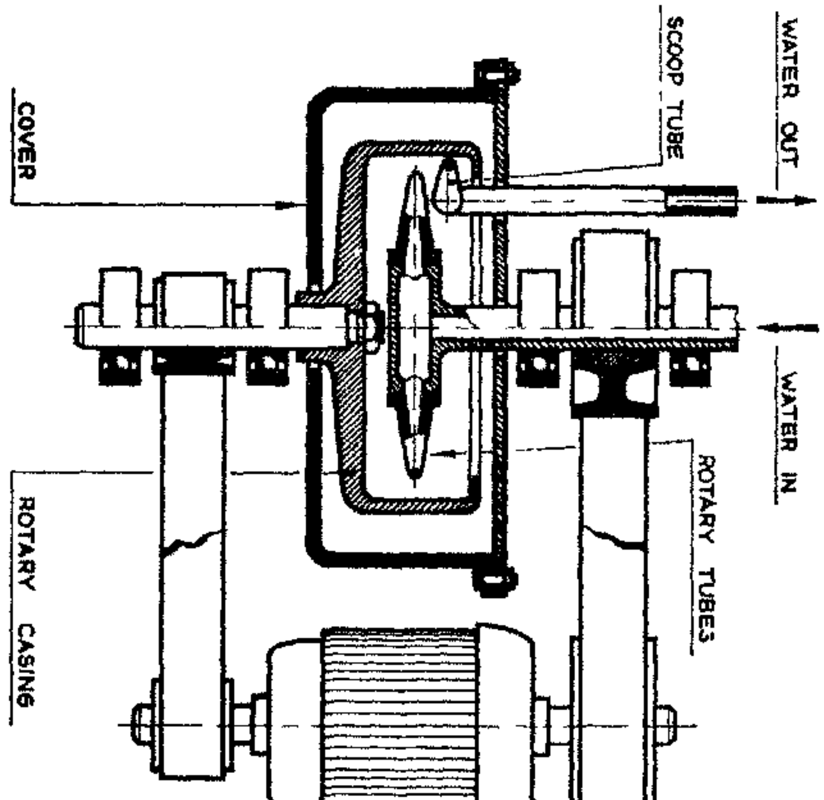


Fig. 2 Sketch of a Belt Driven STV-S Pump.

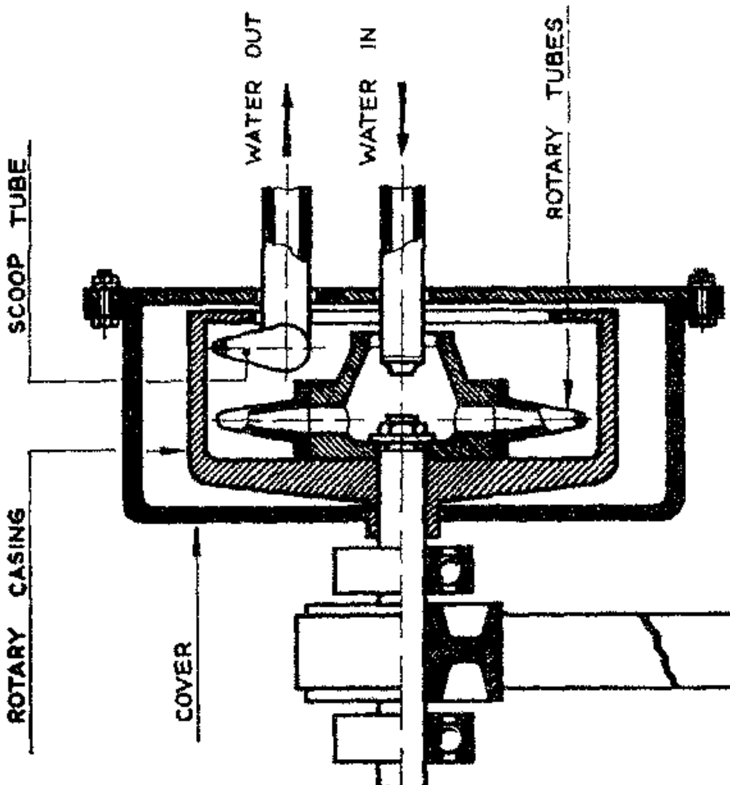
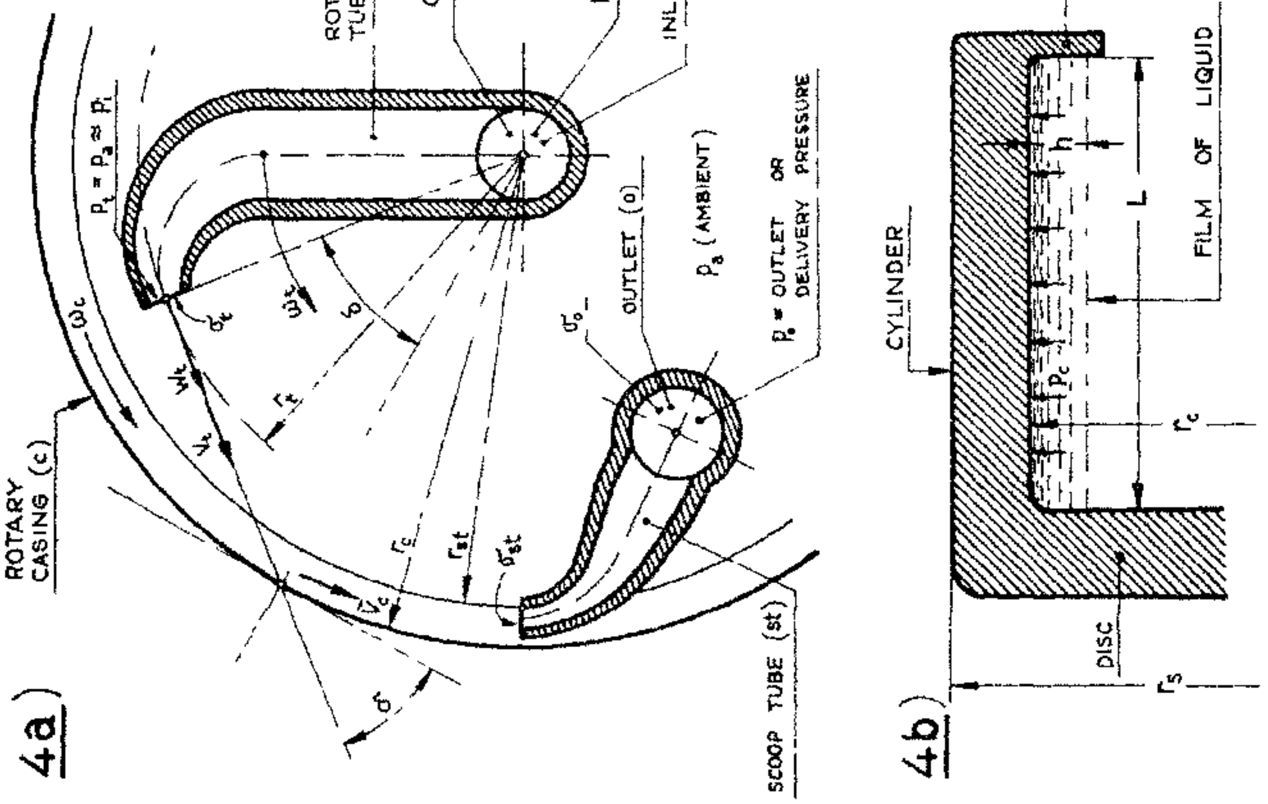


Fig. 3 Sketch of a Belt Driven STV-V Pump.



Figs. 4.a and 4.b Notation.

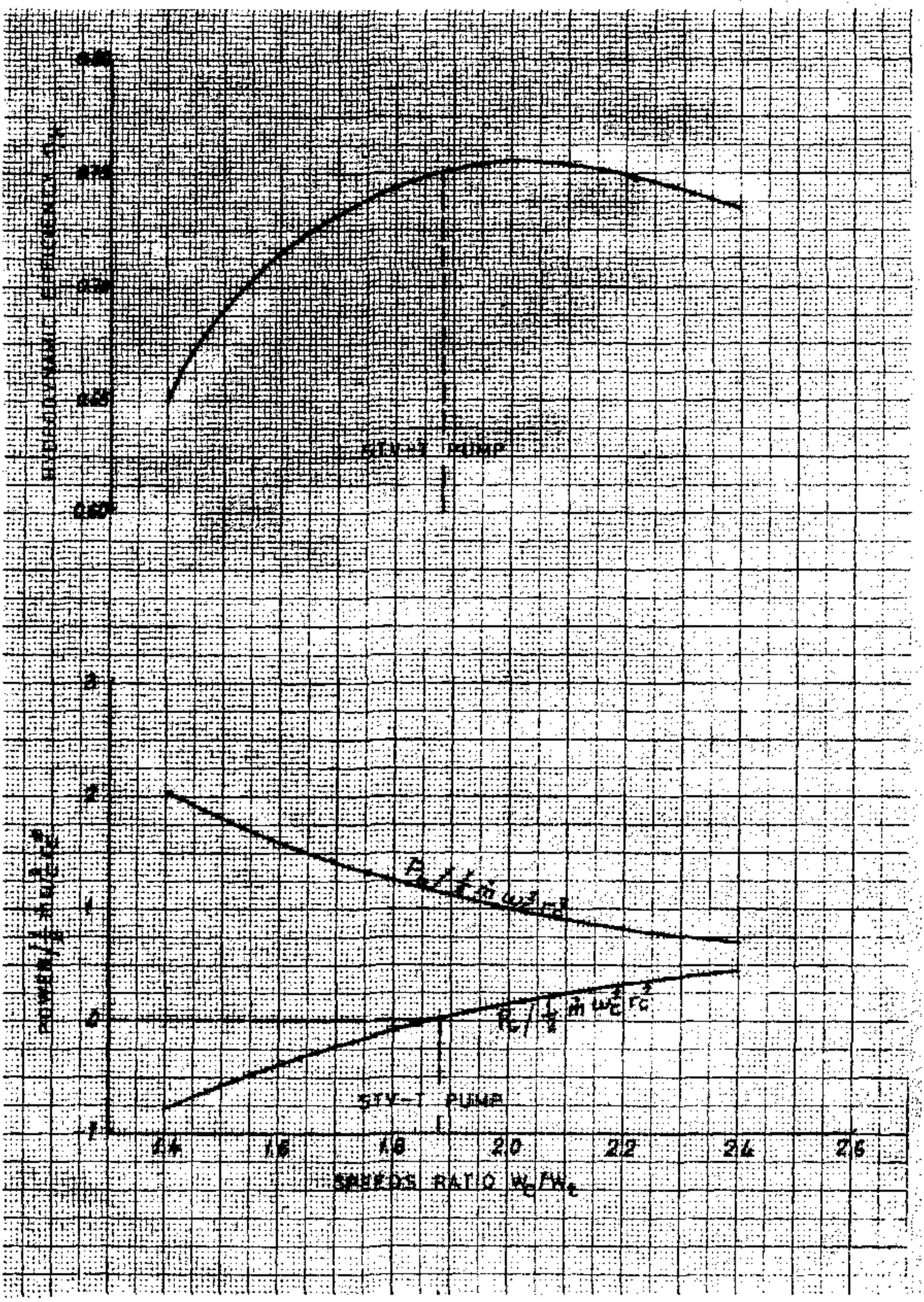


Fig. 5 Efficiency and Power Consumption as Functions of Speeds Ratio.

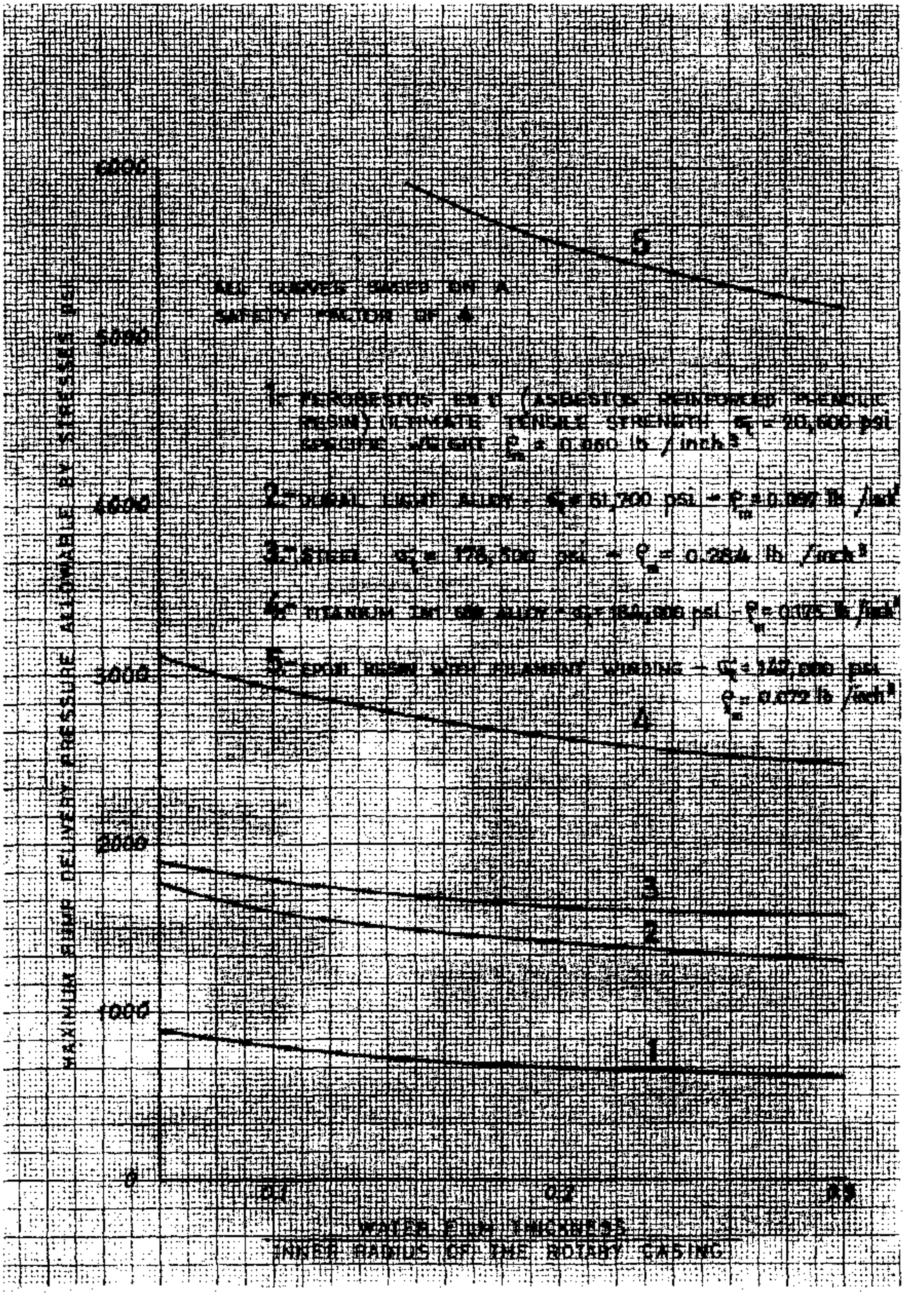


Fig. 6 Max. Pump Pressure for Several Materials.

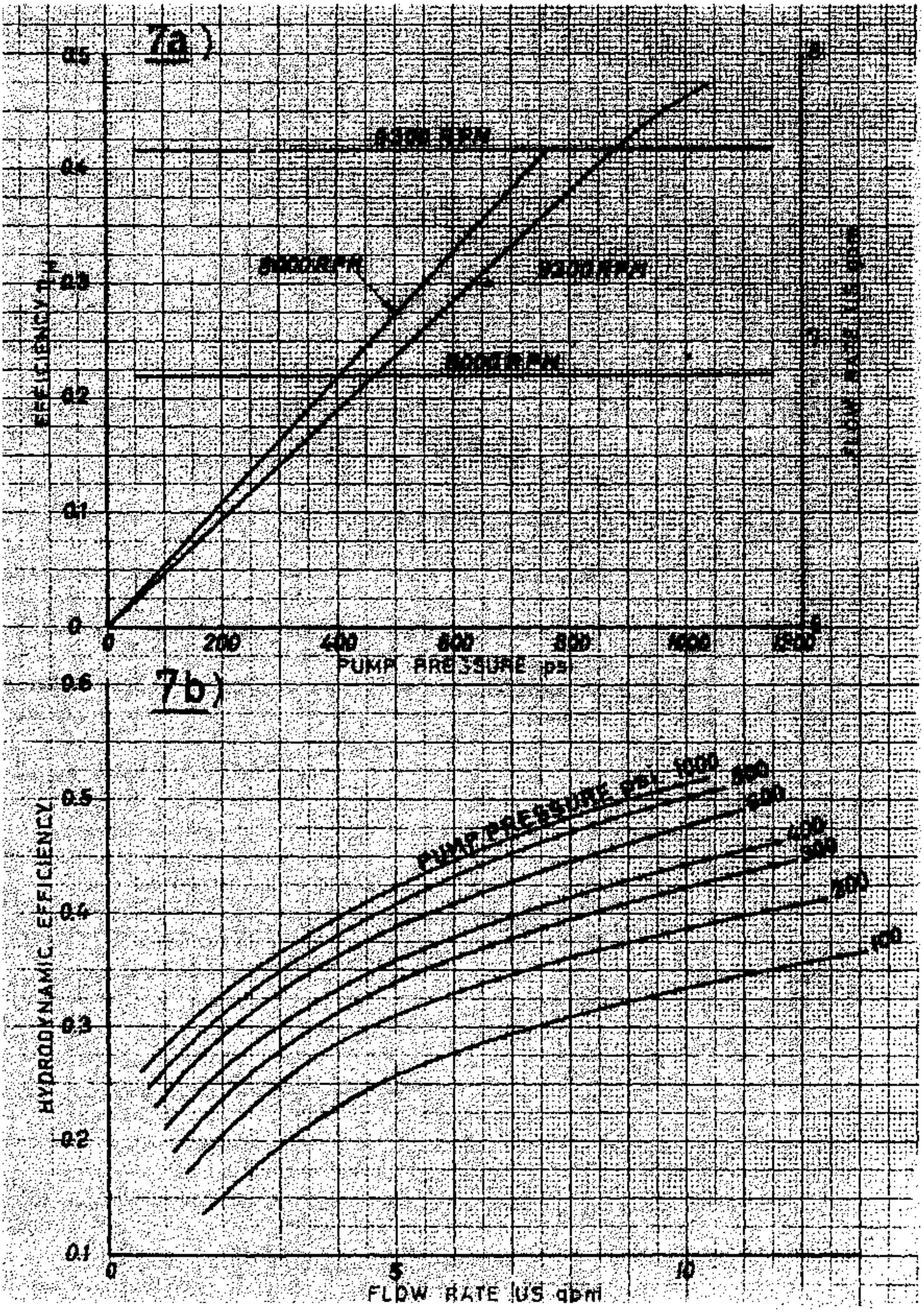


Fig. 7.a and 7.b Performances of STV-V Pumps.

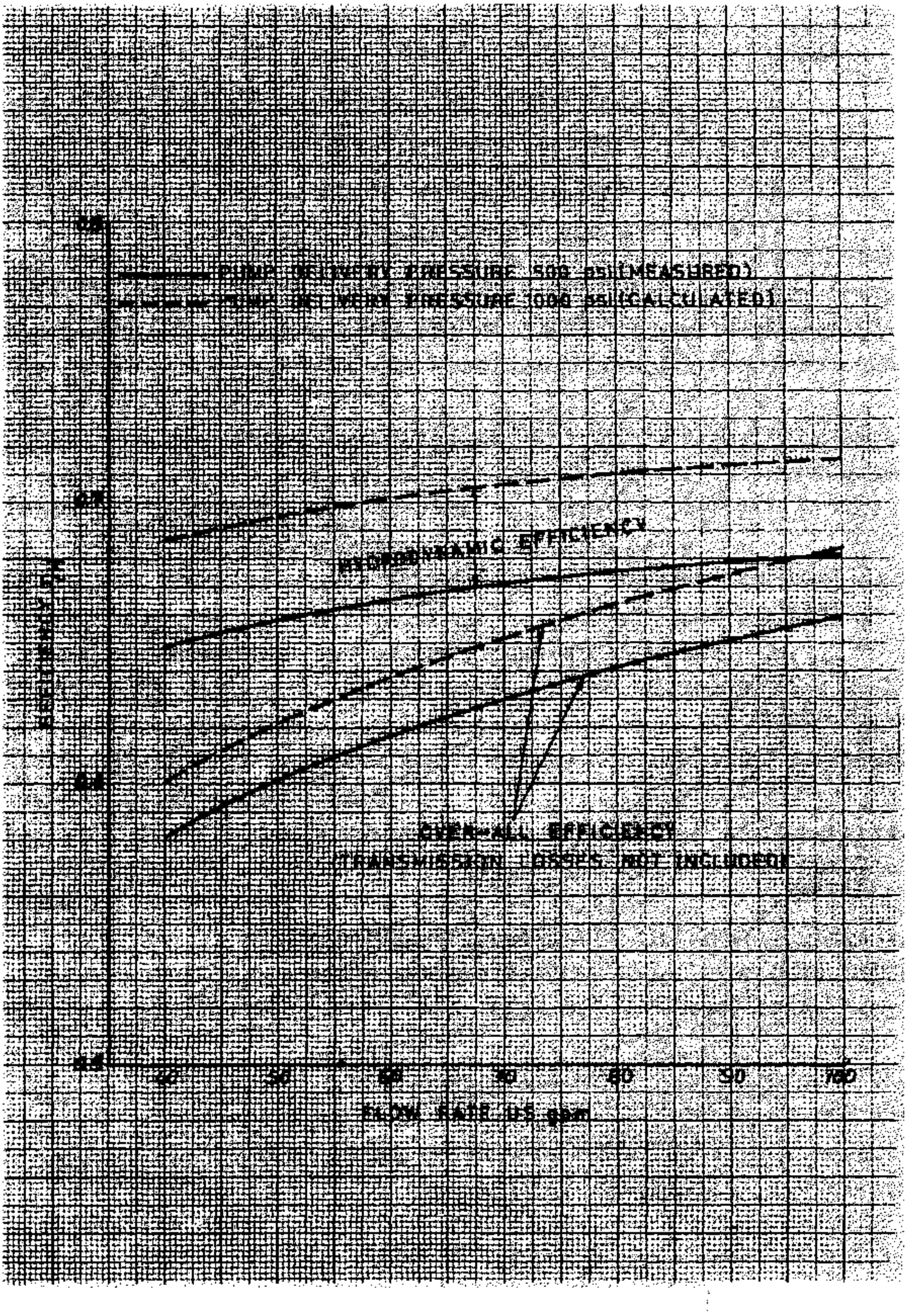


Fig. 8 Performances of STV-S Pumps.

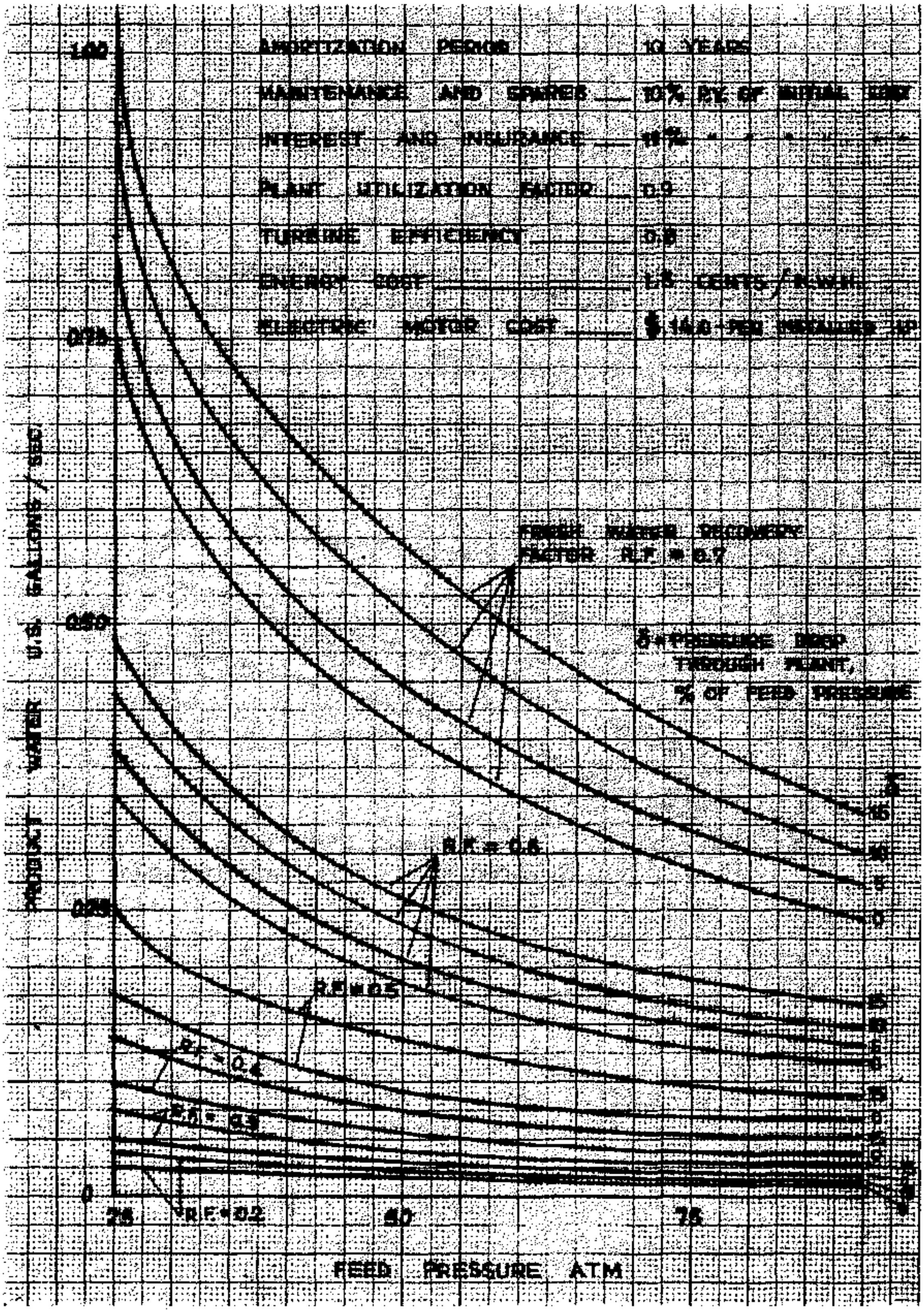


Fig. 9 Economics of Energy Recovery.