ARCHIMEDIAN SCREW PUMP HANDBOOK

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Fundamental aspects of the design and operation of water pumping installations using Archimedian screw pumps.

RITZ-Pumpenfabrik OHG Schwäbisch Gmünd

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Preface

This book has been prepared by RITZ Pumpenfabrik OHG to provide the technical information needed for the calculation, planning, construction and operation of water pumping installations using Archimedian screw pumps. The constantly increasing use that ist being made of such pumps, especially for handling effluents, has resulted in an urgent need for a comprehensive handbook of this nature.

Since this book is issued by a pump manufacturer, particular care has been taken to ensure that an objective approach was maintained when evaluating screw pumps and comparing them with other types of pump, and in the discussion of the advantages and disadvantages of this type of pump. This objectivity is assured because RITZ Pumpenfabrik OHG are manufacturers of all types of effluent pump, and so have no inducement to recommend any particular type of pump other than on strictly operational and economic grounds. The book thus, at the same time, seeks to shed some light on the many different opinions and conceptions regarding the use of screw pumps both as drainage pumps and effluent lift pumps. The technical information is based on ten years experience in the construction and use of screw pumps for pumping water. Particular attention has been paid to providing practical explanations and hints, and theoretical derivations have been included only to an extent necessary to ensure an understanding of the subject as a whole.

It is our hope that this publication will act as an incentive for further improvements, and that it will prove a source of help and advice for the planners and designers of water pumping installations employing Archimedian screw pumps.

If this hope is fulfilled it will be due very largely to all those who have contributed to the success of this book by their valuable suggestions. In particular, acknowledgement is due to Herrn Masch.-Ing. Deck of the Water Supply Department of the North Baden Regional Government at Karlsruhe, who readily placed his experience at our disposal in the form of calculated examples and practical suggestions.

The Author

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Part 1 General

1.1 A brief history

The Archimedian screw pump can rightfully claim to be the oldest type of pump to be used for the conveyance of liquids. In the third century B.C. the Greek mathematician and physicist Archimedes invented the "Archimedian Screw", to which he gave his name, and even in those days this device found practical application as a means of pumping water.

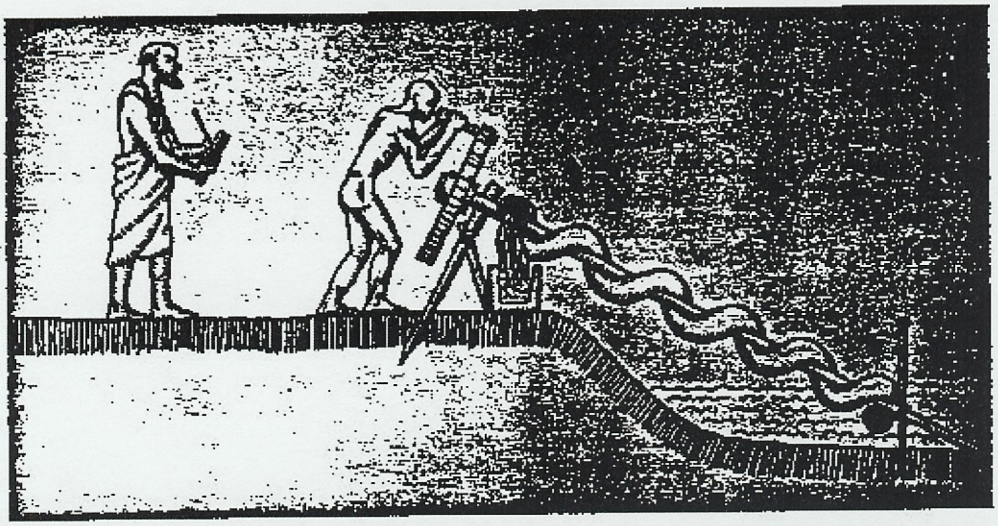
By the next century the Archimedian screw pump had also been adopted by the Romans, who used it in the water supply system that served ancient Rome which, even at that time was developed to a high standard, as well as for irrigation and drainage work. A multi-stage Roman pumping station of this type is shown in Fig. 2.

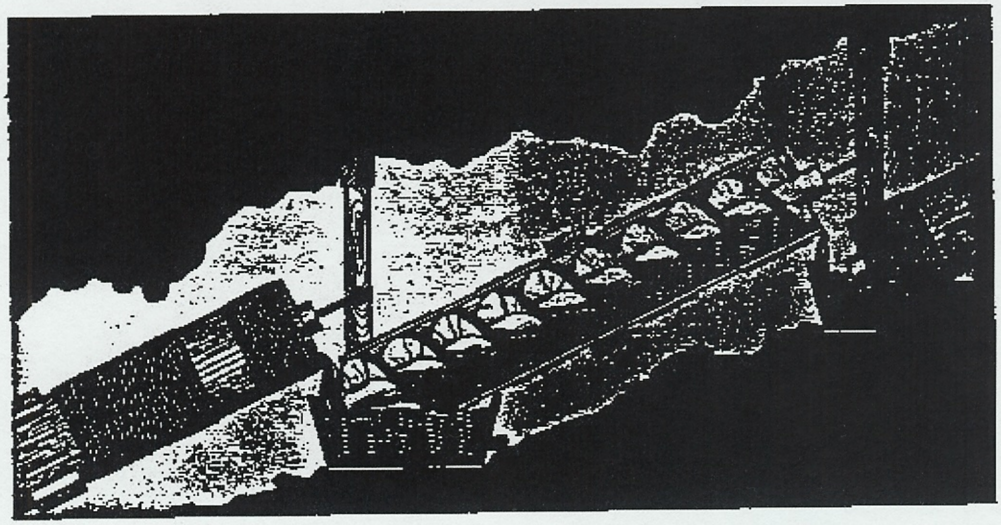
If the Archimedian screw shown in the contemporary illustration (reproduced as Fig. 1) is compared with the Roman pumping station with screw pumps (Fig. 2), which is only about 50 years more recent, it will be seen that the latter ins much closer to the present-day conception of Archimedian screw pumps. The high stage of development that had been attained in the design of water supply systems is reflected in the multi-stage arrangement of the Roman installation.
At a later date the Archimedian screw pump was frequently to be found in ore mines in Spain.

In relatively recent times the principle of the Archimedian screw pump as a means of conveying is encountered only from the 14th century onwards. In particular, historical records refer to the use of the Archimedian screw pump in connection with artifical fountains. Because of its relatively low delivery head, little use was made of the Archimedian screw pump when the main public water supply systems were installed during the 19th century especially since, by then, the "classic" pump for dealing with high heads in the form of the reciprocating plunger pump had become available.

For this reason the Archimedian screw pump was then to be found only where comparatively large quantities of water had to be lifted small distances. One obvious application of the Archimedian screw pump was a drainage pump in low-lying areas, such as the reclaimed land areas of the North Sea and Baltic, and as a drainage pump. A number of these installations, in some cases of wooden construction, are still preserved and in operational condition.

The methods adopted for driving these pumps was another factor that led to them being used primarily for drainage purposes. The state of technical progress at the time of both Archimedes and the Romans was such that they only had human or animal power available, but in the meanwhile the use of wind force had been found to be an ideal and also inexpensive means of driving screw pumps in windy, low-lying coastal areas. Around 1930, there were about 300 Archimedian screw.





1 Contemporary illustration of an " Archimedian Screw "

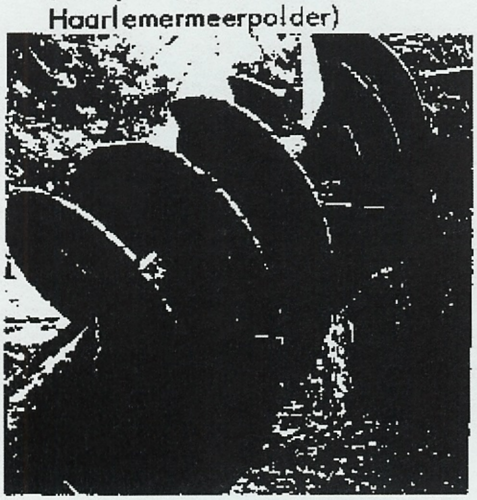
2 Roman water pumping plant using screw pumps, dating from about 200 B.C.

pumps in use in Holland alone for draining low-lying ground, and in flood-tide pumping installations, by far the majority of which were driven by wind power.

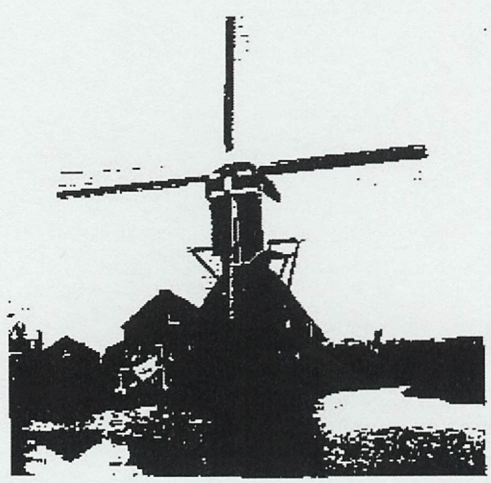
In Germany, however, conditions were less favourable and the use of Archimedian screw pumps become less and less common until the turn of the last century, with a few exceptions on the Baltic coast. As a result, the manufacture of Archimedian screw pumps came to a complete halt in Germany, the few installations required being obtained from Dutch manufacturers. These circumstances led to the erroneous assumption that the Archimedian screw pump was a typically Dutch device.

The economic resurgence of Germany after the last war, accompanied as it was by a general increase in living standards, led to the construction of many main sewage systems and central treatment plants. The assumption made at that time, that the centrifugal pump (which had been invented at the turn of the century) would also prove a satisfactory means of pumping effluent under all conditions, proved to be false. In particular, the trends that had been followed in the construction of centrifugal pumps for handling fresh water, of reducing weight so far as possible, and of operating at higher specific speeds, led to considerable difficulties where effluent was being pumped.

3 18th century Archimedian screw pump used for pumping water (Cruquius pump museum, Haarlemermeerpolder)



4 Archimedian water pump driven by wind power.



Although considerable doubts were expressed, the short effective life that was often experienced where centrifugal pumps were used for pumping effluents, and the high rate of repairs that was involved, led to the decision in 1955 to acquire two Archimedian screw pumps from a Dutch manufacturer and to instal them for experimental purposes in the pumping station of the Weinheim-Nord/Bergstraße treatment plant where, in the past, a number of centrifugal pumps had given unsatisfactory service.

The measurements of performance and efficiency that were made on these Archimedian screw pumps after they had been installed provided a pleasant surprise. Since the subsequent experiments, in which the pumps were subjected to additional loading in the form of abrasive materials and substances that would readily twist round a rotating part yielded equally good results, the authorities concerned decided to instal further Archimedian screw pumps for handling effluents and for drainage. Plants of this type were soon installed at Forchheim near Karlsruhe, at the Ketsch-Brühl cooperative drainage scheme, at Oftersheim, and subsequently at many other places in the South of Germany, with German firms once again taking up the manufacture of these pumps.

The satisfactory results that were obtained in service, the robustness of the equipment, and the small amount of meintenance required led to an increasing demand for Archimedian screw pumps. RITZ PUMPENFABRIK KG, of Schwäbisch Gmünd, who are specialist manufacturers of pumps for drainage purposes, also played their part in this development, and after extensive theoretical investigations started, in 1962, to construct Archimedian screw pumps for drainage purposes.

The decision to enter this field has been fully justified by the results obtained with the RITZ Archimedian screw pumps which have since been installed in many plants, and by the demand that has developed for these pumps. In view of this, these manufacturers have continued to make extensive investigations into methods of improving and further developing the Archimedian screw pump so as to be able to offer their customers pumps that incorporate all the latest technical developments.

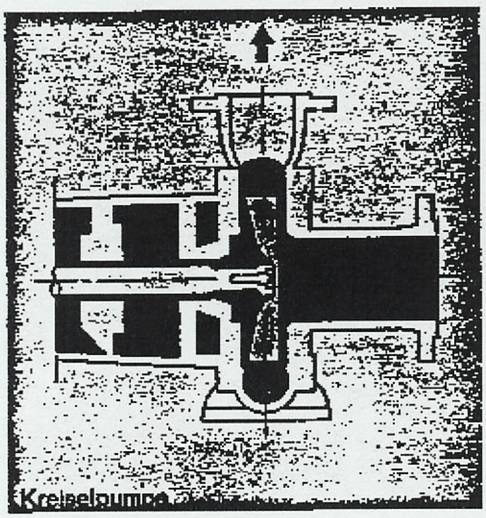
1.2 Comparison with centrifugal pumps

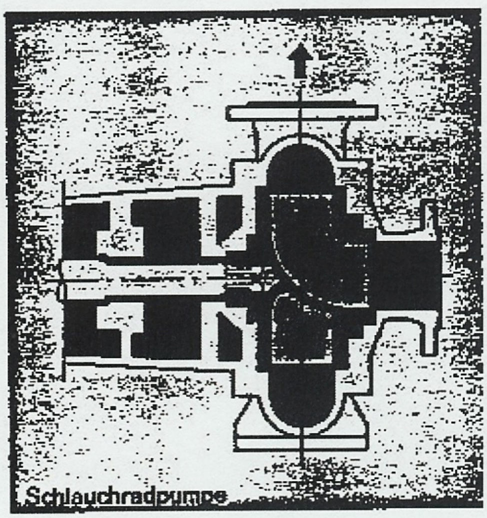
The various centrifugal pumps, such as radial-flow, axial-flow, mixed-flow, guide-passage, and screw impeller pumps that were used for drainage duties and for handling effluents prior to the introduction of the Archimedian screw pump are illustrated, for comparison, in Fig. 5.

Because of the high pumping speeds necessary to produce the required rate of flow, and the consequent high speed of the pumped material as it passes through the pump, such pumps are subject to heavy wear, especially where there is an abrupt change of flow. Both the abrasive action of solid particles that are flung out by the centrifugal action, and stoppages of the centrifugal action in the rotor, due to the presence of substances in the water having a different specific weight, are the main cause of breakdowns. It is therefore almost essential to fit suitable screens at the input to those centrifugal pumps that are used in drainage and sewage pumping plants, if reliable operation is to be assured, since there is always the possibility of bulky or other easily-transportable objects being carried along with the effluent. This proves of particular disadvantage in the case of pumping sub-stations, especially where these are located in built-up areas. In addition to the additional cost of the screens and the need for extra maintenance and attention, the reliability and efficiency of the pumping operation is often endangered. The fact that an Archimedian screw pump is considerably more silent in operation than a centrifugal pump can also often prove an argument for the use of the former pump if there is any danger of the pumping station causing undue public annoyance.

To enable an objective comparison to be made it is necessary to study the delivery characteristics of the two types of pump, both of which are shown in Fig. 6.

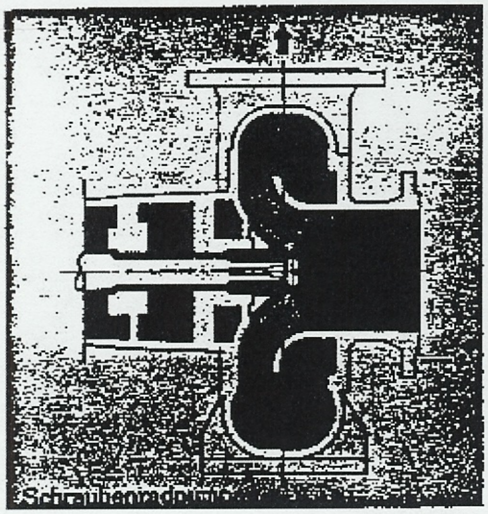
A comparison of the delivery characteristics of the two types of pump shows that a drainage pump that operates on the centrifugal principle can work effectively at the designed delivery-head, and when supplied with the appropriate quantity of



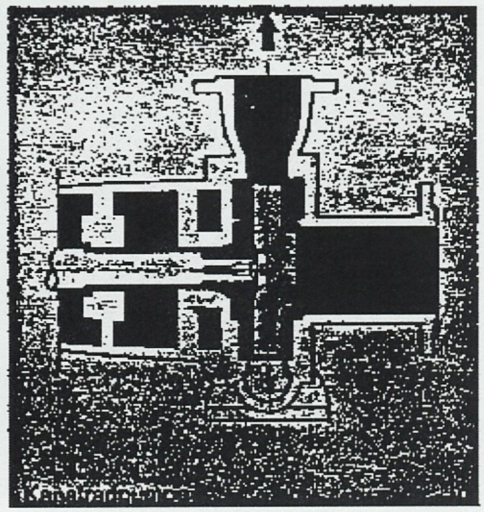


Radial-flow pump

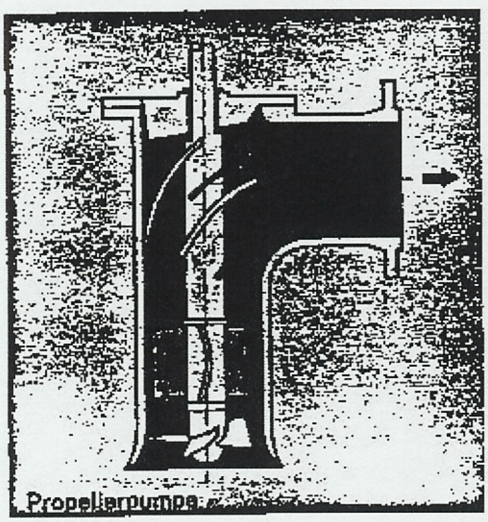
Guide-passage pump



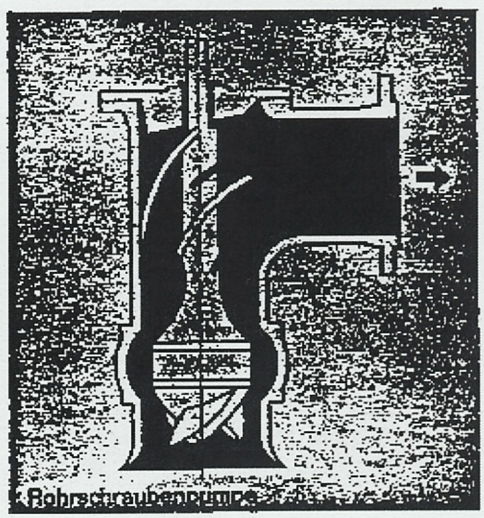
Mixed-flow pump



Enclosed-rotor pump



Axial-flow pump



Screw-impeller pump

5 Diagramatic sketches of various types of centrifugal pump used for drainage duties

fluid that it is designed to handle. Even if damage caused by the pump running partially empty (or even completely dry) is ignored, a centrifugal pump will run at high efficiency only within a very limited range of conditions.

An Archimedian screw pump, on the other hand, can be used for all rates of flow from zero delivery to its maximum capacity without any danger of undue wear or even of damage occurring. A remarkable feature of the latter pump is the very wide range of operating conditions under which it will attain a high efficiency, this extending from about one-third of the rated delivery to the full delivery of which the pump is capable. It is impossible for the prime mover that drives the screw to become overloaded, since the power consumption reduces if the screw runs overfull and floods, or if it is filled below its capacity. Dry running has as little adverse effect on the Archimedian screw pump as does the presence of abrasives, twisted materials, or solid substances, so that one of the major advantages of this class of pump is that it does not require any screens to be fitted.

In addition, the use of centrifugal pumps makes it essential to provide storage reservoirs, or at least to make provision for damming the pipe systems to attain acceptable duty cycles without constant starting and stopping of the pump. In dry weather the effluent will therefore require to be stored for a lengthy period. To prevent the effluent deteriorating, it is, however, desirable that it be passed to the treatment plant as quickly as possible.

The extent to which the use of intermediate storage reservoirs causes an increase in the delivery head of centrifugal pumps, with a consequent drop in efficiency, is indicated in Fig. 7.

As is shown in this illustration, the need for the centrifugal pump to lower the level of fluid in the intermediate storage reservoir, during the periods that it is running, represents an additional head that has to be overcome. In addition, friction losses in the suction and delivery pipes must be allowed for; such losses do not occur in the case of an Archimedian screw pump or, rather, they are replaced by the internal fluid friction losses in the pump itself, and so are included in the figures quoted for the overall pump efficiency. The increase in delivery head for the centrifugal pump will depend on various factors, but it often forms an appreciable proportion of the total delivery head (where this is low) so that the power required to lift a m^3 of water attains a value which alone would, in most cases, justify the adoption of an Archimedian screw pump.

To ensure that the comparison is truly objective it is, of course, necessary also to mention the disadvantages of the Archimedian screw pump, and the limits within which it can be used. The costs of the engineering equipment of pumping stations equipped with Archimedian screw pumps are, in general, considerably higher than if centrifugal pumps were employed and, in addition, there must be sufficient space for the installation of a lengthy structure. The starting current for the electric drive motor of a screw pump is

QH-Linie einer Wasserförderschnecke	=	QH-line for an Archimedian screw pump
Auslegpunkt	=	Design point
QH-Linie einer entspr. Zentrifugal pumpe	=	QH-line for the corresponding centrifugal pump
Schn + Getr	=	Screw + gearing
Zentrifugalpumpe	=	Centrifugal pump
Arbeitsbereich einer Wasserförder-schnecke	=	Useful range of an Archimedian screw pump
Arbeitsbereich einer Zentrifugalpumpe	=	Useful range of a centrifugal pump
Q_{mittel}	=	$Q_{average}$
WFS	=	Screw
Krp	=	Cent.

Abfluß	=	Discharge
Druckverlust	=	Delivery loss
Gleiche Abflußhöhe	=	Same discharge head
Sumpffluktuation	=	Sump level fluctuation
HWS	=	HWL
MWS	=	MWL
NWS	=	LWL
Saugverlust	=	Suction loss

6 Comparison of delivery characteristics of a centrifugal pump and an Archimedian screw pump

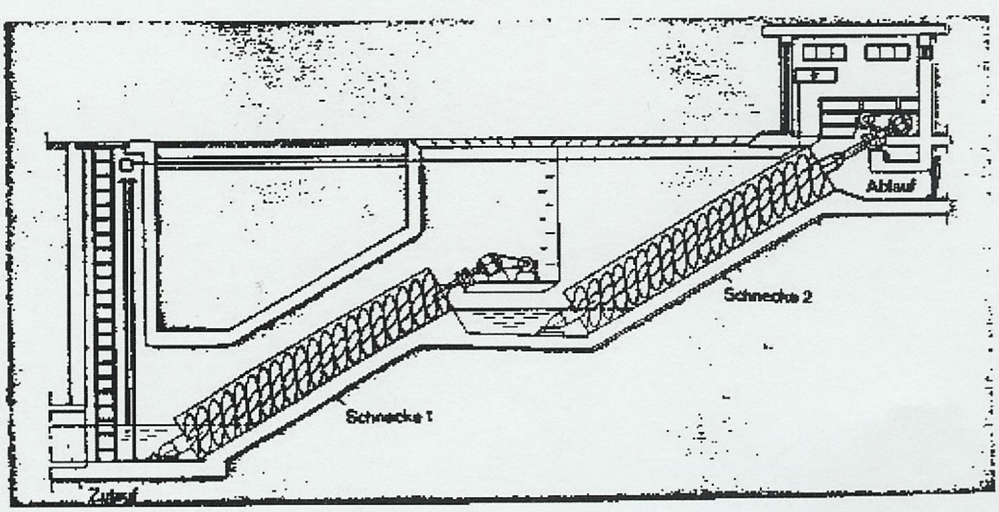
7 Comparison of delivery heads for a centrifugal pump and an Archimedian screw pump

higher than that for a centrifugal pump, and, also, the current drawn when the pump is running is less uniform due to the immersion and emergence of the blades of the screw at the suction end, and the impact of the blades on the water in the delivery tank. The necessary reduction of the motor speed to the speed of the screw also represents an additional expense.

The main disadvantage in the use of an Archimedian screw pump is, however, the limited delivery head that can be attained. For a single screw-pump unit this will be limited by the length of screw that can be achieved, taking into account the deflection due both to its own weight and to the weight of the medium that it has to pump. Even if the wall thickness of the shaft tube is increased, and if the screw pump is installed at a steep angle the small diameter of the shaft tube will limit the attainable delivery head, for a low-capacity screw pump, to 4 to 5 metres, while the single-stage delivery heads of the largest units built to date amount to 8 metres. Further information on this point will be found in the Chapter "Determination of screw length", and in Diagram 1.

If a two-stage Archimedian screw pump as shown in Fig. 8 is regarded as the maximum that could be contemplated, in view of the extensive structural work involved, the figures quoted above can be doubled. Even with the lower capacity units it is then possible to attain delivery heads, which, in most cases, will meet the requirements for the pumping of effluents and for drainage purposes.

Zulauf	=	Inlet
Schnecke	=	Screw
Ablauf	=	Discharge



In this general discussion on drainage and effluent pumping it should finally be pointed out that the use of an Archimedian screw pump is possible only where the liquid that has been raised can drain under gravity from the upper discharge basin. Possibilities of improving the economics of Archimedian screw pumps, where the level of the liquid in the upper discharge basin varies, will be discussed in greater detail in Part 2, although it must be pointed out that, wherever possible, this type of measure should be avoided – especially when dealing with effluents and drainage.

PART 2 BASIS OF CALCULATIONS

2.1 Symbols and Dimensions

Even though it is not the primary aim of this book to provide the designer of Archimedian screw pumps with the necessary design information, some theoretical discussion is unavoidable if the various relationships are to be understood sufficiently for practical application.

The fact that no relevant literature exists in the German language emphasises the desirability of this approach, since technical terms relating specifically to the Archimedian screw pump cannot be assumed to be generally familiar.

The following discussions are based on publications by Muysken (1), Horch (2), Bekkering (3), and Delprat (4), which are all listed in the bibliography at the end of the book (Part 6). The publications on this subject, some of which are more than 50 years old, have had to be thoroughly revised, especially as they are concerned solely with the use of Archimedian screw pumps for land reclamation purposes, where the delivery heads involved are usually low, and since no allowance is made for the features peculiar to the handling of effluents. The latest work to be published which, like all the others is in Dutch, is that by Muysken (1); this dates from 1932 and covers the determination of efficiency in some detail. The first article to deal with the subject in any depth in the German language was that by Nagel (5) in 1959; this discussed the use of Archimedian screw pumps for drainage and sewage pumping duties on the basis of the latest developments in this field, but was couched in very general terms.

The following symbols and dimensions have been used in the discussion of the basic theory or Archimedian screw pumps, their geometric forms, and their dimensions :

D	=	External diamter of screw	(m)
R	=	Radius corresponding to D	(m)
d	=	Diamter of screw shaft tube	(m)
r	=	Radius corresponding to d	(m)
L	=	Length of screw equipped with blades	(m)
S	=	Pitch of one screw thread	(m)
σ	=	$\frac{D}{S}$ = ratio	(non-dimensional)
δ	=	$\frac{d}{D}$ = ratio	(non-dimensional)
β	=	Angle of inclination of screw to horizontal	(deg)

s_{sp}	=	Gap width = Clearance between external diamter of screw and through	(m)
s_w	=	Wall thickness of shaft tube	(mm)
a	=	No. of starts of screw	(non-dimensional)
h	=	$\frac{S}{a}$ = Flank clearance between two adjacent screw threads	(m)
H	=	Effective useful delivery head	(m)
I	=	Volume of body of water between two adjacent sc.ew theeads	(m^3)
Q_n	=	Nominal delivery = theoretical delivery	(m^3/s)
Q	=	Effective delivery	(m^3/s)
Q_1	=	Leakage loss	(m^3/s)
N_n	=	Effective useful work performed by screw	(h.p.)
N_s	=	Power required to drive screw shaft	(h.p.)
N_A	=	Power required at gearbox flange	(h.p.)
N_{vm}	=	Mechanical power losses	(h.p.)
N_{vh}	=	Hydraulic power losses	(h.p.)
N_{vl}	=	Leakage power losses	(h.p.)
N_{va}	=	Discharge losses	(h.p.)
N_{vr}	=	Losses due to water friction	(h.p.)
η_{ges}	=	Overall efficiency of screw and gearing	(%)
η_s	=	Screw efficiency	(%)
η_m	=	Mechanical efficiency	(%)
n	=	Speed of rotation of screw	(rev/min)
v	=	Velocities of motion and of flow	(m/s)
λ	=	Ratio of leakage to nominal delivery $\frac{Q_1}{Q_n}$	(non-dimensional)
ε	=	Delivery increase ratio = $\frac{Q}{Q_n}$	(non-dimensional)
I_1	=	Leakage loss factors	(non-dimensional)
q	=	Delivery factor	(non-dimensional)
e	=	Frictional loss factor	(non-dimensional)
z	=	Resolution factor for graphical method	(non-dimensional)
μ	=	Coefficient of friction	(non-dimensional)
i	=	Switching frquency	(operations/hr)

2.2 Design Terminology

A knowledge of the following major features is required, when planning a water pumping installation, if a plant is to be installed that is capable of meeting certain basix requirements as regards delivery rate and delivery head.

a) The angle to the horizontal at which the screw is installed
b) The external diameter of the screw
c) The trough dimensions
d) The diameter of the screw shaft or tube
e) The number of threads or starts on the screw
f) The pitch of the screw
g) The speed of rotation of the screw
h) The length of screw that is fitted with blades
i) The inflow conditions or the design of the bottom end of the screw
k) The discharge conditions or the position of the spill point

To enable these values to be determined with sufficient accuracy for constructional purposes it is necessary to make a thorough investigation in any particular case both of the major features listed above and also of the side-effects which are influenced by them; the procedure for this is briefly described below.

It is also necessary to introduce the terms contact point, filling point, spill point, and non-return discharge basin level, all of which are explained in the text and the illustations.

2.3 Determination of Delivery

The basis for determining the capacity is the calculation of the unit volume of water. This is that volume contained between two adjacent blades on the screw and the riser tube. During every revolution of the screw each blade picks up this volume of water from the supply basin and moves it upwards. The delivery of the pump will then be the product of this unit of water, the number of starts of the screw, and the speed of rotation of the screw.

The volume of this unit of water, and hence the capacity of the screw, can be varied for a given external diamter by varying the speed or by varying the following factors (which may involve a change in the form of the screw :

a) the ratio of the pitch to the external diamter $= \frac{S}{D} = \sigma$

b) The ratio of the diamter of the screw shaft tube to the external diamter $= \frac{d}{D} = \delta$

b) The inclination of the screw to the horizontal $= \beta$

It is not possible to derive a simple relationship which will show, in a purely arithmetical manner, the way in which the unit of water, and hence the delivery of the pump, varies with σ , δ , and β . The volume of a unit of water for a series of screw forms was therefore determined by the graphical process (1) as shown in Fig. 9, and the values obtained have been listed as the q-values in Table 3 (in Part 5 - Tables) to enable the delivery to be determined. These values were obtained with δ varied between 0.4 and 0.65, β between 22° and 40°, and the number of starts between 1 and 3, while a fixed value of 1.0, based on experience, was used for $\sigma = \frac{S}{D}$. Intermediate values of q can be interpolated with sufficient accuracy for practical use from the q-values given in the Table on p. 123 (p.110 I).

The graphical method for the determination of the delivery is also employed for determining other factors, e.g. the leakage loss and the efficiency.

To enable the quantity of water between successive blades to be determined graphically, a vertical projection of a unit volume of water onto a plane parallel to the screw axis is shown in Fig. 9. Here C-D is the screw axis and A-B the water surface corresponding to the angle of inclination β , while E-F and G-H are sections through the upper screw blade and I-J and K-L are sections through the lower screw blade, and represent the water surface at the points of intersection.

By dividing the diamter of the screw into a number of shells, it is possible to calculate the quantity of water between two adjacent blades by measuring the angles of the arcs to the upper and lower water surfaces; for each shell then the approximate quantity.

$$\frac{\alpha 1 \stackrel{+}{} \alpha 2}{2 \cdot 360} \cdot 2\pi \cdot r \cdot h \cdot \frac{D}{z}$$

is calculated. The unit volume of water is then

$$1 = \frac{\pi \cdot S \cdot D^2}{360 \cdot Q \cdot z \cdot a} \cdot \Sigma f (\alpha_1 + \alpha_2)$$

and the nominal delivery for a screw is

$$Q_n = \frac{\pi \cdot S \cdot n \cdot D^2}{a \cdot 60 \cdot 360 \cdot z} \quad \Sigma\, f(\alpha_1 + \alpha_2).$$

Substituting

$$\frac{\pi \quad \cdot \sigma}{60 \cdot 360 \cdot z} \sum f \cdot (\alpha_1 + \alpha_2) = q,$$

the principal formula for the determination of the theoretical delivery becomes

$$\boxed{Q_n = q \cdot n \cdot D^3.}$$

A number of values of q for the normal screw forms and pitches of three-start screws is given in Table 3 of Part 5 - Tables. Using the data given in Section 2.8 it is also possible to use these values for determining the deliveries of Archimedian screw pumps having a different number of starts.

Although it might be expected that leakage losses would cause the actual delivery to be less than that calculated by the above method the delivery will, in fact, be higher if the installation is correctly designed and manufactured. In his investigations Muysken (1) arrived at a figure of 10 to 12 % greater delivery, and Horch (2) recorded an increase of as much as 30 %, while several measurements that have been made by the author (5) have yielded results that are between 11 % and 18 % higher than the calculated delivery. The conditions that must be fulfilled if this large increase in capacity is to be achieved are :

1. The theoretical water level in the supply basin corresponding to the height of the filling point must be attained as a minimum value.

2. The calculated value of the gap must not be exceeded.

3. The level of water in the discharge basin must not exceed the level above the spill point corresponding to non-return conditions.

4. The speed must be so high that flooding is just avoided.

If these conditions are met it can safly be assumed that the nominal delivery determined by the combined calculation and graphical method will be exceeded by 15 %. The actual delivery then becomes :

$$\boxed{Q = 1{,}15\ Q_n = 1{,}15\, q \cdot n \cdot D^3.}$$

2,4 Leakage Losses

Leakage losses have an adverse effect on the delivery and the efficiency, and should therefore be accurately determined. Their influence is, however, usually greatly overestimated by the uninitiated observer.

Over the arc E-H shown on the vertical projection of the unit body of water, in Fig.10, the water will leak into the chamber below under the action of the constant differential pressure $h = \frac{S}{a} \cdot \sin\beta$, while over the arcs A-E and D-H the pressure difference will vary between h and O.

The amount of water that leaks through also depends on the flow coefficient μ and the size of the gap s. Experience shows that it is safe to assume $\mu = 1$, and if the velocity coefficient is given the same value, so that the average velocity is $v = \sqrt{2gh}$, the leakage loss becomes

$$Q_l = 2{,}5 \cdot S_{sp} \cdot D \cdot \sqrt{D}$$

and the leakage ratio is

$$\lambda = \frac{Q_l}{Q_n} .$$

In practice, when pumping effluents the leakage losses are often reduced by deposits on the trough which decrease the gap. This to some extent compensates for the almost invevitable variations that occur when the gap is constructed, especially where concrete-lined pipes are concerned. Measurements of the time taken for a screw to empty after it has been stopped and prevented from rotating in the reverse direction confirm this, even though the leakage conditions are slightly more adverse when the screw is stationary than when it is running (owing to the absence of the "entraining" effect of the flowing water). Nevertheless, if the time that an Archimedian screw pump fitted with a reverse-rotation interlock takes to empty after it is stopped is measured, it is possible to draw useful conclusions regarding the size of the gap, and hence of the leakage losses that are likely to occur. This therefore forms a simple means of checking that the gap mension is as specified.

The assumption that, owing to leakage losses, the amount of water in the chambers will reduce towards the discharge end of the pump is not borne out in practice. The amount of water that flows into each chamber f .n the one above is precisely the same as the amount that the chamber in turn loses, by leakage, to the next below. The leakage losses in the uppermost chamber are compensated from the discharge basin, so that there is no reason for any variation in the water level between the uppermost and the lowermost chambers. Instead, there is merely a constant flow of water back along the screw from the discharge basin to the supply basin. The only assumption is that the size of the gap is virtually constant over the entire length of the screw. There is therefore no relationship between the leakage losses of a screw and its length or delivery head.

It should also be borne in mind that the velocity with which the leakage water flows along the screw is usually much less than that which would be calculated on a basis of the pressure drop between two adjacent chambers, since the water being pumped and the leakage water are moving in opposite directions where they are in contact. Effluents having a particular degree of contamination are likely to exhibit exceptionally satisfactory conditions in this respect.

The size of the gap between the screw and the trough will depend on the screw diameter and should not exceed

$$\boxed{S_{sp} = 0,0045 \sqrt{D}}$$

if the leakage losses are to be maintained within the limits predicted by the leakage loss calculation. It is assumed that both the screw and the trough are weälll made, and this can undoubtedly be achieved in practice. The leakage losses will then vary D and β, and will lie between 3 and 12 % of the delivery, the higher values relating to a screw of small diameter installed at a steep angle. If the leakage gap is increased to allow for the deflection of the screw where long screws are concerned, or due to poor workmanship, the leakage losses will vary within certain limits in proportion to the changed gap widthS $_{sp}$.

A manufacturer who supplies a screw alone can therefore never guarantee a certain delivery or efficiency without at the same time specifying an upper limit for the gap width. The accuracy with which the trough is made will thus have a considerable effect in determining the capacity and efficiency of an Archimedian screw pump.

2.5 External diameter of Screw

If, for similar screw forms, the screw plane in a radial direction is perpendicular to the axis, the volume of a unit body of water, and hence the delivery of the screw, will be proportional to the cube of the external diamter of the screw. This is shown by the formula for the delivery of a scfew that was given above, namely $Q_n = q . n . D^3$.

Re-arranging this in terms of the diameter, which is the required factor, we obtain :

$$\boxed{D = \sqrt[3]{\frac{Q_n}{q . n}}}$$

The variable is initially only q, which varies with the angle of installation β, the number of starts a , and the ratio of the external diameter of the screw to its shaft diameter if the ratio δ of the pitch S to the external diameter D is assumed constant (S=D). The ratio $\frac{S}{D}$ = 1 has proved to be the correct choice - in particular for three-start screws and values up to 35°. If the screw is installed at a steeper angle it seems desirable to reduce the pitch ratio so as to obtain a fuller chamber. This is especially true for screw pumps having a lesser number of starts than a = 3.

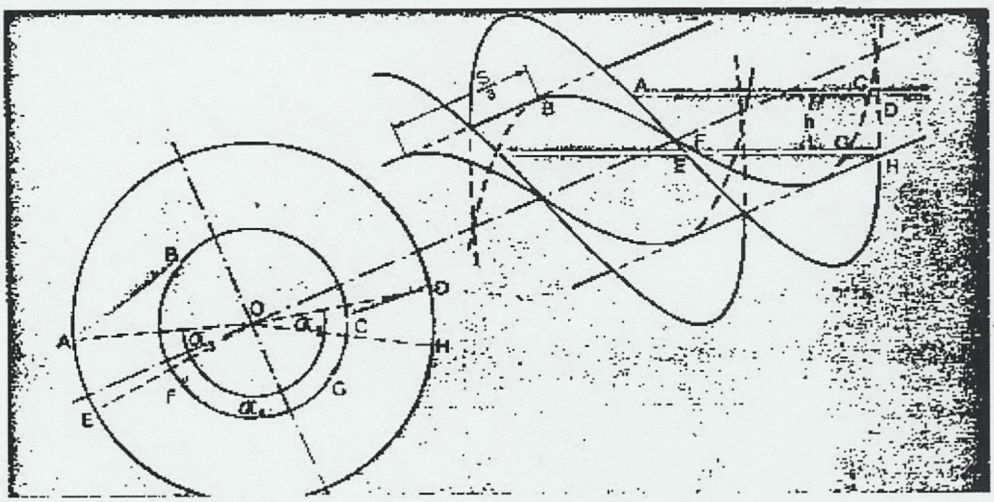
In the RITZ Performance Table, which is reproduced as Table 5 (pp. 111 to 119), the screw diameters have values which are roundddd to the nearest 100 mm, less twice the gap width S_{sp}, so that the nominal size recorded in fact represents the internal diameter of the trough or pipe. In the Type Number the figure 11 indicates, in general, that the item concerned is an Archimedian screw pump, and the figures that follow the 11 are the rounded-off nominal diameter in decimetres. A total of seventeen different diameters, ranging between 400 mm and 4500 mm, were selected when establishing this range of types. The standard range thus covers deliveries extending at the optimum speed from 13.5 litres/s (two-start screw of diameter 394 mm at an installation angle $\beta = 40°$) to 7640 litres/s (three-start screw of external diameter 4481 mm at $\beta = 30°$). No differences were made in the values of the diameters D and d for a given type in order to enable the main components, such as the screw blades and the shaft tube, and also the trough, to be standardised as far as possible. This simplification has a beneficial effect on the manufacturing costs of screws. The possible reduction in delivery that can be achieved by reducing the speed of rotation is discussed in detail in Section 2.10. Speed variations of this type are allowed for in Table 5. It is, however, not possible to increase the deliveries above the optimum values for n quoted in the Table, since the resultant flooding will reduce the unit body of water held between the screw blades, and hence the delivery.

2.6 Dimensions of Shaft Tube

The shaft tube is an important component of an Archimedian screw pump, since it has to perform both static and mechanical functions and also serves a hydraulic purpose.

From the mechanical point of view it is the element to which the blades are attached, while its static properties, which depend on its diameter and wall thickness, govern the deflection and, with this limited to an acceptable value the length of the screw is determined and, hence, the head that a single-stage unit can cover. From the hydraulic point of view the shaft tube is one of the factors that determines the unit volume of water that can be trapped between adjacent blades. Finally, at the point where the lower blade is attached, the head of the shaft tube prevents the water flowing back along the shaft tube to the chamber below. On the other hand the shaft tube occupies some of the volume of the chamber enclosed between two adjacent blades. It is therefore necessary to determine the ratio of the shaft tube diameter to the external diameter of the screw which will require the least amount of material and which will produce an optimum filling of the chamber and, hence, an optimum delivery.

The ratio of the shaft tube diameter d to the external diameter of the screw D (which is derived from theoretical investigations into the filling of the chamber between ad-



10 Vertical projection of the screw and representation of the water surfaces in a chamber enclosed by screw blades

jacent threads) leads to a much larger shaft tube diameter than was adopted in practice in earlier plants.

Theoretical investigations using the graphical method (see Fig. 9) and practical measurements on full-size plants and during the course of model tests have shown that maximum filling and, hence, optimum deliveries are obtained only if the ratio of the shaft tube diameter to the external diameter of the screw

$$\delta = \frac{d}{D} \approx 0.45 \text{ to } 0.55$$

These ratios also produce the most economic results from the point of view of the optimum utilisation of material. In any event, the value of the ratio δ should be maintained between 0.4 and 0.6 so as to arrive at satisfactory results for the filling, which is one of the factors that determines the efficiency of a screw. Only in the case of screw pumps having a low output may there be some justification for increasing the diameter of the shaft tube, to reduce the permissible deflection, or to achieve the maximum possible shaft length. This will be the case, in particular, where the desired improvement in the static properties cannot be achieved by increasing the wall thickness of the shaft tube, although it must be emphasised that, in general, the wall thickness of the shaft tubes (of the smaller screws in particular) is usually increased to reduce the deflection. Welded steel pipes to DIN 2458 (June 1966 Edition) are normally selected for the shaft tube. This Standard is in complete agreement with ISO Recommendations R 336, apart from a few exceptions as regards supplementary sizes. The material is steel to DIN 1626, Sheets 1 to 4, and consists of Grade St 37.2 steel. A selection of steel pipes for use in the manufacture of shaft tubes is listed in Table 4 on p. 110.

2.7 Ratio of the Pitch to the External Diameter of the Screw

Unfortunately it is not possible to arrive at any simple arithmetical relationships which enable the filling of the chamber between adjacent blades to be determined as a function of the ratio of the pitch to the external diameter of the screw, or which can be differentiated to give a maximum. If further variables such as number of starts, inclination, and the ratio of the shaft tube to the external diameter of the screw, which was discussed above are included, this becomes an impossibility.

The only solution then is to use the combined graphical and arithmetical method, which has already been recommended in connection with the determination of the degree of filling and the calculation of the delivery, and to apply this to a range of screw forms, finding the optimum values on a trial and error basis.

This enables the q-values to be derived. These are essential for calculating the delivery data and, at the same time, make it possible to determine at a glance the value that is an optimum from the purely hydraulic viewpoint.

The values for the ratio

$$\sigma = \frac{S}{D}$$

that have previously been used in practice have ranged between S = 0.8 D and S = 1.2 D. The values that are obtained will then largely depend on the inclination of the screw and the number of starts. A screw that is installed at a steep angle will achieve optimum filling of the chambers if the ratio of pitch to external diameter is reduced; for example, if β exceeds 30°, to S = 0.8 D and less. Since this applies where the number of starts is a = 3 then, for values of a = 2 or a = 1, it would (in practice) be desirable to select a still smaller pitch relative to the external diameter, as shown in the following 3 examples (Fig. 11) of an Archimedian screw pump for which δ = 0,45, $\beta = 30^\circ$, and σ = 1, for the numbers of starts a = 3, 2, and 1.

From these examples it can be seen that, with a two-start screw (even at $\beta = 30^\circ$), there is little wetting of the upper screw blades due to the filling of the chamber, and that the shaft tube does not dip into the water in the chamber at all at this point. Matters become even worse for a single-start screw under the same conditions. More information on the volumes that remain in the chambers is given in Section 2.8 <u>Number of Starts</u>.

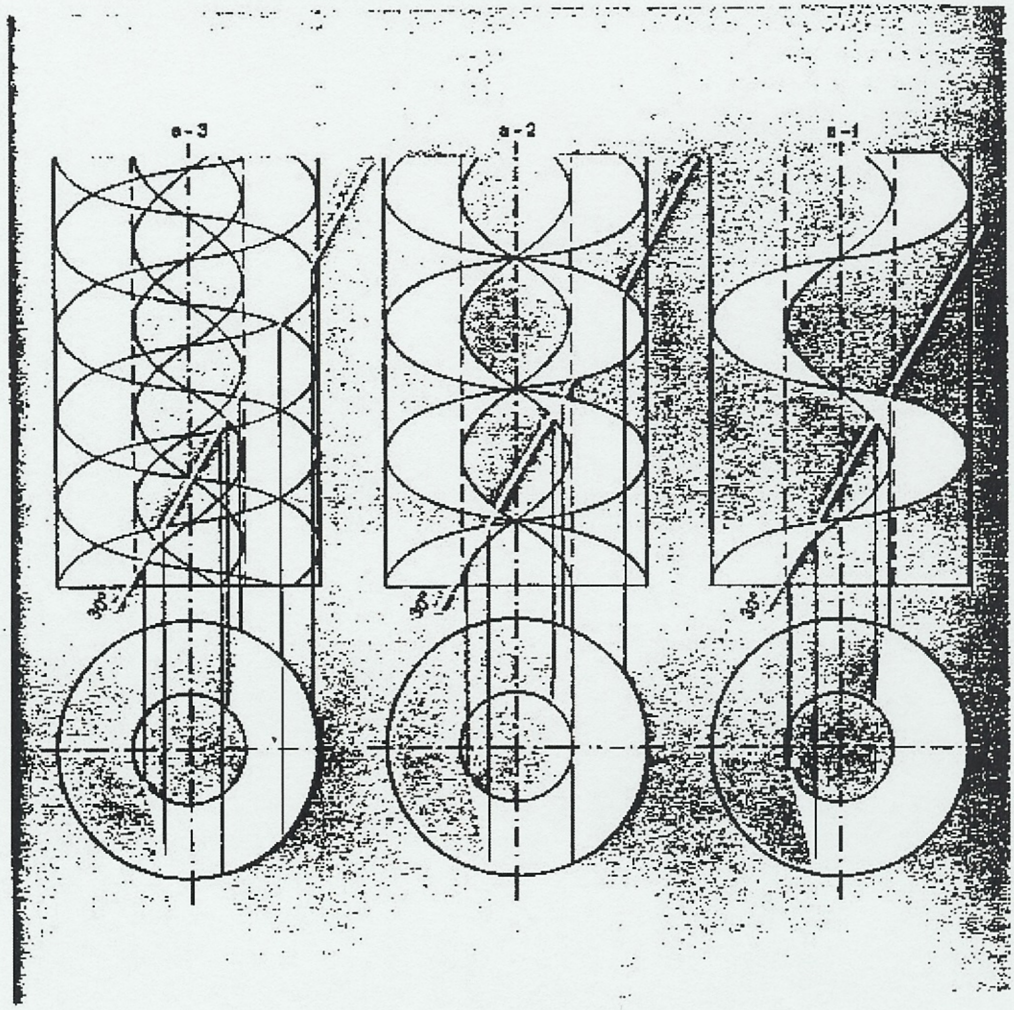
These discussions on the effect of the pitch-ratio lead to the suggestion that it may be desirable to employ a range of values, such as for

$\beta < 30^\circ$ S = 1,2 D

$\beta = 30^\circ$ S = D

$\beta > 30^\circ$ S = 0,8 D

This would enable the best values from the hydraulic point of view to be obtained. In practice this would, however, lead to a very large number of screw forms being



11 Filling of chambers for the same screw form with different numbers of starts

required, with the resultant manufacturing difficulties. The decision was therefore made not to vary the pitch but, instead, to ;ect the optimum value for the angles of inclination likely to occur in practice, namely

$$\sigma = \frac{S}{D} = 1{,}0$$

This decision can be justified in that it is possible to compensate for the reduced delivery by slightly increasing the diameter of the screw, which carries with it only a minor penalty as regards the amount of material required. In addition, the larger pitch reduces the chance of a blockage occurring if solid substances are passed through the pump, and the stronger shaft enables the length of the shaft to be increased.

2.8 Number of Starts of a Pump Screw

In the past Archimedian screw pumps have been made with numbers of starts a varying from 1 to 3, the ee-start screw probably being that most frequently used, If all other factors remain unchanged, the delivery of a given screw for various numbers of starts varies approximately as follows :

$$Q_{a = 1} : Q_{a = 2} : Q_{a = 3} = 0{,}64 : 0{,}8 : 1{,}0.$$

This means that if the conditions are otherwise unchanged the delivery of a single-start screw will be about 64 % that of a three-start screw, and of a two-start screw about 80 % that of a three-start screw. (See also note to Table 3 on p. 110).

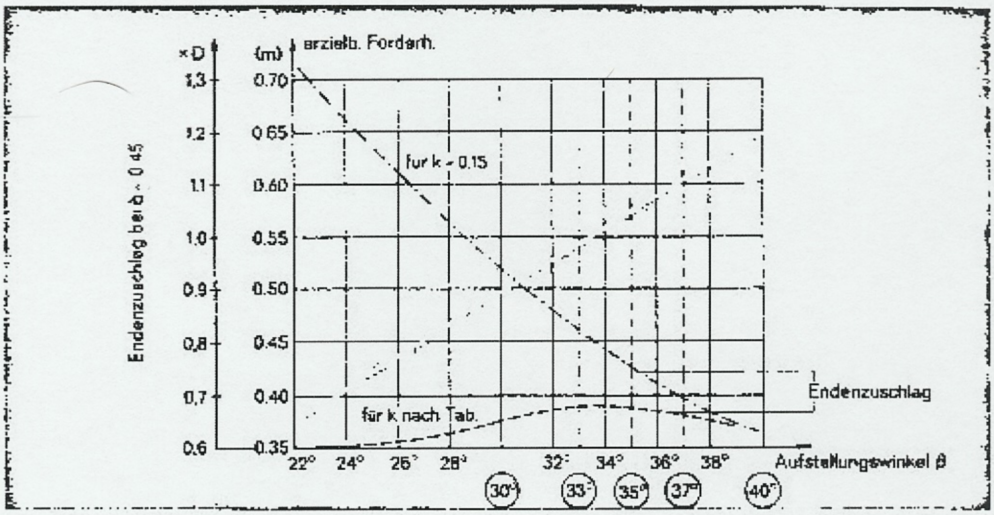
As an approximation it can be concluded that the capacity of an Archimedian screw pump is reduced by about 20 % for each unit reduction in the number of starts. If the weights per metre run, given in Table 6 are compared it will be , that the proportions are much the same, so that there is no inducement to depart from the well-known three-start screw from the point of view of savings in materials. Apart form the smaller amount of space required, due to the smaller external diameter of the screw, the advantage of the three-start screw is that it is largely unaffected by variations in the water levels in the supply and discharge basins. Blade impact, vibration, and heavy mechanical loading of the bearings, gearing, and the power unit can be the results under adverse conditions of reducing the number of starts. The danger of blade impact, especially on the water in the discharge basin, is particularly likely when the screw is installed at a steep angle. It is therefore necessary, to a large extent to dispense with the non-return effect at the discharge basin when using a screw with a small number of starts. The increase in diameter that accompanies a reduction in the number of starts may prove of advantage when the deflection of the screw would otherwise cause the size of gap to be increased to the point where it is no longer possible to cope with a given delivery head. This applies, in particular, where small deliveries and large heads are involved, as shown in the following example. Water is to be raised over a height of 4.0 metres at a rate of 45 litres/s. While it would be possible to achieve this delivery using a Type 11.05 at $\beta = 30^\circ$ and $a = 3$, the total length between the bearings, which is limited to 7.3 metres to keep the deflection to an acceptable value, is insufficient. Instead it is possible to use the Type 11.06, with $\beta = 30^\circ$ but $a = 2$, running at a slightly lower speed; for this angle of inclination it is possible to achieve a total length of up to 9.0 metres with this screw, and this will easily produce the desired head. If it is possible to change the angle of inclination, it is also possible to use the Type 11.06 with $\beta = 35^\circ$ and $a = 2$, in which case the length of the screw can be increased to 9.3 metres so that, with the steeper inclination, it is possible to provide a delivery head of about 5,0 metres. The larger spacing between adjacent blades that results from the reduction in the number of starts makes it less likely for a blockage to occur if solid substances are carried along in the water.

2.9 Angle of Inclination to the Horizontal

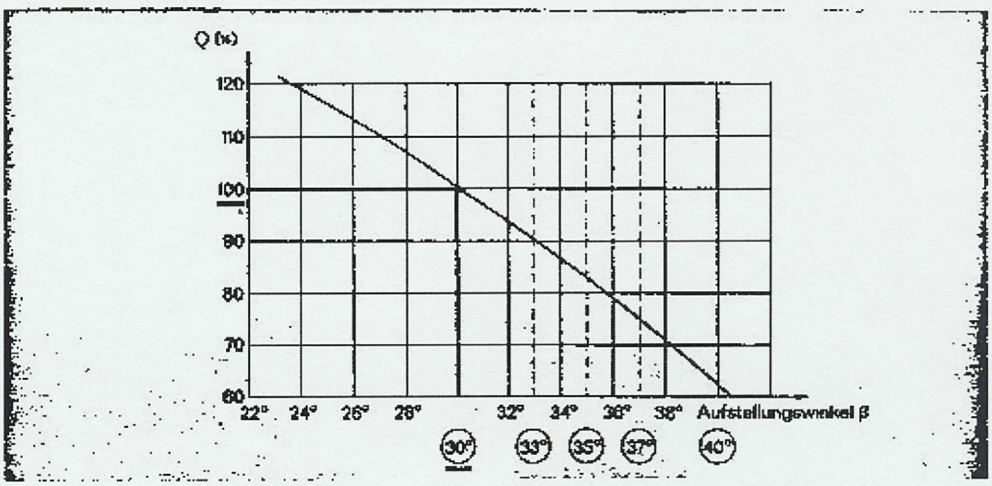
The Dutch literature (1), (2), and (3) deals with angles of inclination ranging from 22 to 30°. Since these publications are concerned solely with the use of the Archimedian screw pump for water drainage in landreclamation schemes where the delivery heads are low, there is no inducement to consider any steeper inclinations. In fact, Muysken (1) states that the optimum efficiency is achieved with a low delivery head when $\beta = 26^\circ$.

When an Archimedian screw pump is used for handling effluents the delivery head is often required to be higher than for reclamation schemes. For this reason it soon became necessary to consider values of β higher than 30°. The approximate delivery head that can be achieved per metre run of screw equipped with blading is shown in Figure 12. In this diagram the reduction in the non-return effect at the discharge basin and the changes in the inflow conditions at the supply basin when the angle of inclination is increased have been assumed to be of secondary importance and have been ignored.

Purely from the standpoint of the amount of material employed it is not economical to increase the angle of inclination β above 30°. As the



Endenzuschlag bei	=Addition due to end effects at ...
Aufstellungswinkel	=Angle of inclination
erzielb. Förderh.	=Delivery head attainable
für k	=for k
für k nach Tab.	=for k from Table
Endenzuschlag	=Addition due to end effects



Aufstellungswinkel — Angle of inclination

12 Delivery head attainable per metre run of screw equipped with blades and corresponding end effects

13 Approximate change in deliveries for changes in angles of inclination

angle of inclination to the horizontal becomes steeper, the capacity of a given screw will drop approximately in accordance with a given law, as is the case when the number of starts is changed. If, for example, the capacity of a screw is taken as 100 % at $\alpha = 30^\circ$, the capacity will decrease or increase as the angle of inclination to the horizontal is made more or less steep in the manner shown in Figure 13.

From this diagram it is possible to deduce that over the range of $\alpha = 22 - 40^\circ$ that if practicable there will be a reduction in capacity of about 3 % for every degree increase in the angle of inclination. The change in capacity is approximately the same for any number of starts of the screw. Variations in σ over the range $\sigma = \frac{d}{D} = 0.4$ to 0.6 also have very little effect on the change in capacity.

As indicated, when discussing the determination of the delivery rate and the effects of the pitch ratio, when the angle of installation is made steeper the ratio of the pitch to the external diameter of the screw and also the ratio of the shaft diameter to the external diameter of the screw should be changed in order to ensure that the extent to which the chamber is filled, and hence the delivery rate, remain about the same. The shaft diameter d should be increased, while the pitch ratio should be reduced. In the interests of standardisation of the screws, and in order to simplify production these variations have not been made, and it appears that this will cause no undue economic penalty in any case arising in practice.

2.10 Speed Selection

Formulae for the determination of the optimum speed of an Archimedian screw pump have been published by Muysken (1), Horch (2), and Bekkering (3) and in Hütte (6). The decision to use the formula derived by Muysken:

$$n = \frac{50}{\sqrt[3]{D^2}}$$

is justified, since the values obtained from this produce the most satisfactory results in practice.

The speed given by this formula is the largest number of revolutions per minute for which, with the chambers between blades filled to an optimum extent, an overflow of water by way of the screw shaft back into the supply basin is just avoided. That the speed increases slightly for the larger diameters is of no importance, since with the smaller screws even a slight overflow represents a higher proportion of the total quantity being delivered than it does with the larger screws where the delivery rate is higher.

The assumption that there is a fixed relationship between the speed n and the external diameter of the screw D is derived from the insertion of the above value for n in the delivery rate formula. $Q_n = q \cdot n \cdot D^3$, or $Q = 1.15 \cdot q \cdot n \cdot D^3$ so that the actual delivery rate for a screw can be expressed as

$$Q = 57.5 \cdot q \cdot D^2 \cdot \sqrt[3]{D}$$

It is thus possible to determine the delivery rate of a screw pump merely from the external diameter of the screw and the delivery factor q, which is dependent on β, ϕ, δ and a (as given in Table 3).

The optimum speed corresponding to a fiven external diameter of the screw which is derived from this can be taken from Fig. 14.

Finally, on the question of speed it should be pointed out that an increase in the speed above the value given by the Muysken formula appears highly undesirable. The angleof inclination does not affect the optimum speed. The same applies to the other variable β, ϕ and a within the limits β = 0.8 to 1.2, δ = 0.4 to 0.6, and a = 1 to 3.

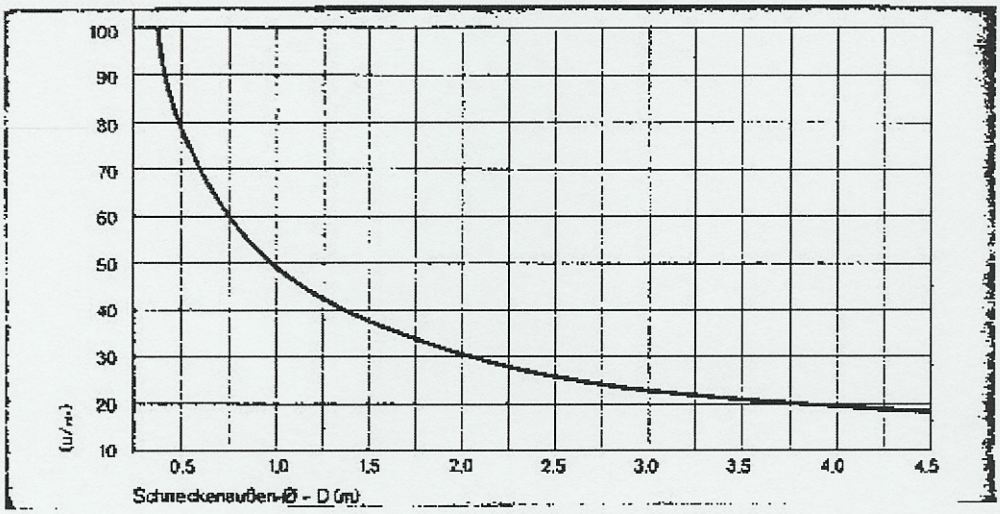
To enable a particular delivery rate to be obtained it may, of course, be necessary to accept variations from the figures for the speed given above, but the speed should never be allowed to exceed

$$n = \frac{50}{\sqrt[3]{D^2}}$$

In any given case the formula for determining the speed then becomes:

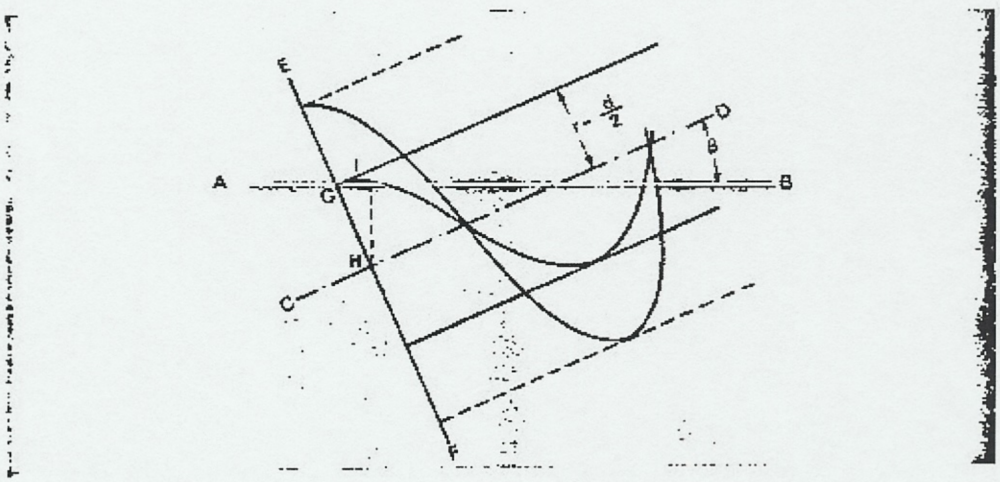
$$n' = \frac{Q'}{\mathcal{E} \cdot q \cdot D^3}$$

The values for q that are given in the Table, for use with a full chamber can be employed in this formula provided that the speed given by the formula, rounded off to the nearest half r.p.m., is not more than 30 % less than the speed recommended by Muysken. Since the value of q and the nominal speed are directly proportional to the delivery rate, the delivery rate of a given screw can be reduced below the nominal delivery rate within certain limits bychanging the speed provided that the correction supplied by the delivery increase ratio $\mathcal{E}$ for full delivery ($\mathcal{E}$ = 1.15) is applied with no undue reduction in efficiency. The limiting value for the speed



U/min =rev/min

Schneckenaussen dia. D =External diameter of screw D



14 Speed curve showing optimum speeds as a function of the external diameter of the screw D.

15 Vertical projection of screw for calculating the necessary depth of immersion at the suction end.

reduction of 40 % should, however, not be exceeded if possible.

As can be seen from the limiting factors that affect the variation of the speed, it is probably not economical to reduce the speed by more than the quoted figure of 40 %, in order to have reserve capacity in hand to allow for future extensions of the plant, etc.; Variations in the speed should be limited to the changes that can be achieved by changing the belt pulleys. In view of the exceptionally good performance of Archimedian screw pumps under part-load conditions it is not worthwhile to go too far with changes of speed, although their use for handling effluent often encourages such measures.

2.11 Effect of Depth of Immersion of Bottom End of Pump

To enable complete filling of the chambers to be achieved, and hence also the optimum delivery rate, it is necessary for the immersion of the screw at the bottom end of the blading to reach at least the filling point G shown on the sketch.

This is the highest level the water level can attain without flowing back along the shaft. If, as shown in Fig. 13, E-F is the bottom end of the screw, a water level A - G would suffice to fill the chambers completely. To achieve the full delivery rate the centre point H of the bottom of the blade-carrying part of the screw must therefore be immersed to a depth of H - I. The distance H-I can be regarded with sufficient accuracy for practical purposes as being equal to

$$\boxed{H - I = r \cdot \cos \beta}$$

although distance G - H is slightly smaller than $r = \frac{d}{2}$. Even with large screws, however, the difference between the value obtained by using this formula and the value caclulated by accurate methods is only a few centimetres, so that it is quite satisfactory to use this formula for determing the water level in the bottom basin.

Although greater immersion will not enable the delivery rate to be increased, a considerable drop in delivery is experienced if the water level in the bottom basin falls below the value.

An example of the variations in delivery rate that result from changes in the depth of immersion for a particular screw (D = 0.6 m, d = 0,26 m, $\beta = 30^\circ$, H-I= 0,112m are shown in Fig. 16.

The relationship between depth of immersion and delivery rate that can be derived from this diagram is approximately true for all other sizes of screw. In the case of effluent handling plants, which are very often operated under part-load conditions, this point is of particular importance. An attempt will therefore be made to find an approximate relationship between the depth for full immersion :

$$H - i + \frac{D}{2} \cdot \cos\beta = \left(\frac{d+D}{2}\right) \cos\beta$$

and the delivery rate corresponding to the water level concerned.

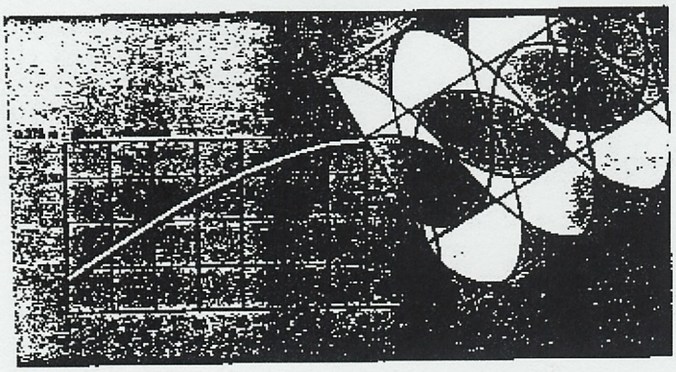
When the water level reaches the initial immersion or contact point T, the delivery rate will be zero, since the very slight immersion of the outer edges of the blades will do no more than make up the leakage losses. If the depth for full immersion and the maximum delivery rate are both taken as 100 %, the diagram shown in Fig. 17 will be obtained, which shows approximately the relationship between immersion and delivery for any screw.

As a rule of thumb it can be assumed that over the upper 30 % of the depth for full immersion (corresponding to full delivery) each 10 % drop in the depth of immersion $\left(\frac{D+d}{2}\right) \cos\beta$ above the contact point is associated with a 20 % Q_{max} drop in delivery. If the depth of immersion is 50 % the delivery will only be about 25 % of the maximum value, and the delivery then drops r pidly to the point where the leakage losses only are covered, and so there will be zero net delivery. These approximate figures can be used for deciding on the economical fluctuation range for the water level, the supply basin dimensions, and the frequency of cycling of the pump water. These factors will then enable the water levels in the supply basin at which the pump should be started and stopped to be determined.

2.12 Discharge of Water at the Top of the Pump

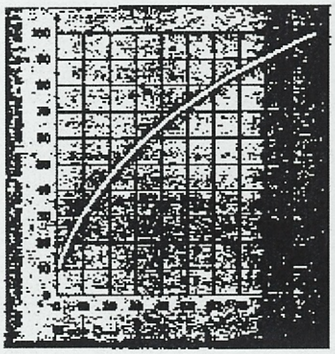
At the top end of the pump, both the trough and the screw blades are usually cut-off along a plane at right-angles to the screw axis. There is no point in continuing the blading on the screw beyond the end of the trough, with the intention of holding back the water in the discharge basin, since a sealing effect will be achieved only where the blades and trough are in contact, allowance being made for the necessary gap between these two components.

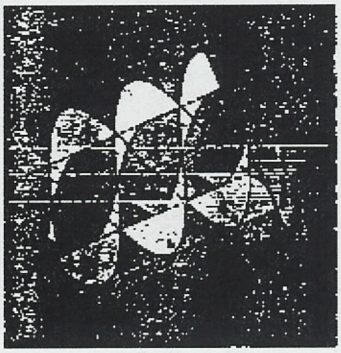
The position of the discarge of spill point S relative to the maximum water level in the discharge basin should be selected so that it is impossible for the water in the discharge basin to flow back down the screw. Looking at Fig. 18, the screw is drawn in a position where the level in the chamber between blades a and b corresgünst. Anfluthöhe = satisfactory depth of immersion



% der Gesamtanfluthöhe ... = % of depth for full immersion above contact point = ...

% von Q_{max} = % of Q_{max}





16 Example of variation in delivery rate with reduction in depth of immersion

17 Relationship between depth of immersion and delivery rate

18 Projection of screw for determination of optimum level in discharge basin

ponds with the level A-B in the discharge basin. A short time before this position was reached the blade a was assisting in holding back the water in the discharge basin.

At that moment the level of the water in the chamber between blades a and b was lower than the level in the discharge basin. If the two had been in connection, the water would have been able to flow away down the shaft of the screw at point G. When the shaft has turned into the position illustrated, however, blade b is able to seal the reverse flow, and the water can flow past blade a, along the straight line E-F.

Storm, Buyzing (7), Horch, and Muysken have carried out experiments which have enabled them to determine, for a number of screw forms, the length of K-S (in Fig. 18) as a function of the external diamter D of the screw needed for the water level to be such that it just does not flow back. When the pump is working the water level in the chambers will, however, be higher than the nominal level. In addition, the dynamic action of the ends of the blades makes it possible to work against a higher water level. At the same time, if the dynamic effect is utilised too extensively there will be a considerable drop in efficiency together with very uneven running of the screw, since the water in the discharge basin has then to be forced back by the impact of each blade. This is particularly true where screws installed at a steep angle are concerned, since the end of the blade then impacts the water in the basin at a very flat angle.

While it may be justifiable to take the fullest possible advantage of the level of water in the discharge basin in the case of land-drainage pumps, where the delivery heads are comparatively low, in sewage pumping plants, where the heads are in general higher, it is possible to allow a little more freedom in this respect. It is also advisable to work a little bwlow the limiting values with sewage pumping plants, since the screws are usually installed at a steeper angle. Under these circumstances it is recommended that the values quoted below are adopted as the minimum values for K-S as a function of the angle of inclination :

β	22°	26°	30°	33°	35°	37°	40°
K - S	0,45 D	0,38 D	0,30 D	0,26 D	0,22 D	0,19 D	0,15 D

k-values for determining permissible height of water above the spill point

The values quoted apply to three-start screws and will prove adequate in all cases encountered in practice. It is only if several screws are working in parallel and are discharging into a common basin that it will be necessary to make specific calculations.

In general, and especially in the case of screws where the number of starts a is less than 3, it is advisable to adopt the value K-S = o,15 D. This limiting value should also not be exceeded if the screws run at a lower speed, since as the speed is reduced the ability of the screw blades to retain water also drops.

2.13 Lift of an Archimedian screw pump

Reference is made to Fig. 6 on p.17, under the heading "Comparison with Centrifugal Pumps" in Part 1 of the book. According to this the term "effective delivery head" for an Archimedian screw pump[illegible]tes to the "physical differences in heights of the surfaces of the water in the supply basin and the water in the discharge basin" when the pump is running at full capacity. From the discussions under the headings "water level in supply basin" and "Water level in discharge basin", a relationship is sought between the water level in the supply basin, which is located at H-I = r cos β , above the centre of the bottom end of the screw where the blading commences, and the water level in the discharge basin which is located at a height K-S = 0.15 D above the spill point.

Details of this should be studied in the section on the length of the bladed portion of the Archimedian screw, which follows, since there is a direct relationship between the delivery head and the length of a screw. In the calculations for the determination of the efficiency and the power requirements this geodetic difference in the heights of the water-levels in supply and discharge basins should be inserted as the actual delivery head. For a comparison of the delivery heads of centrifugal pumps and Archimedian screws, particular reference is made to Fig. 7 (on p.17) in the introductory section.

2.14 Determination of bladed length of screw

Fig. 19, which is also relevant to the preceding section on Lift, is intended to illustrate once more the relationships between the water levels in the supply and discharge basins and the delivery head, on the one hand, and the length of screw that is equipped with blading on the other hand.

The bladed length of an Archimedian screw pump can be determined simply from the information given in Fig. 19, while the necessary length of the shaft tube will depend on the design and constructional features of the pumping installation.

So far as the water level in the supply basin is concerned, it has already been stated that, to obtain optimum filling of the chamber formed between the blades, the following approximate relationship (which is, however, sufficiently accurate for all practical purposes) should be fulfilled, namely, that the water level should be a distance

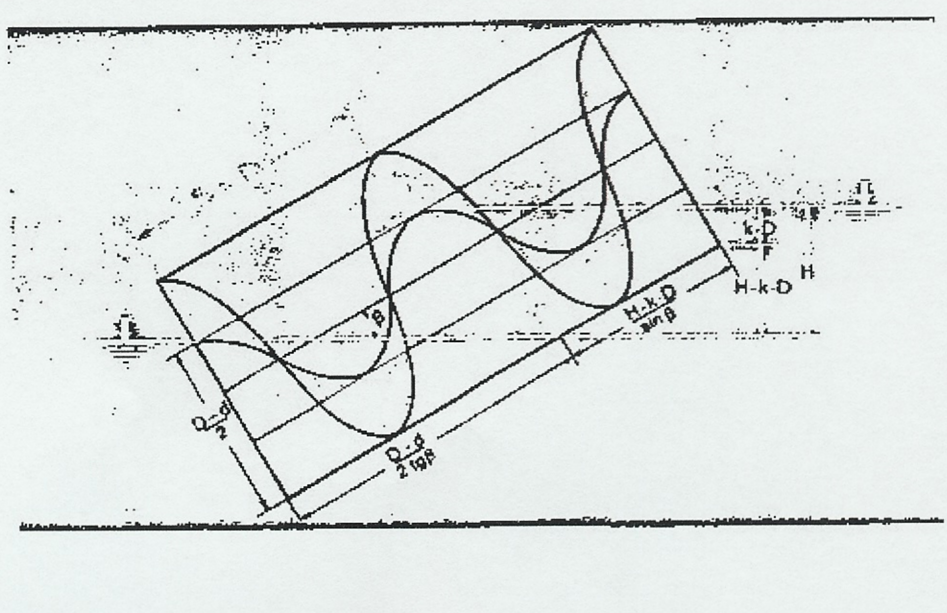
$$H - I = r \cdot \cos \beta$$

above the centre of that surface limiting the bottom end of the screw, while, to prevent the water in the discharge basin flowing back down the screw, and to avoid heavy impact of the screw blades on the water surface, the level in the discharge basing should be K - S = k. D above the spill point. The values for $k = \frac{K-S}{D}$ for various angles of installation are shown in the Table on page 42.

The effect of $\delta' = \frac{d}{D}$ should also be taken into account when deciding the design of the screw since, as δ' increases, the value of k can also be increased slightly. The effect of any variations in the pitch ratio $\sigma = \frac{S}{D}$ have not been considered, since the entire discussion is based on $\sigma = 1$.
With a smaller number of starts, i.e., a less than 3, the value of K-S should not exceed 0,15 D, i.e. k = 0,15. The length of the screw to be equipped with blades then becomes:

$$\boxed{L = \left(\frac{H}{\sin\beta} + \frac{1+\delta}{2\,tg\,\beta} - \frac{k}{\sin\beta}\right) D.}$$

19 Diagram for determining the length of screw to be equipped with blades from the delivery head



The formula $Q_n = q.n.D^3$ and the method of calculating L given above now enable us to calculate the diamter and length of an Archimedian screw pump for a given delivery rate and head at its maximum permissible speed n, the only limiting condition being that the gap should be $s_{sp} = 0.0045 \sqrt{D}$. Mention should be made of the possibility of the length of an Archimedian screw varying, due to changes in temperature (sunshine or low temperatures). Further information on this point will be found in Chapters 3/3.2 and 3/3.14.

2.15 Determination of Efficiency

Until quite recently only Muysken (1) and the author (5) (the latter in the paper "Basis of Calculation for Archimedian Screw Pumps", have presented complete data for the calculation of the efficiency, taking the various factors that affect this into account. The matter will not be discussed in detail here, but for completeness and to assist in an understanding of the problem the factors that influence the efficiency, will be listed :

a) Mechanical losses — in the bearings and the blade linings due to friction, in power transmission, and speed control.

b) Hydraulic losses — due to friction on the blades, due to friction on the shaft due to friction on the rider tube, due to impact at the start and end of the screw.

c) Leakage losses — due to water that flows back and the power needed to raise this water again.

d) Discharge losses — Kinetic energy in the water that is discharged.

Friction in the bearings and of the blade linings can be kept to a relatively low level by careful selection of the components and the manufacturing methods, and by good lubrication. The most important factor is to prevent dirt penetrating into the bottom bearing and to exclude any materials that may get caught up and twisted around it.

The losses in the power transmission, which basically consists only of a few sets of spur gearing for reducing the speed, and possibly a pair of bevel gears to change the direction of the drive, are also not of major importance, even though their unavoidable presence is a disadvantage compared with centrifugal pumps. The usual method of connecting the power unit to the gearing by means of V-belts has a beneficial effect both on the transmission elements and the prime mover since it tends to absorb any shocks that may arise during starting. The overall mechanical efficiency η_m for a screw of modern design can, in practice, be taken as

$$\boxed{\eta_m \sim 0{,}9 \text{ to } 9{,}95}$$

depending on the nature of the bearings and the reduction gearing.

The effects of the friction of the fluid over the trough, the blades, and the shaft tube of a screw (or, in other words, the hydraulic losses) are of decisive importance in accounting for the differences in the power supplied and the useful power output (or, the effeciency of the unit). The Archimedian screw pump differs form the centrifugal pump in that there is no transformation of kinetic energy into pressure energy. In the case of the screw the water moves in a very regular manner, and follows complete paths with no reversal of motion. The fluid resistance that occurs when the wetted surfaces dip into and emerge from the fluid represents a frictional loss which can be calculated with accuracy. It increases with the square of the speed, and is proportional to the wetted area. The frictional coefficient to be employed is a function of the Reynolds number. The contact between the water and the components of the screw and the resultant friction causes water to be carried over with the blades, especially at high speeds, and this is one of the reasons why the delivery rate can exceed the theoretical value. Approximately half the total resistance is accounted for by actual friction, and half by volume of water within the screw blades. The effect of the screw shaft on the hydraulic losses is comparatively small, especially since this acts as a helix and with a relatively small lever arm.

Also, part of these hydraulic losses are those losses due to the shock that occurs when one of the lower ends of the screw blades dips into the water, and the corresponding effect at the top end of the screw. If the water levels are correct these losses are small, since the immersion at the lower end and the emergence at the upper end can be practically shock-free. Unfortunately, under practical service conditions, it is possible to arrange for the correct values to obtain only in the rarest cases or for brief periods, especially insofar as the water level in the supply basin is concerned.

At the upper end of the screw heavy impacts, associated with high hydraulic losses, will occur if the water level in the discharge basin is such that it is trying to flow back and has to be repulsed by the upper ends of each blade. It is not possible to calculate the magnitude of this loss accurately. It is, however, comparatively easy to avoid it, or at least to keep it within reasonable limits by exercising care in maintaining the water in the discharge basin at the correct level relative to the spill point. Only if this is done will the maximum efficiency attainable be achieved.

The hydraulic efficiency thus includes all the loss factors in the power term N_{vh}, which represents the power required to overcome the hydraulic resistances. The ratio of this power term to the useful work performed by the screw, $N_n = \frac{1000 \cdot Q \cdot H}{75}$ can be expressed in shortened form as :

$$\frac{N_{Vh}}{N_n} = e \cdot n^2 \cdot D \cdot \frac{L}{H}$$

where e is a friction loss factor

$$e = \frac{4{,}5}{60 \cdot q} \left(\frac{a \cdot w_t}{\sigma} + \frac{1{,}94 \cdot \sigma^3}{\varepsilon \cdot 10^8} \right)$$

and, in this expression, w_t can be determined as follows (if the wetted portion of the screw is assumed to be $u \cdot \pi \cdot d \cdot \frac{S}{a}$, and u is taken approximately as $u = \frac{\varphi_6 + \varphi_7}{z \cdot 360}$:

$$w_t = 0{,}4 \; \frac{\pi \cdot \sqrt{\pi^2 \cdot d^2 + S^2}}{3600} \cdot u \cdot n^2 \cdot \frac{S}{a}$$

The work that is required to again raise the water that has run back can easily be determined :

$$N_{Vl} = \frac{Q_1 \cdot H}{75}$$

Since, when comparing $\frac{N_{Vl}}{N_n}$ the delivery head value also appears in the denominator, this cancels out so that, as has already been stated when discussing the leakage losses, the delivery head has no effect on the leakage loss.

All types of pumping machinery are subject to a discharge loss, since the medium being pumped has a cdrtain velocity when it leaves the pump. While the magnitude of this loss can be reduced by suitable design of the outlet (in the case of a centrifugal pump) the water that is discharged from the screw pump is in a highly turbulent condition, so that there is no possibility of recovering kinetic energy. On the other hand, the relationship

$$\frac{N_{Va}}{N_n} = 14{,}1 \cdot \frac{S^2 \cdot n^2}{H} \cdot 10^{-6},$$

shows that the discharge losses decrease relatively as the delivery head increases. Looking at this point in isolation, a screw pump will exhibit a higher efficiency as the delivery head increases.

If all these various losses are added arithmetically, the power required on the screw shaft becomes:

$$N_s = m \, . \, N_A = N_n + N_{v_h} + N_{n_l} + N_{v_a} \quad :$$

and

$$\frac{N_n}{N_A} = \text{ges} \quad \text{and} \quad \frac{N_n}{N_S} = \frac{\text{ges}}{m}$$

so that

$$\frac{m}{\text{ges}} = \frac{N_s}{N_a} = 1 + \frac{N_{v_h}}{N_n} + \frac{N_{n_l}}{N_o} + \frac{N_{v_a}}{N_n} \quad .$$

With this formula we are in a position to determine the efficiency of a screw from the anticipated loss factors, by inserting the individual values of these loss factors into the efficiency formular as follows:

$$\frac{m}{\text{ges}} \simeq 1 + e \, . \, n^2 \, . \, D \, \frac{L}{H} + \frac{l_2}{. \, n \, . \, D} + 14{,}1 \, . \, \frac{S^2 \, . \, n^2}{H} \, . \, 10^{-6} \, .$$

Using this formular the efficiency can then be calculated by inserting the dimensions of the screw and the appropriate valued of e (= friction loss factor) and l_2 (= leakage loss factor).

One must realise, however, that the result of this calculation is based on a number of assumptions which may not be correct for an individual case in practice. Mention may be made in particular of the idealised inflow conditions. If the immersion of the bottom end of the screw is too low, the chambers will not fill to the optimum extent while, if the specified value for H - I is exceeded, eddying and impact losses will occur, and energy will be dissipated. Similar conditions prevail at the top end of the screw, excepting that there is easy to maintain satisfactory conditions at the expense of reducing the effective delivery head by keeping the immersion of the spill point, K-S, to a low value.

The gap which has been assumed to have the value $s_{sp} = 0.0045\sqrt{D}$. If the actual gap width differs from this value, a suitable correction will have to be made and this can be done relatively simply. Even a comparatively small increase in the gap width s_{sp} will, however, exert a major influence on the efficiency of a screw pump.

The efficiency that is calculated on the basis of the equation given should usually be regarded as the highest efficiency of which an Archimedian screw pump is capable. There are a number of reasons why the efficiency may drop below this value, while a higher efficiency is unlikely.[1] The formula can never be differentiated to yield an expression which will indicate the circumstances under which the efficiency will become a maximum. This is because of the many coefficients that have been incorporated and which have only a specific relationship to the shape of the screw. In any particular case it is, however, possible to determine the relative efficiency for various forms of screw with very little effort by carrying out check calculations, and so to arrive at a maximum value.

Despite the limitations on the equation for determining the efficiency of those Archimedian screw pumps which have been mentioned, the results obtained in a number of practical tests have shown that the formula provides efficiency figures to a perfectly acceptable accuracy. It seems highly likely that it can predict the efficiency to within the accuracy of the tolerance formula normally used in regulations governing centrifugal pumps, namely $\Delta\eta_{zu\ 1} = \frac{1}{45}(1 - \eta_{gar})$. This is true, however, only for the circumstances under which, as indicated above, the efficiency is likely to be a maximum.

If the delivery rate is to be regulated by varying the speed, so as to enable a predetermined delivery to be obtained, it is possible only to reduce the speed below the maximum value given by the above formula. For the change in delivery rate the new quantity of the reduced speed will be given by the ratio $\frac{Q}{Q'} = \frac{n}{n}$, corrected by the delivery increase ratio ε (= 1,15) or ε', where

$$\varepsilon' = \varepsilon \cdot \frac{Q'}{Q} \text{. Then}$$

$$Q' = Q \cdot \frac{\varepsilon'}{\varepsilon}$$

and, from the formula for the delivery rate, the corresponding speed is given by

$$\boxed{n' = \frac{Q'}{\varepsilon \cdot q \cdot D^3} .}$$

So far as the efficiency is concerned the effect of the new speed is that the two factors for friction and discharge losses are reduced, but the leakage loss increases in proportion. If these modifications are inserted in the efficiency formula we obtain :

$$\frac{\eta_m}{\eta_{ges}} = 1 + e \cdot n^2 \cdot D \cdot \frac{L}{H} \cdot \left(\frac{n'}{n}\right)^2 + \frac{\overset{1}{2}}{\varepsilon' \cdot n \cdot D} + 14,1 \, \frac{S^2 \left(\frac{n'}{n}\right)^2 \cdot 10^{-6}}{H} \, .$$

For speed variations of up to 30 % below the speed corresponding to optimum conditions the drop in efficiency that results will be within acceptable limits.

Summarising this discussion of the efficiency of Archimedian screw pumps it may be concluded that, for a small screw, the increase in leakage loss in largely compensated for by the reduction in the discharge losses. This finding may be regarded as a warning against running large screws at too high a speed. For a low delivery head, the friction losses and the discharge losses have the largest adverse effect on the efficiency of a screw, this effect being particularly marked for large-diameter screws.

There is a considerable scatter in the values of the individual loss factors for various screw forms, but these differences largely cancel each other out, so that the total loss remains sensibly constant, and the efficiencies obtained with various screw forms differ by only a few percent.

There is also no reason why any appreciably higher efficiency should be expected from a large screw including its gearing than for a screw of smaller dimensions. The use of the value η_m = 0,9 includes a certain allowance for constructional tolerances, and appears fully justified for practical use.

It may further be concluded, therefore, that there is no significant difference in the efficiencies of large and of small screws.

To enable a comparison to be made with the efficiencies of centrifugal pumps, the normal pump efficiency η_p must be compared with the overall efficiency η_{ges} = $\eta_m \, \eta_s$. Here, η_m takes account of all the mechanical losses in the reduction gearing and the belt drive.

Both values are inserted in the same way into the formula for determining the power required. In other words, the power required at the shaft of a centrifugal pump must be compared with the power required at the input flange of the gearbox since the speed reduction, which is essential for the Archimedian screw pump, is not required with a centrifugal pump, to the advantage of the latter.

2.16 Calculation of the Power required to Drive the Pump

If the actual delivery rate required of an Archimedian screw pump and the delivery height are known, the power required at the input flange to the gearbox will be given by

$$N_A = \frac{1.1 \cdot Q \cdot 1000 \cdot H}{75 \cdot \eta_{ges}} \quad \text{(h.p.)}$$

In view of the fact that relatively large masses have to be brought into motion when the motor is started it seems advisable to include the factor 1.1; this will provide for a 10 % reserve of power, which should be adequate. Starting is eased by the fact that the screw is initially empty, and the full power required to raise the water is not needed until a relatively long time after the start. In the majority of cases it is not possible to keep the current consumption absolutely steady when driving an Archimedian screw pump. This can be due both to the cyclic immersion of the bottom ends of the blades into the water in the supply basin, and also the impact of the top ends of the blades on the water surface in the discharge basin, especially when the blade form is such that these two coincide. This becomes of particular importance in the case of screws with a small number of starts and a low delivery height. In many cases, however, a fluctuating ammeter reading can also indicate that the screw is not aligned correctly or that the sag is too much (due to the excessive length of the screw). The investigations reported in Chapter 2.17 have therefore been made into the acceptable length of shaft tube between the bearings.

2.17 Strength Calculations for the Pump Screw

Because of its simplicity and robustness the Archimedian screw pump is becoming increasingly popular for pumping effluents. In view of the delivery heads that are involved when pumping effluents there is, however, a natural tendency to use screws of undue length.

If the length of the screw is increased to the extent where the sag caused by the weight of the screw itself, acting in the vertical direction together with the weight of the water in the chambers between the blades becomes larger than the gap width, the tips of the blades will scrape against the trough. Even though the current methods of construction, for the most part, result in the trough being made of concrete with the cement-lining moulded to shape by the screw itself (so that some allowance is made for the sag) the sag f must not exceed the gap width s_{sp} if the stresses on the material are not to reach undue values. The tendency to make the troughs of prefabricated concrete sections, which are simple to instal, makes it necessary to restrict the sag of the screw, as does the use of steel troughs.

The shaft tube is for all practical purposes the only load-carrying element in an Archimedian screw. Its load-carrying cross-section is shown in Fig. 20.

The factors to be employed in the strength calculations are defined by the dimensions d_i and D_a, where d_i is the internal diameter of the shaft tube and D_a is its external diameter. The static strength calculation can be simplified by regarding the shaft tube as beam freely supported on two supports, assuming that the load is uniformly distributed over the full length between the supports which are located distance l apart (Fig. 21). The deflection then becomes :

$$f = \frac{P \cdot 5 \cdot l^3}{E \cdot J \cdot 384}$$

And, for the condition that f is not greater than s_{sp}, we obtain

$$P_{zul} = \frac{f \cdot E \cdot J \cdot 384}{l^3 \cdot 5}$$

where P is the permissible load. The weights of the screws (in kg/metre run) and their water contents (in litres or kg/metre run of screw) can be obtained for this calculation from Table 6 (pp. 12 128), and the weigths per metre run of the readily available steel tubes of various wall-thicknesses and diameters, together with their Section Modulus and Moment of Inertia are given in Table 4 (p. 110). The values given are taken by courtesy of the Deutscher Normenausschuß, Berlin, form DIN Standard 2458.

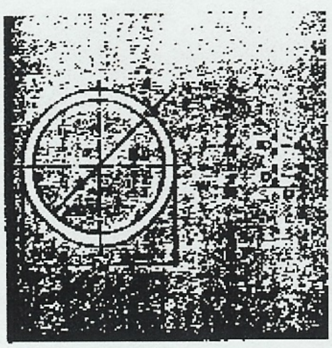
To simplify matters the permissible loadings of steel tubes (as a function of the support spacing) are given in the loading diagram (Diagram 1) in the collection of Diagrams starting on p. 130, for the particularly critical region up to 509.0 mm external diameter of the shaft tube and for wall thicknesses of 7.1 and 12.5 mm, it being assumed that the permissible sag $f = s_{sp}$. If the shaft tube diameter exceeds 500 mm it will be possible to achieve screws of length up to 10 metres between the bearings without any difficulty. It is, however, strongly advised that the sag be calculated for any particular case where even longer screws are required for this diameter. The screw weights and the water contents per metre run can then be taken from Table 6, as indicated above. The bouyancy of those parts of the screw that are immersed in the water is neglected in order to provide a factor of safety.

When estimating the deflection of the shaft tube it is most important not to assume that the blading will act as a stiffener. In addition the calculation of the static strength, in the manner shown above, assumes that the shaft tube consists of a single length, and that the welding and the material properties also conform to DIN 2458. If it proves essential to join several lengths of tube together, avoid locating a welded seam at or near $\frac{L}{2}$, which is the statically critical point. Special investigations will have to be made into the welded seams in such a case.

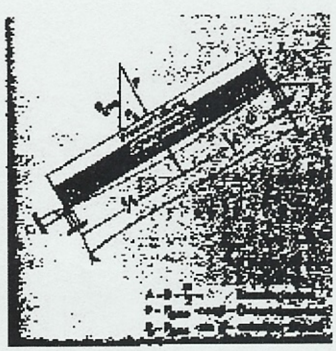
To eliminate the stresses produced in the screw shaft tube as a result of the welding it may be necessary to anneal the finished screw in order to avoid the danger of brittle fracture. This process is advisable particularly where extremely long screws of small diameter and with fabricated shaft tubes are concerned.

Einzellagerlast = Load on each support
Gesamtlagerlast = Total load on supports
achsialer Schub = Axial thrust

20 Cross-section through shaft tube and data for calculations



21 Loading assumptions for shaft deflection



CONSTRUCTION AND OPERATION

3.1 Conditions for the Use of Archimedian Screw Pumps

Some of the points concerned have already been discussed in Chapter 1.2, where a comparison was made between the operational and delivery characteristics of screw pumps and centrifugal pumps. Here, we shall consider the basic conditions which must be met if screw pumps are to be used.

1. It must be possible to accommodate the structure required to house the screw pump, which is usually of considerable length,
2. It must be possible to provide the required delivery head in at most two stages,
3. It must be possible for the water to flow away freely from the discharge basin.

The size of the structure required is discussed further below in the section on "General Layout of Pumping Plants". As an approximation it may be assumed that the overall length of the structure, including supply basin and machinery recesses, will be about 2,5 times the delivery head for a single-stage pump arranged completely underground. In the case of a two-stage pumping installation the length of the structure will increase to 3 or 3,5 times the delivery head. An approximate figure for the total width of the structure is 1,5 times the sum of the external diam ers of the screws employed.

If shortage of space makes it necessary to construct an intermediate pumping station under the roadway itself, it is e ential that the necessity to make provision for access for occasional inspection and m atenance is not overlooked.

One of the main factors that determines whether a screw pump can be employed is the delivery head that is required. Even if the screw is installed at an inclination greater than $= 30^{\circ}$, the overall length of the screw will still in practice be about twice the head. Because, with small capacity screws in particular, the small shaft diameter causes the length of the screw to be limited to about 8 metres (for reasons of static strength), the delivery head for single-stage installations using small screws of up to D = 700 mm will have to be restricted to 4,0 metres, while for larger diameters it can rise to about 6, or a maximum of 8 metres.

Another requirement is that the water that has been raised by the pump should flow away under gravity from the discharge basin, since it is quite impossible to produce a higher pressure at the outlet of an Archimedian screw pump than that corresponding to a discharge height of about 0,15 D above the spill point. In the majority of cases it is therefore not possible to use a screw pump in conjunction with an effluent siphon system, unless communicating pipes are adopted and some other means of flushing is available. The screw pump is also not suitable the discharge pipe requires a higher head than that produced by gravity, due to the height of the discharge point, i.e. if the discharge pipe has to be pressurised.

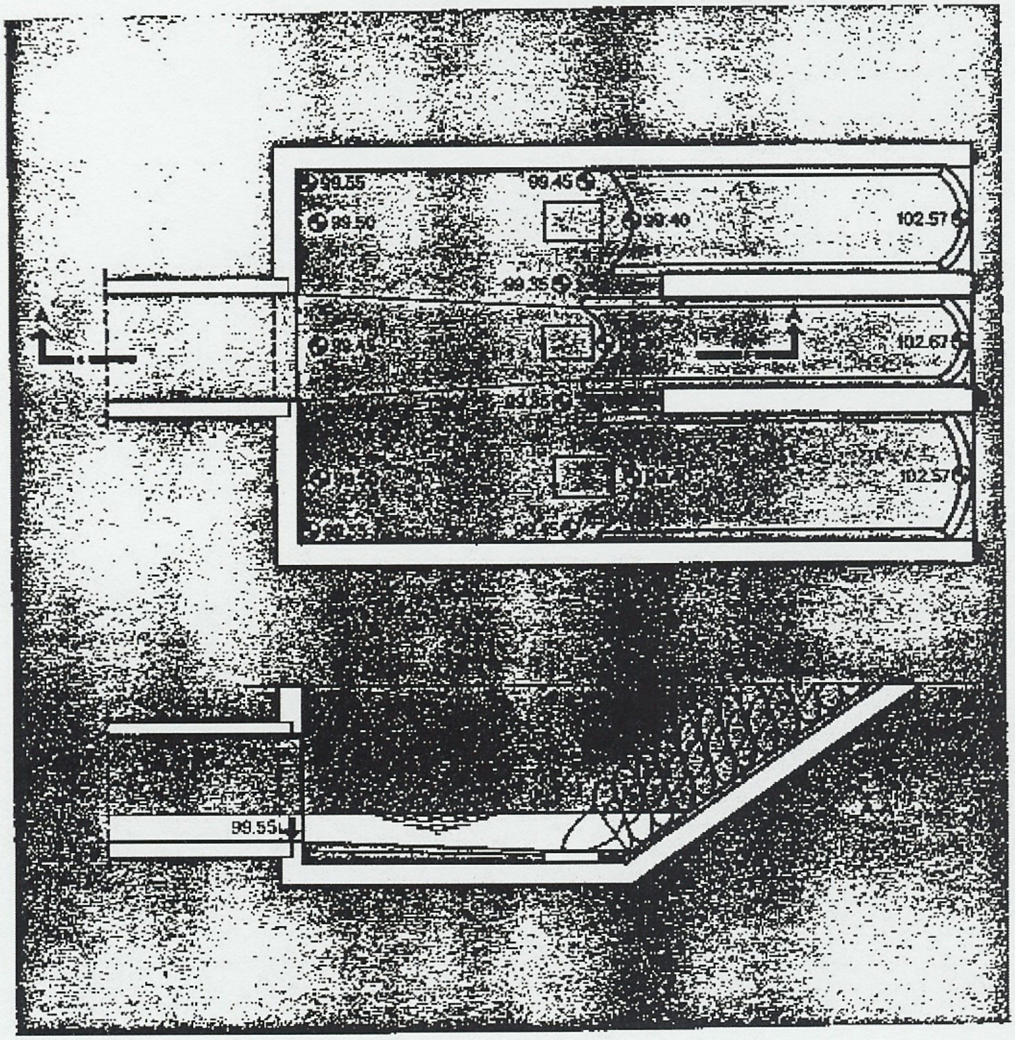
3.2 General Layout of Pumping Plants

There are many possible solutions for the structural and architectural arrangements of Archimedian screw pumping installations so far as the construction of the head gear is concerned. As regards external appearance, many pumping stations have been built which make a pleasing addition to the landscape owing to the detailed design of the structure and the colour schemes that have been adopted for the visible parts of the structure. This enables sewage plants, which in the past have been rather neglected in this respect, to have an appearance commensurate with their social and economic importance and, indeed, to form an object of interest to the observer.

A number of such pumping plants equipped with screw pumps are illustrated in the illustrations section of this book. The skill that their designers have exhibited in the use of form and colour becomes particularly apparent if they are compared with the older generation of sewage plants. The Archimedian screw pump is also becoming increasingly popular for use in intermediate pumping stations, since the structure can be located completely underground without any spe .fic operational disadvantages arising. The principal advantage in these installalations is the absence of any screens, and hence of the difficulties encountered in cleaning the screens and removing the debris. The silent operation of a screw pump also often makes it possible for an intermediate pumping station to be located in direct proximity to a residential area, where there could be no question of installing centrifugal pumps owing to their high noise level and the resultant nuisance that they would cause to the inhabitants. The hydraulic noises produced by centrifugal pumps, which are often transmitted over consi rable distances by the delivery pipes, and which prove particularly disturbing at night, are especially unpleasant.

The basic parts of a structure for housing an Archimedian screw pumping station are the supply basin, the screw troughs, and the main discharge basin and channel, above which the head gear, which contains the motors and reduction gearing for driving the pumps, is located. The term head gear is used, even where the entire installation is located underground.

The supply basin is constructed so as to give the correct immersion of the bottom end of the screw, as discussed in Part 2/2.11, attention being paid to the correct water level and to the arrangements necessary to ensure that the rate at which the pump cycles on



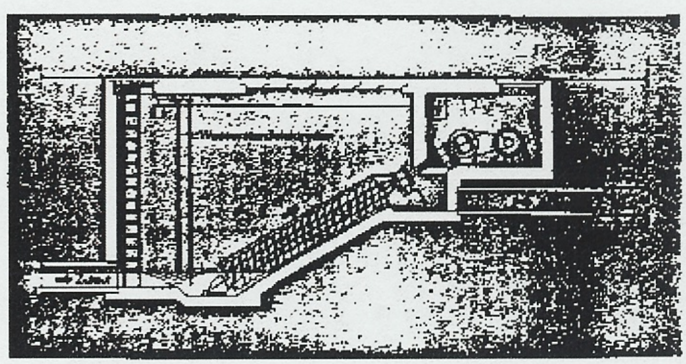
22 Section and plan of a supply basin
showing spot heights

on and off is satisfactory. If it is possible to accumulate effluent in the supply pipe, the supply basin can usually be made very small and shallow. The bottom of the basin should slope toward the contact points of the screws to prevent any stagnant areas forming in the basin with the resultant unpleasant accumulation of slurry and silt. Before starting to plan the installation it is therefore necessary to make a thorough investigation of the feed system to the screw pump, paying particular attention to the permissible levels in the supply pipe, if the optimum economic design of the pumping installation is to be achieved. The majority of design faults can be traced to errors in the determination of the water levels in the supply and discharge basins.

The way in which the total capacity is divided among several pumps, and the number of pumps to be installed, also has a major effect on the level of the bottom of the supply basin, if one bears in mind that, to achieve the optimum delivery rate, the depth of immersion of the screws must be at least $(r + D) \cos\beta$ above the contact point if the full delivery of which the pumps are capable is to be achieved. Where several Archimedian screw pumps are working in parallel, the control system should be such that they are switched-on and off at differences in the water level of at least 0,10 m between the individual switching points, unless other factors (such as the diameter ratios and immersion depth) impose different conditions.

Wasserstandelektroden = Water-level electrodes
Zulauf = Supply
Ablauf = Discharge

23 Exemple of an intermediate pumping station located below ground



All the points to be considered when designing a supply basin are summarised in Fig. 22, which is an example of a properly designed supply basin.

The width of the structure is usually determined by the diameters of the screws that are to be installed, the approximate value of 1,5 times the sum of the screw diameters having been quoted above. This applies unless other considerations, such as a convenient arrangement of access stairs, or the dimensions of the storage basin, impose different requirements. If additional space is required for the supply basin it is preferable to increase the length of the structure, since the width of the sump basin exerts a major influence on the overall layout and the construction costs of the pumping station. It is also usually easier to solve any problems that may arise in the location of a pumping station, especially an intermediate pumping station, on a plot or on the line of a road if a long, narrow structure is involved.

The question of the screw troughs, which form part of the structure and which help determine its size, is discussed separately in Section 3.3. The basic length of the structure can be determined without difficulty from the delivery head required which, in turn, controls the length of the screw, as described in Part 2/2.14. More detailed information on the requirements for the discharge pipe from the discharge basin has also been given in Part 2/2.12. The circumstances in which several screw pumps can discharge into a common channel, without any danger of return flow (as compared with the circumstances under which separate discharge channels with non-return valves to prevent return flow will be required) is discussed separately in Section 3/3.8 "Individual and Parallel Operation of Archimedian Screw Pumps".

The arrangement of a pumping station in which the entire structure is located underground is shown in Fig. 23 (p. 57), while several of the illustrations in the picture section, Part 8, show installations which have proved successful from the constructional and the architectural points of view. Examples of the two main types, the underground and the open pumping station, are shown in the plans and sectional drawings of Fig. 23 (p. 57) and Fig. 24 (p. 60). Hybrid arrangements can also be adopted. An enclosed installation, located above ground, (covered screw troughs) will be preferred where the pumping station is to be located in a rural area and where the ground-water table is near the surface, or where, to preserve the appearance of the landscape, the pumping station is to be as inconspicuous as possible, yet undue expense is to be avoided. This solution has the advantage of being quite inconspicuous and also that, in the majority of cases, no special fencing is required, since the installation is well protected against unauthorised interference.

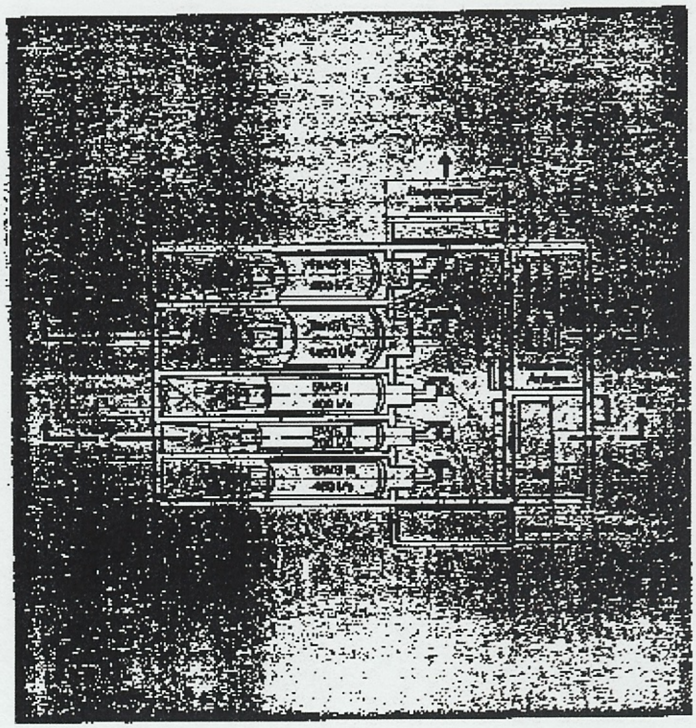
Fig. 33 shows the simple method of installing the switch-gear and controls and fuses, and the cable end connections and the wiring. One of the illustrations shows an example where the machinery and the electrical equipment are all installed together in a sheet-metal casing. The use of Archimedian screw pumps often enables a sewage pumping station to be made very attractive in appearance, and to make a pleasant addition to the lan , so that there is no disadvantage at all in ignoring the former rule that a sewage plant should be as unobtrusive as possible. If the screw pump is located in the open it is necessary to bear in mind the variations in the length of the screw that can arise due to the effects of temperature. If, as an extreme case, it is assumed that the temperature can vary by $\pm$ 50° C about the mean temperature of + 20°C that prevailed at the time of manufacture, due to intensive sunshine (up to + 70°C), or extremely-cold winter temperatures (down to − 30°C), the change in length per metre run of the screw will amount to $\pm$ 0.575 mm. The change in length over the whole length of the screw will then have to be accommodated by the adoption of a suitable design for the bearings. The diameter of the screw will also vary by the same value of 0.575 mm per metre, but only half this change in diameter affects the gap width. It must be pointed out that, if the changes in length or diameter that can arise due to temperature variations are ignored, considerable difficulties and kdowns can result from the consequent misalignment of the bearings, additional deflection of the screw, and rubbing of the screw against the trough. Whether these factors justify covering the screw in will not be discussed here. Wherever possible enclosure is to be avoided due to the dangers of corrosion that it brings in its train.

3.3 Construction of the Pump Trough

The pump trough is the semi-cylindrical channel in which the screw rotates and so pushes the water from the supply basin to the discharge basin by means of the lower flanks of the screw blades. Until some ten years ago this pump trough almost always consisted of a steel structure. The current stage of building technology, however, ensures that there is nowadays no difficulty in making this trough of concrete with the result that, at the present time, the use of the steel trough is restricted to very small sizes or to special cases, such as portable troughs.

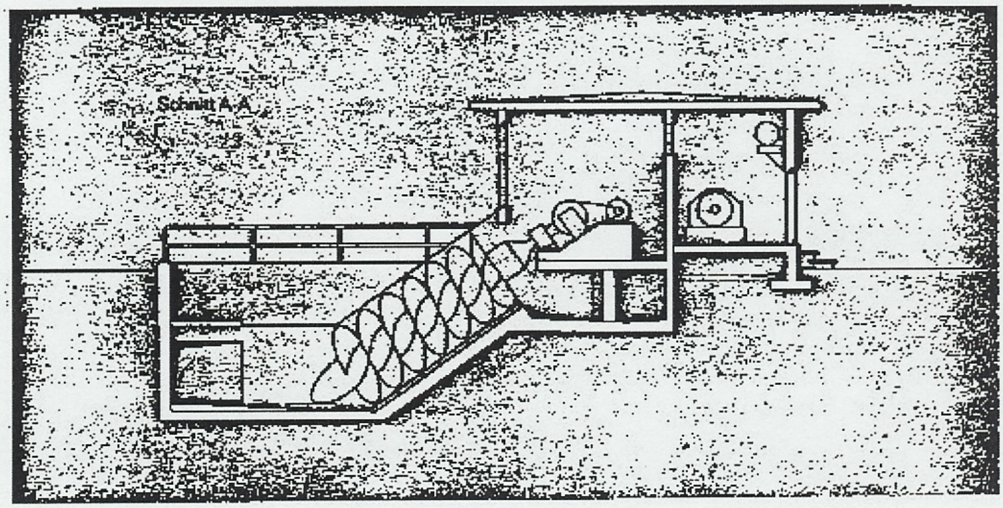
The usual practice at present is for the foundation of the trough, which forms its main structural element, to be built in unfinished form with the rest of the structure, and to be in the shape of the lower half of an octahedral figure, a template being used to maintain the section to the correct shape, and the vertical sides being carried up to their full finished height. The screw is then fitted, and a strip of steel bar, having the same thickness as the desired gap between the screw and the

Wasserstandelektroden	=	Water-level electrodes
Regenwasser zum Vorfluter	=	Rainwater to main drain
Notstromanlage	=	Emergency generator
El-Teil	=	Electricity substation
Schmutzw. z. Klarwerk	=	Effluent to treatment plant

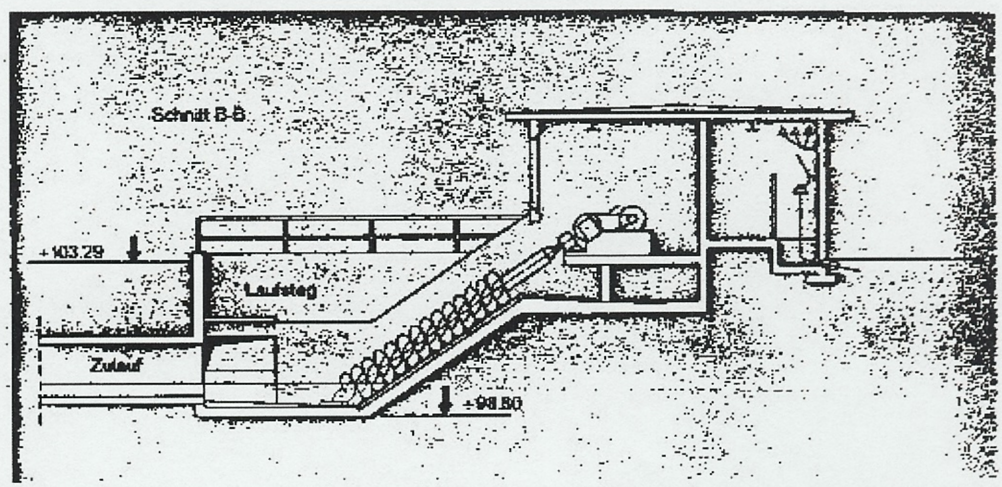


24 Example of an open-type pumping station equipped with five Archimedian screw pumps

Schnitt = Section



Schnitt = Section
Laufsteg = Catwalk
Zulauf = Supply pipe



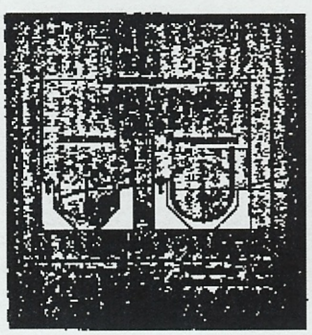
trough, is clamped to the outside of the blades over the full length of the screw. The screw is then rotated slowly, and high-grade cement mortar is fed in at the bottom end, so that the corners of the concrete octahedral, which is dampened, are filled out, while the surplus material is moved along by the rotating screw blades. In this way the screw forms its own bed. A diagrammatic representation of the two main stages of this process, showing the concrete foundations and the finished pumping trough is given in Fig. 25.

This method of making the trough offers the advantage that it automatically allows for the deflections of the screw due to its own weight. It can therefore be used for screws that are longer than the limits given by the strength calculation described in Part 2/2.17. Nevertheless, since other factors such as blockages between the screw and the trough, torsional and vibration effects, and eccentricity of the shaft tube all have to be taken into account, it is not advisable to exceed the values for L_{max}, given in Diagram 1, to any appreciable extent.

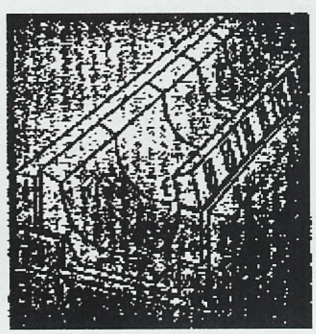
The construction of prefabricated concrete components has not advanced to the stage where there is every inducement to make pumping troughs from prefabricated sections. The considerable extent to which the external diameters of the screws have been standardised is another point in favour of the use of prefabricated components. Apart from

Umfassungsgewande	=	Surrounds
Rohzustand	=	Semi-finished condition
Fertigzustand	=	Finished condition
Vorbeton	=	Initial concrete
Füllbeton (...)	=	Cement liner (Placed by screw)

25 Section trough semi-finished and finished stages of pumping trough concreted in place



26 Construction of a pumping trough using prefabricated components



the high standard of manufacture to close dimensional tolerances, the main advantage is probably the smooth surface finish that can be achieved, and the simplicity with which the components can be assembled. The deflection plates which extend above the centreline of the screw can take the form of box-section prefabricated elements, placed on top of the side of the trough and secured by means of foundation bolts let into the trough components. This eneables them to be removed without difficulty if the screw has to be dismantled. A perspective view of this method of constructing the trough is shown in Fig. 26.

From the design point of view there is no difficulty in varying the delivery head of an Archimedian screw pump by making one of the upper sections of the trough adjustable, or by fitting an extendable apron. In land reclamation work such measures are indeed often adopted due to the importance of the delivery head. For an effluent pump, however, where changes in delivery head are less common, complications of this type are best avoided unless special conditions obtain, owing to the possibility that they may become faulty.

3.4 Influence of Screw Design on Pumping Installation

The planning of a pumping station using an Archimedian screw pump should always be preceded by a comprehensive investigation to determine which of the possible solutions will be the most economical from the point of view of the initial and the running costs.

So far as the initial costs are concerned it is wrong to take only the costs of the machinery andequipment, or only the building costs, into account.

To enable these economic calculations to be made it is necessary for the supply and discharge conditions to be fixed, and for the quantities of water that will have to be handled at various times to be known with as much accuracy as possible. The mean quantity of water to be handled during the course of a year should also be taken into account. If an Archimedian screw pump is to be used as an intermediate pump, it is in most cases necessary to make provision for the delivery rate to vary from the minimum dry weather flow to the maximum flow under storm conditions, since it is rare for main drains to be available for carrying off rainwater from the intermediate pumping stations, and it is not practicable to specify that a separate rainwater drainage system be provided. Some information on the economic distribution of the load among the various screws to meet the requirements of the differing amounts of water that have to be handled under the different conditions are given in section 3/3.9, so that further discussion of this point can be omitted here, since the distribution of capacity is one of the main factors affecting the overall costs of the plant. Here, we will first consider the question of the inclination of the screw to the horizontal, and the number of starts of the screw for practical cases. Some information on the number of starts has been given in Part 2/2.8, and on the choice of the angle of installation in Part 2/2.9, and this should be borne in mind when studying these points.

It is not possible to lay down any general rule for the correct choice of the angle of Installation, since this is affected by a large number of factors peculiar to the plant concerned.

It is true that an increase in the angle of installation will cause a reduction in the length of the structure required, and also of the screw. As the angle of installation is increased, with all other factors held constant, there is a decrease in the capacity of the screw so that, to achieve a given delivery rate, the diameter of the screw will have to be increased. The reduction in the length of the structure will thus be accompanied by an increase in width, and it will be necessary to compare the advantages and disadvantages of this arrangement when attempting to find the most economical solution.

If the possibility of changing the number of starts is being considered, this will not affect the length of the structure. The reduction in the capacity of the screw that results from a reduction in the number of starts will, however, again make it necessary to increase the screw diameter, and hence the width of the structure, if a given delivery rate is to be maintained. In this case any savings that may accrue in the cost of the machinery will have to be weighed against the increased cost of the structure. To estimate the costs of the various types of screw it is possible to make use of the total weight given in Table 6, as shown in the following example:

For a dry-weather flow of 20 to 100 litres/s and a storm flow of 1000 litres/s maximum, the capacities of the screws to be installed in an intermediate pumping station were chosen to be 2 x 60 litres/s (for pumping the effluent) and 2 x 600 litres/s (for pumping the rainwater). The delivery head involved is 4.0 metres from the highest permissible immersion depth at the bottom of the screws to the highest discharge basin level.

For pumping the effluent at a rate of 60 litres/s maximum per screw, and a head of 4,0 metres it would be possible to use the Type 11.06 installed at an angle $\beta = 30^{\circ}$ and with $a = 3$; at the maximum permissible speed of 70 rev/min this screw can deliver 68.5 metres/s. A speed reduction in the ratio

$\frac{n}{n'}$ equal to $\frac{Q}{Q'}$, would give

$$n' = n \cdot \frac{Q'}{Q} = 70 \cdot \frac{60}{68,5} = 61 \text{ rev/min.}$$

The screw concerned is a three-start one having an external diameter of 593 mm, and its shaft tube diameter of 355,6 mm would enable the length of the screw to have a maximum value of 8,7 metres, for a wall thickness of 12,5 mm, if the deflection f at the centre of the screw is not to exceed the gap width s_{sp} of 3,5 mm. A shaft tube having a wall thickness of 7,1 mm would not be suitable since with this, the length between the bearings would be restricted to 7,45 metres.

The angle of inclination β has been fixed at 30°.

The actual length of the screw for the two effluent screws would be :

$$L = \frac{H}{\sin\beta} + \left(\frac{1+\delta}{2\,\mathrm{tg}\,\beta} - \frac{k}{\sin\beta}\right) . D = \frac{4,0}{0,5} + \left(\frac{1+0,6}{2 \,.\, 0,577} - \frac{0,15 \,.\, 0,593}{0,5}\right).$$

$$0,593 = 8,72 \text{ m}$$

and this would just enable the deflection to be kept within the required value. $\frac{\eta_m}{\eta_{ges}}$ in this case would become 1,1464, which gives a value of η_{ges} = 78,4 % (if we assume η_m = 90 % as previously recommended).

For the same angle of inclination the screws for the rainwater flow should be of Type 11.16 at = 30°, these again being three-start screws which have a diameter of D = 1588,6 mm and a shaft tube diameter of d = 711,2 mm for a 10-mm wall thickness. This screw can have a length of up to 10,1 metres for f = 5,0, s_{sp} = 5,7 mm. The rated speed of 37 rev/min, when the delivery would be 808 litres/s, should be reduced to

$$n' = 37 \,.\, \frac{600}{708} = 31 \text{ rev/min}$$

to enable the desired delivery of 600 litres/s to be achieved. Then $\frac{\eta_m}{\eta_{ges}}$ will have a value of 1,1288, corresponding to η_{ges} = 79,7 %. The length of the screw to be equipped with blading would then be :

$$L = \frac{4,0}{0,5} + \left(\frac{1+0,45}{2 \,.\, 0,577} - \frac{0,15 \,.\, 1,589}{0,5}\right) 1,589 = 9,22 \text{ m}.$$

If we ignore, for the moment, the fact that the rainwater screws can be designed to suit the maximum immersion conditions, only the effluent screws needing to have the ability to empty the supply basin completely during the operating period.

For the same size of supply basin the length of the structure 1_B to accommodate the screw troughs is obtained from the horizontal projection of the maximum length of screw equipped with blades :

$$1_B = L \cdot \cos \quad = 9,22 \cdot 0,866 = 8,00 \text{ m}$$

While, from the information given in section 3/3.2, the width of the structure b_B becomes :

$$b_B = \quad D \cdot 1,5 = (2 \cdot 0,6 + 2 \cdot 1,6) \; 1,5 = 6,60 \text{ m.}$$

The encompassed space for the structure to accommodate the screw trough then becomes

$$\frac{8,0 \cdot 6,60}{2} = 26,4 \text{ m}^3 .$$

The basic weight of the four screws to be installed in this pumping station, without that portion of the shaft tube that is not equipped with blades, the bearings, trough, or aprons is then, from the Tables :

$$G = 2 \cdot 8,72 \cdot 149,8 + 2 \cdot 9,22 \cdot 367,4 = \text{approx. } 9410 \text{ kg.}$$

If we now compare this arrangement with one in which the screws are installed at an angle of 37^o, for example, and in which the number of starts $a = 2$, we would use for the effluent pumping screws Type 11,07, for which $\frac{m}{ges} = 1,1720$ (corresponding to $_{ges} = 76,7$ %), and which can still just be run at their rated speed of 63 rev/min (Q = 62,3 litres/s). The length of the screw that is equpped with blades then becomes :

$$L = \frac{4,0}{0,602} + \left(\frac{1 + 0,51}{2 \cdot 0,753} - \frac{0,15 \cdot 0,692}{0,602} \right) \cdot 0,692 = 7,22 \text{ m.}$$

Dealing now with the rainwater screws it is found that, under the same conditions and for the same values of   and a, the Type 11,20 with a rated delivery of 716 litres/s is suitable, so that the speed can be reduced to :

$$n' = 32 \cdot \frac{650}{712} = 27 \text{ rev/min.}$$

Since $n = 27$ rev/min is better than 70 % of n_{max} (which would be 0,7 . 32 = approx. 22,5 rev/min) the value for $\frac{\eta_m}{\eta_{ges}}$ can be taken as 1,1429, corresponding to the full value of η_{ges} = 78,7 % given in the Table. The corresponding length of the rain-water screw then becomes :

$$L = \frac{4,0}{0,602} + \left(\frac{1+0,46}{2 \cdot 0,753} - \frac{0,15 \cdot 1,987}{0,602}\right) 1,987 = 7,58 \text{ m}.$$

The length for the structure to accommodate the screw trough then becomes :

$$l_B = 7,58 \cdot 0,8 = 6,06 \text{ m}$$

and the width of the structure will be

$$b_B = (2 \cdot 0,7 + 2 \cdot 2,0)\ 1,5 = 8,10 \text{ m}.$$

The encompassed space for the structure to accommodate the screw troughs is then $\frac{6,06 \cdot 8,10}{2} = 24,5 \text{ m}^3$. The basic weights of the installed screws per metre run can again be taken from the Tables. Since the lesser length of the effluent screws enables the smaller wall thickness of 7,1 mm to be employed, the figure becomes :

$$\Sigma G = 2 \cdot 7,22 \cdot 107,8 + 7,58 \cdot 423 = 7918 \text{ kg}.$$

Summarising the results of this comparison it will be seen that, for a reduction in the screw efficiency of the effluent screws of 1,7 %, and of the rainwater screws of 1,0 (due to an increase in the angle of inclination β from 30° to 37° and a reduction in the number of starts from 3 to 2) there will be a reduction of 1,9 m^3 in the encompassed space of the structure, and a reduction of 1492 kg in the weight of the screws.

To enable cost figures to be inserted in the economic comparison it may be assumed that a figure of 300,-- DM/m^3 can be adopted for the cost of the structure, and that the finished cost of a screw amounts to 2,- kg; these figures will be sufficiently accurate for a rough preliminary calculation.

There should now be no difficulty in comparing the savings in the initial costs with the effects of the reduction in the efficiency of the effluent screws, which are the ones that are most frequently in use. If the cost of power for running the screw pumps is taken as 0,10 DM kWh, the following figures will be obtained in this example :

Reduced cost of structure	$1{,}9\ m^3 \times 300{,}\text{--}\ DM/m^3$	=	570,-- DM
Saving in weight	$1492\ kg \times 2{,}\text{--}\ DM/kg$	= approx.	2.980,-- DM
	Total saving	=	3.550,-- DM

If the overall efficiency drops from $\eta_{ges} = 0{,}784$ to $\eta_{ges} = 0{,}767$, as a result of the increased inclination of the screw and the reduction in the number of starts, and if the mean rate of inflow is 40 litres/s so that, during the course of a day, 3450 m^3 of effluent have to be raised through 4 metres, the daily increase in the cost of electrical power becomes :

$$A_t = \frac{Q_T \cdot H}{367{,}2 \cdot \eta_{ges_2}} - \frac{Q_T \cdot H}{367{,}2 \cdot \eta_{ges_1}} = \frac{3450 \cdot 4}{367{,}2 \cdot 0{,}767} - \frac{3450 \cdot 4}{367{,}2 \cdot 0{,}784}$$

$= 1{,}20$ kWh/day
or 0,12 DM/ day

In round figures the reduction in efficiency would thus result in a negligible annual increase in the electricity cost amounting to only 34,80 DM. The cost of the annual depreciation of the reduced building costs, calculated for the expected life of the installation, far exceeds the additional cost of the power in this particular case. It has thus been shown that in this example the choice of steeper screws having a reduced number of starts would be the more economical solution, despite the reduction in the efficiency. It would, however, be quite wrong to assume that this conclusion is of general validity; similar calculations should be made for each particular case.

3.5 Possibilities for Modifying Performance

The question of the choice of the speed of rotation of Archimedian screw pumps was discussed in some detail in Chapter 2/2.10. It was established that the speed that is given by the formula

$$n = \frac{50}{\sqrt[3]{D^2}}$$

is the optimum speed of the screw at which water will just fail to flow from one chamber down to the next below along the screw shaft. If the speed is kept below this value, the delivery will, within certain limits, be reduced in proportion to the screw speed, i.e.

$$\frac{n}{n'} = \frac{Q}{Q'} .$$

It would be a reasonable proposition, therefore, to choose a screw for an Archimedian screw pump which has the correct geometrical dimensions to give the delivery required, when the pumping plant is fully completed and running at maximum capacity, and to allow for the smaller quantities of water that have to be handled (in the initial stages of operation of the plant) by, at first, running the screw at a lower speed than that appropriate to the optimum delivery rate.

In practice this could be achieved by first fitting a gearbox giving a higher reduction ratio, and then changing the gearbox when the stage is reached where full delivery is required. This would, however be a very expensive procedure, and should be avoided if possible.

However, there is another, much easier method of changing the capacity of the pump at a later date. This consists of changing the belt pulleys of the V-belt drive transmitting power from the motor to the gearbox, which in any case will be fitted to avoid shock loads on starting, in which case the change in speed is proport'··'! to the mean belt pulley diameter d_R

$$\frac{d'_R}{d_R} = \frac{n'}{n}$$

It is possible to successfully change the speed in this way, and without undue expense, down to a speed of $n' = 0{,}6\ n$ without any major drop in efficiency occurring. The possibility of fitting a two-start screw initially and replacing it with a three-start screw at a later date is also often considered. The change in pumping capacity that can be achieved in this way has been discussed in section 2/2,8, and it was shown that it is reduced approximately in the proportion 0.8 : 1. If, then, the speed is also reduced to 0.6 n, an overall reduction in pumping capacity in the proportion 0.48 : 1.0 will be achieved or, in other words, the delivery rate of the pump is approximately halved.

In practice such a procedure can hardly be recommended. since the motor will, in any event, have to be capable of producing the power required when the plant is operated at full capacity, if continued use is to be made of any major parts of the mechanical and electrical equipment of the pumping station when its capacity is increased. If now an electric motor, rated at the power required to operate the pump at full capacity, is operated under part-load conditions, its efficiency will drop, and an increased reactive current will be required. Such an installation will not normally be economic to operate.

Nevertheless, there are a very few occasions that arise (in practice) where it is necessary to adopt such a solution since, as mentioned above, the Archimedian screw pump will work without any unacceptable reductions in efficiency down to a delivery rate of about 30 % of its rated capacity. The screw pump appears therefore to be particularly suitable for use where a subsequent increase in delivery rate is envisaged. Changes in delivery head, on the other hand, are difficult and expensive to effect at a later date. The complications involved in the use of adjustable scfews can be seen in the illustrations section. Nevertheless, the construction of adjustable troughs has proved justifiable for certain specific cases.

3.6 Storage Volume on the Inflow rate

Pumping capacity, and pump cycling frequency is established, and is then differentiated to give the maximum storage volume, the maximum storage chamber will be obtained when the inflow rate is one-half the pumping capacity. The calculations should be based on the pumping capacity of the largest effluent pump. Since, as mentioned previously, an Archimedian screw pump will still work with a comparatively high efficiency if the delivery rate drops to about 30 % of the design capacity Q, only 65 % of the rated capacity of the effluent pump should be taken as the average delivery rate over an operating cycle, as given by

$$Q_{av} = \frac{(1,0 + 0,3) \; . \; Q \, .}{2}$$

The cycling frequency i (cycles/h) is discussed in Chapter 3/3.7. The minimum storage volume will then be given by

$$J_{storage} = \frac{0,65 \; . \; Q \; . \; 3600}{2 \; . \; i} \; .$$

If a value of 12 cycles/h is chosen for i, the volume can be expressed directly in terms of the delivery rate of the largest effluent pump, i.e.

$$J_{storage} \cong \text{approx. } 100 \; . \; Q \, (m^3)$$

where the delivery rate is expressed in m^3/s, as specified in the list of symbols and dimensions. This formula thus indicates that the inflow storage volume for screw pump should be about 100 times the delivery rate (per second) of the largest effluent pump to be installed. Experience shows that the maximum cycling frequency of 12 cycles/h, under the most adverse inflow conditions (which has been assumed in this case) can, in fact, be withstood by all the components concerned without any difficulties arising.

In many cases it is possible to hold back a certain amount of water in the supply pipes, and the size of the basin can then be reduced by this amount. This possibility should not, however, be exaggerated, owing to the danger of sludge being deposited, especially in pipes that have only a slight slope.

Since the delivery rate of an Archimedian screw pump immersed to below the centre of the screw shaft will, ingeneral, exceed the figure of 30 % of the maximum delivery rate which is still economically justifiable, the depth to which the water level in the supply basin can fall will be at least equal to H-I = r. cos . Ignoring the possibility of using the supply pipes as a back-up storage volume, the surface area of the basin required will then be given by :

$$F_{basin} = \frac{J_{storage}}{r \cdot \cos\beta}$$

where r is the radius of the shaft tube of the largest effluent pump that is installed. The width of the basin will be governed by the diameters of the screw that are installed, so that it is now possible to calculate the length of the basin without difficulty. If the surface area given by the formula is too large, some limited use can be made of the back-up storage capacity of the piping.

Since the pipes run full, to some extent, when the rainwater pumps are in operation, the volume that results can be regarded as a storage volume, and it will usually not prove necessary to increase the dimensions of the basin. A check should be made in each case, however, to make sure that the back storage volume of the pipes, together with the storage volume that is provided in any event amounts to at least 60 times the delivery rate of the largest rainwater pump. This figure presupposes that the cycling frequency under the most adverse conditions will amount to 18 cycles/h; in view of the ratity with which such conditions will occur, this figure is quite acceptable.

3.7 Determination of Cycling Frequency

The various types of cycle under which equipment can be operated are arranged in a system of nominal cycle classes in the Rules for Electrical Machinery and Apparatus (VDE 0530). For example, a distinction is made between continuous and intermittent operation; whereas continuous operation means that the machine is running more or less continuously, under the conditions of intermittent operation the running time or full-load time is soshort that the motor is unlikely, in practice, to attain its normal operating temperature. and will usually cool to the ambient temperature while it is stopped.

Under the most adverse conditions the motors driving Archimedian screw pumps can be described as operating under intermittently stopped conditions rather than intermittently running; the periods during which the motors are running at, or near, full load alternating with those periods during which they are stopped - periods of such short duration that the motors and the switchgear cannot cool to the ambient temperature. The determining factor is the duty cycle.

Since the largest number of starts occurs when the rate of inflow corresponds to half the pump delivery rate, the duty cycle will then amount to 50 % of the total service time.

Very similar conditions prevail in the hydraulic engineering industry, where surge tanks are employed, although the starting conditions are rathermore severe for screw pumps than for centrifugal pumps because of the larger moving masses. Under those conditions, cycling the plant at the rate of 8 - 15 cycles/h has proved quite satisfactory and has not led to complications, so that the assumption of i = 12 cycles/h appears quite acceptable, especially since the large piping system associated with a sewage plant ensures that the conditions under which the maximum number of cycles is necessary are not often attained, and then only for brief periods.

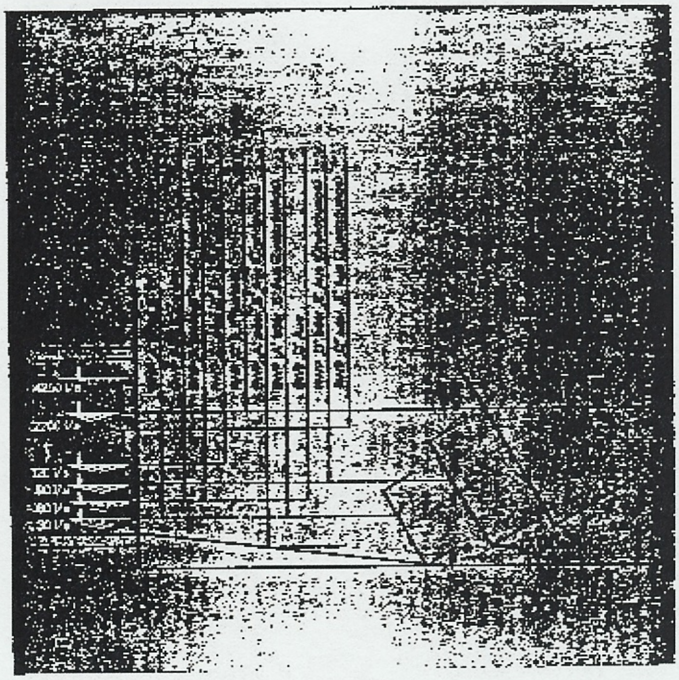
To prevent the storage basins required for the rainwater pumps becoming too large, a cycling rate of 18 cycles/h can be assumed; this is adequate to deal with the comparatively rare case when the inflow of water is at a rate precisely equal to half the pump delivery rate. If the size of the storage basin, calculated on the basis of effluent flow, is less than 60 times the delivery rate/s of rainwater pump, the dimensions should be vbased on the latter figure. Fortunately, however, it is possible to use the capacity of the pipes up to the highest permissible static head when the rainwater pumps are in use, and it will therefore probably prove possible to avoid any increase in the storage volume.

The cycling rates mentioned can be dealt with without difficulty by the electrical switchgear and controls, if these points are borne in mind when selecting and installing the switchgear. Switchgear testvoltages should comply with VDE 0111, and the minimum clearances with VDE 0101. VDE 0660 specifies a life of one million ope ations for simple air-break and oil-immersed contactors, and this can be increased to up to ten million ope rations in the case of heavy-duty equipment designed for intermittent use.

3.8 Individual and Parallel Operation of Screw Pumps

If several Archimedian screw pumps are installed side by side in a pumping station, the structural arrangements, both on the inflow side and also, more particularly on the outflow side, must accommodate all possible circumstances that can arise in service.

SWP	=	Effluent pump
RWP	=	Rainwater pump
A oder B	=	A or B
A oder B mit C	=	A or B with C
Ein	=	On
parallel	=	Parallel
Aus	=	Off
Grundpumpe	=	Base-load pump
Zusatzpumpe	=	Supplementary pump



27 Example showing the locations of the ends of the screws and the water levels at which switching operations occur for five Archimedian screw pumps operating in parallel

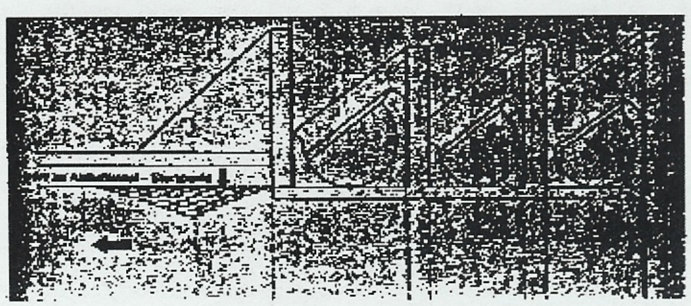
At the lower end, the screws should extend only sufficiently to enable them to operate at their full rated capacities when the water level in the basin is at the appropriate maximum level. This applies more particularly to the rainwater screws, while the effluent screws (and, in particular, the unit with the lowest capacity) must be capable of emptying the supply pipes and the basin completely. This requirement arises from the fact that, as stated earlier, deposits of all types are to be avoided in the supply basin and also in the pipes where these are used as a back-up storage medium. The aim must therefore be to empty the basin completely during each operating period of the base-load pump.

Apart from any limitations imposed by the cycling frequency, the supply basin must be designed in such a way that the differences in levels between the points at which the different pumps are started and stopped must be at least 10 cm. A diagramatic view of the arrangement of the bottom ends of the screws in the supply basin for the example chosen to illustrate the method of dividing the total load among a number of screws is shown in Fig.27.

Any errors that may be made in the design of the top end of a pumping plant have a greater effect than errors at the bottom end of the plant, and can be rectified later only at considerable expense. If Archimedian screw pumps, working in parallel, are to discharge into a common discharge basin or channel, spe ial care must be taken

HHW im Anflußkanal = Sturzpunkt = Max. water level in discharge channel = spill point

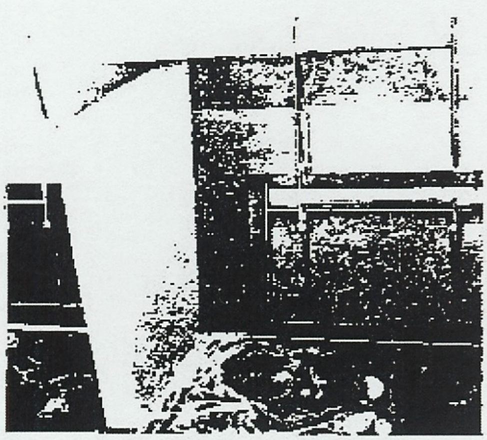
28 Arrangement of spill points where several pumps discharge into common channel



to ensure that the spill points of all the screw troughs are higher than, or at least level with, the highest water level that can occur in the discharge channel under the most adverse conditions. If this point is neglected, the water will run back over the spill point if a screw is stopped. Quite apart from the loss of water that this involves, dirt can be deposited between the blades of the screw and the trough, causing a blockage which might make it difficult, or even impossible, to start a screw that has been stopped for a considerable time.

If screws are to work in parallel and discharge into a common channel without any non-return devices being provided, there is the additional requirement that it is essential that the spill point of the smallest screw to be installed is at a higher level than the maximum water level in the discharge channel. The only reliable solution where the screws discharge into a common multiple-flow discharge channel is to raise the level of the spill point as shown in Fig. 28.

It is usually worth while to carry out an economic investigation to determine whether it might not be preferable to carry the discharge from the screws along separate channels provided with non-return valves which then, in turn, discharge into the common discharge channel, rather than to raise the spill points with the consequent permanent increase in the delivery head. With high delivery rates and major variations in the water level in the discharge basin economic reasons will cause this solution to be adopted.



29 Use of non-return flaps in discharge channels where there is possibility of spill point being submerged

If, for example, a pumping station having a screw with a diameter of 1 is considered, the spill point will have to be raised by at least 0.20 if, as a safety measure, it is decided that it should be located at least 5 cm above the discharge head of 0.15 D which is sufficient to prevent return flow. If the average delivery head of such a plant amounts to 4.0 m, the delivery head, and hence the pumping costs, will be increased by 5 % of the total pumping costs. If the cost of this, calculated over a period of ten years, amounts to more than the cost of constructing separate discharge channels with non-return valves, the latter solution should be adopted. The appearance of such an installation is shown in Fig. 29.

Finally it must be pointed out that unsatisfactory design of the supply and discharge channels can cause blockages and eddying. Careful attention should be paid to this point, particularly where high-capacity screw pumps are concerned. It is advisable to have model tests carried out if difficult conditions are encountered in a large plant.

8.9 Economical Capacity Distributions

In general, the delivery rates of screw pumps should be chosen so that one pump, or preferably two : - more pumps working simultaneously, can handle the maximum inflow. In the case of pumping plants where no alternative provision is made for dealing with rainwater, and no suitable storage area is provided, the distribution of the capacity is one of the basic requirements. If the build-up of water is not a disadvantage, the drainage system can be used for the temporary storage of rainwater during periods of heavy reinfall. In such cases the maximum permissible water level in the system should be determined with the utmost care for each individual installations.

In the majority of cases separate pumps, should be provided to deal with the dry-weather flow and with the storm flow. If it is impossible to provide an emergency outlet near the pumping installation, the largest rainwater pump should also have a reserve capacity. The following example of a combined drainage system shows a suitable methof of selecting the capacities for the various screw pumps installed in a pumping station.

Basic data :	Number of inhabitants served	=	6000
	Specific sewage flow	=	200 litres/person per day
	Total built-up area	=	85 ha
	Max. rainfall run-off	=	125 litres/s per ha
	Assumed mean impermeability factor ψ	=	0,4

Requirement :

The effluent, in quantities up to five times the dry-weather flow averaged over a 14 h period, is to be passed to the treatment plant, while the rainwater is to be passed direct to a main drain.

Sewerage flows

a) Effluent : $6000 \times 0,200 = 1200\ m^3/day$

$$Q_{14} = \frac{1200}{14} = 86\ m^3/h \qquad = \quad 24\ \text{litres/s (max. effluent flow)}$$

whence 5 x dry-weather flow $\quad = \quad 5 \times 24 = 120$ litres/s
(max. effluent pumping capacity)

$$Q_{18} = \frac{1200}{18} = 67\ m^3/h \qquad = \quad 18,5\ \text{litres/s (mean effluent flow)}$$

$$Q_{37} = \frac{1200}{37} = 32\ m^3/h \qquad = \quad 9\ \text{litres/s}$$

(minimum effluent flow = night flow)

b) Rainwater

$$Q_R = 85\ .\ 125\ .\ 0,4 \qquad = \quad 4250\ \text{litres/s}$$

(with no storage capacity in supply pipes)

Selection of screw pump capacities

for effluent : two screws, each 30 litres/s (screw A + B)
one screws, 60 littes (screw C)
(used together, can deal with five x dry weather flow)

for rainwater: two screws, each 2200 litres/s (screws D + E)
(used together, can deal with Q_R)

With this selection of pump capacities, one of the smaller effluent pumps A or B, which can be used alternately or in parallel, can handle both Q_{14} and Q_{37}. Since Q_{37}, with a flow of 9 litres/s, corresponds to about the minimum economical pumping capacity, there are unlikely to be any breakdowns of the pump selected to handle the base load.

With this selection of capacities the various operating modes become :

Q = 0 to 30 litres/s: small screw A or B
Q = 30 to 60 litres/s: small screws A + B
or large effluent screw C
Q = 60to 90 litres/s: effluent screws A or B and C
Q = 90 to 120 litres/s: effluent screws A + B + C
Q = 120 to 2200 litres/s: rainwater screw D or E
Q = 2200 to 4250 litres/s: rainwater screws D + E

The water levels are shown in the supply basin at which the various pumps are started are shown in Fig. 27.

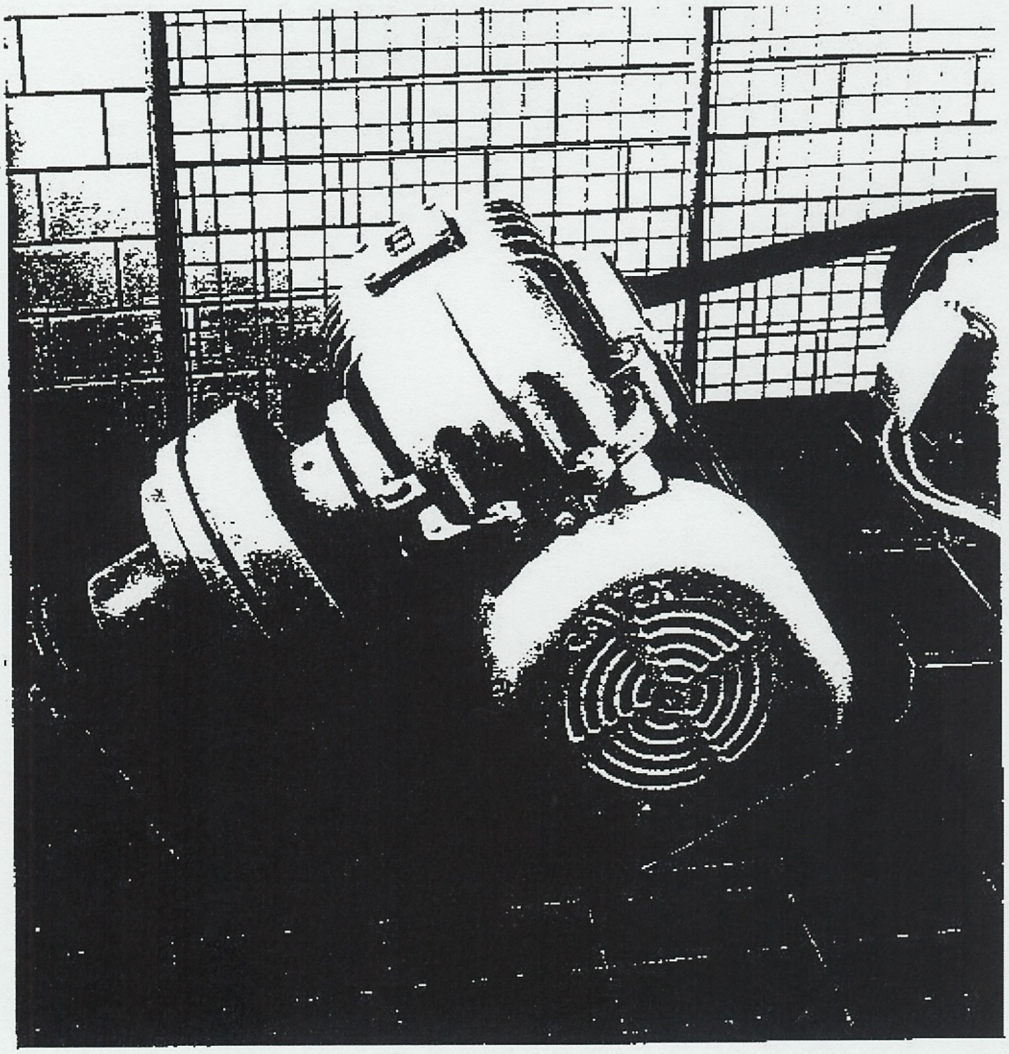
Because of the high standing charge that is made for electricity drawn f. m the public mains supply (approx 12,-- DM kW per month) it is often desirable to operate the two rainwater pumps solely from the emergency generators, which have to be provided in any event. The additional costs that arise are usually compensated for in a period of five to ten years.

The selection of pump sizes for the pumping station that has been described enables all operational requirements to be met. The storage volume in the supply basin required for this installation is calculated from 100 times the capacity of the largest effluent pump (60 litres/s) to be 6,0 m^3. To prevent an excessive cycling rate in the event of heavy rainfall there must be a storage volume of $60 \times 2,2 = 132$ m^3 available in the drainage system and the basin combined, and a check will have to be made from the maps and plans of the drainage system to handle a flow rate of this magnitude and the permissible depths of water in the pipes make it probable that there will in fact be sufficient volume available.

As regards the engineering design of Archimedian screw pumping plants, it should finally be fpointed out that it is advisable for the civil engineer planning the plant not to specify the angle of installation, number of starts, or speed of the pump too closely. A specification that is too detailed can prove very restrictive to the manufacturers when they are preparing their quotations for the machinery, and makes it impossible to compare the advantages and disadvantages of the various possible solutions. It appears advisable, therefore, to establish contact with the manufacturers as soon as the water levels on the supply and discharge sides and the rate at which water will have to be handled are known.

3.10 Reduction Gearing

As ha s already been mentioned, when comparing the disadvantages of Archimedian screw pumps compared with centrifugal pumps, additional devices are required to reduce the speed at which normal electric motors run to a speed suitable for the operation of the pump. In addition to the actual reduction gearing, which is essential, the use of a V-belt drive between the electric motor and the gear-box has proved very successful. With this arrangement it is possible to vary the reduction ratio of the belt drive by a suitable choice of belt pulleys, and this enables the overall reduction ratio to be selected to give the required speed of the screw while using standard gearboxes with a fixed selection of ratios. Another advantage of this arrangement is the smooth starting of the electric motor that results from the elasticity of the belt drive, and as a result geared electric motors arenow seldom used, especially for the higher powers, even though they occupy less space and require smaller foundations. As stated, when a V-belt is used, the reduction of the speed of the motor



30 Drive to screw shaft by means of V-belt and reduction gearing

to the speed of the screw shaft always involves standard reduction ratios which can be achived by using two gear-reduction stages, one comprising a pair of spur gears and the other a pair of bevel gears, or a worm and pinion, to change the direction of the drive; with this arrangement there will be no operational difficulties nor any undue maintenance required. The only work involved is an occasional check on the oil level and an oil change at the intervals recommended in the manufacturer's instructions. Since the frequency at which the oil requires to be changed depends on the type of gearing and the size of the gearbox, no general recommendations as to this can be made.

If the screw pump is used only infrequently, as may be the case with stand-by units or rainwater pumps, the gearbox oil changes must not be regulated solely by the number of hours that the unit has run, since there is then the danger that the lubricating properties of the oil might deteriorate due to ageing. Further information on this point will also be found in the manufacturer's instructions. The selection of the design loads for the gearing depends on the operating conditions, and this point is discussed in more detail in Chapter 3.11, where reference is made to extensive investigations that have been carried out into the starting loads on the transmission elements. It was found that, provided the correct method of starting was adopted, no undue stresses were imposed on the reduction gearing during the starting of Archimedian screw pumps, and that when normal running conditions had been attained the load on the gearing became very uniform, reaching a maximum value when the pump was running at the ideal design point. No higher stresses were imposed on the motor or gearing either if the screw immersion was less than the ideal value or if flooding took place.

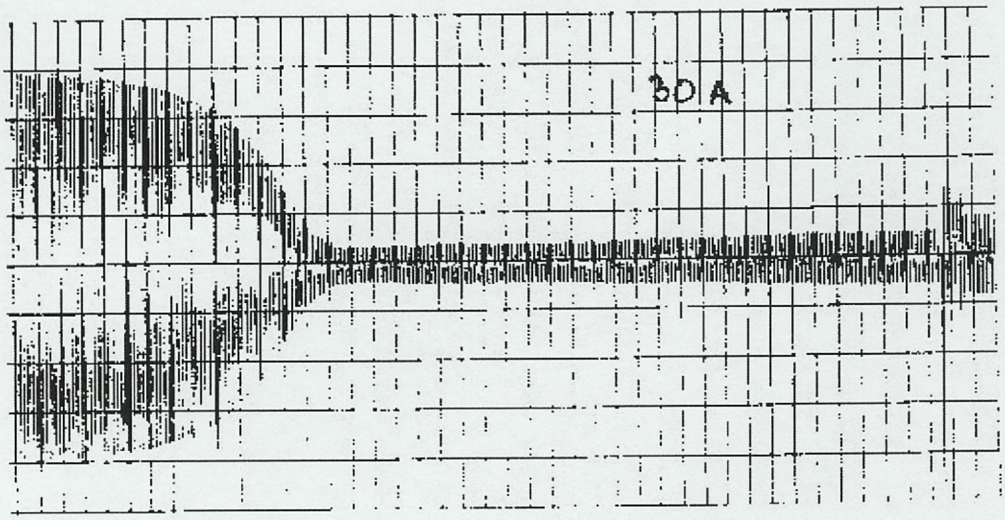
3.11 Propulsion Motors and their working Conditions

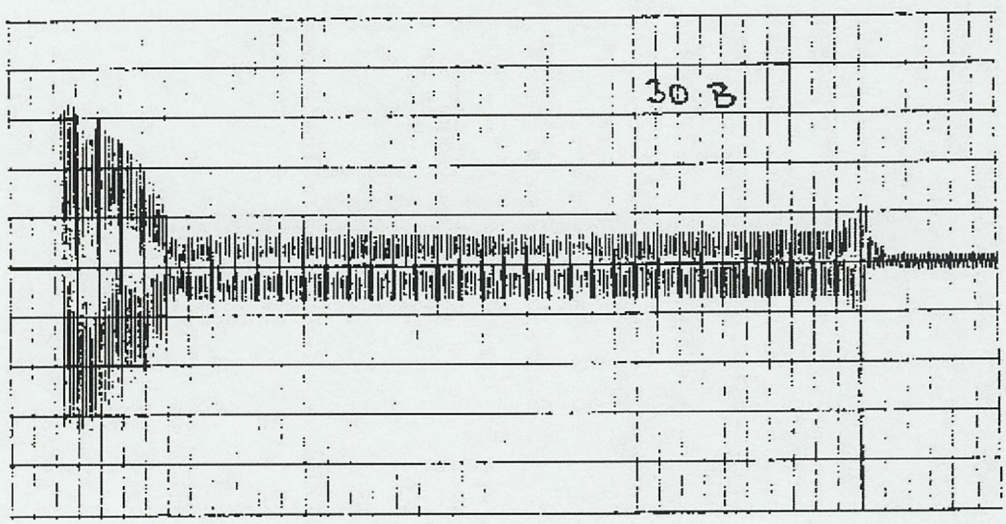
The motors used for driving Archimedian screw pumps are almost exclusively of the three-phase asynchronous type with squirrel-cage rotors (Standard motors to DIN 42 673) or, where higher powers are involved, bar conductor or double squirrel-cage motors are used; these motors will adapt themselves to any loading up to the limiting torque during acceleration. When these three-phase motors are used for driving Archimedian screw pumps, and the normal star-delta starters are employed, experience shows that the load torque arising, based on the total current consumed, lies below the starting torque with an adequate factor of safety, allowance being made for the acceleration of the motor and gearing. The motors should be of Type B 3 and Enclosure Class P 22, or P 33 if required to work under particularly damp conditions, e.g. in an underground plant. When running at their service speed these motors achieve a stable condition in which the torque and back torque can be in balance even under the most adverse conditions. On the conclusion of the startingphase the motor will always strive to return to its service speed if the speed varies from this value. During the star starting phase the low value of the starting voltage ensures that the starting torque that can be achieved is less than the serviced torque, so that there can be no overloading of the power transmission and speed reduction equipment. The Archimedian screw pump also imposes no difficult conditions as regards starting times and starting behaviour on the driving motor. In general, the torque characteristics of normal three-phase motors will meet the starting and operating requirements, since the screw pump can be regarded as a machine that is started under no-load conditions, due to the fact that it does not commence delivery until some time after it has been started. The relatively large masses of the screw shaft which have to be set in motion, require to be accelerated only to a low speed so that, in this respect, the starting torque is also held at a low level. Experience shows that a star running time of 3 to 5 seconds is sufficient to enable the appropriate service speed to be attained and to enable the starting current peak to drop to an acceptable level.

To investigate the loading conditions obtaining during the starting of screw pumps, extensive tests have been made on a number of plants of different capacities. The starting-current records, which were taken by means of electron ray oscillographs, had the values shown in Fig. 30 a - c. The measurements were made on motors equipped with star-delta starters, since this type of starter is used almost exclusively for driving screw pumps owing to the low back torque, and has proved very successfully.

In generl, it will meet the requirements of the power supply authority, and it produces the smooth starting that is required from the point of view of the loads imposed on the gearing and transmission elements.

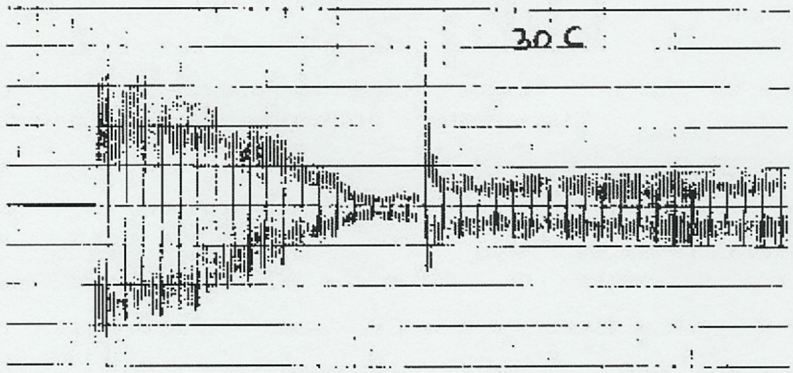
When a star-delta starter is used, the starting current and starting torque are only about 30 % of the values attained when direct-on-line starting is employed. Taking into account the torque required for accelerating the motor itself and the gearing, the starting load is certain not to attain the value of the normal service load provided the ratio of the starting current to the rated service current does not exceed 4 : 1. The following current-time diagrams, in which the envelope curves of the frequency-related instantaneous current values represent the actual current values, are characteristic examples of the current consumption of Archimedian screw pumps during starting and when operating. See Fig. 30 d for details of the various features shown on these diagrams.



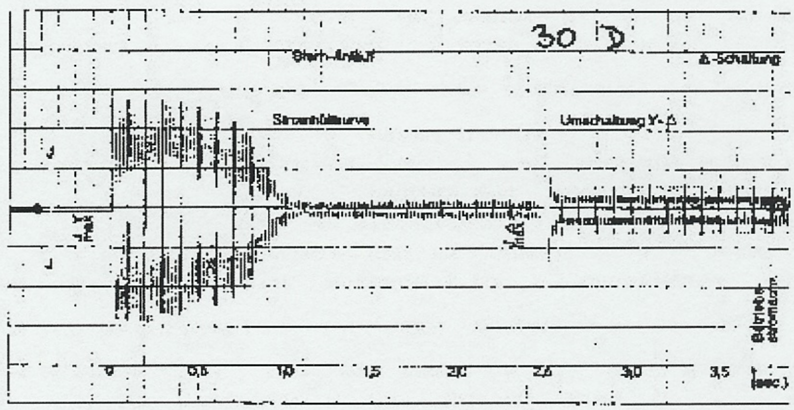


30 a Starting-current curve for the 50 kW 960 rev/min drive of an Archimedian screw pump, 380 V, 99 A, 50 Hz, with star-delta starting

30 b No-load starting current of a 7,5 kW 1440 rev/min thtee-phase motor. 380 V, 15,4 A, 50 Hz, with star-delta starting (with belt pulley)



Stern-Anlauf	=	Star starting
Schaltung	=	Delta operation
Stromhüllkurve	=	Current envelope curve
Umschaltung	=	Star-delta changeover
Betriebsstromaufn.	=	Service current



30 c Starting-current curve for the 90 kW 1480 rev/min drive of an Archimedian screw pump, 380 V, 167 A, 50 Hz, with star-delta starting

30 d Description of the oscillograph traces showing the results obtained during measurements on a 5.5 kW three-phase motor, service current 8,0 A

The results of these starting tests show that the starting current obtaining during the star phase amounts to 2.5 to 3.5 times the service current. Since, with this type of starting the starting torque is only about half the rated torque, these measurements lead to the conclusion that, when starting an Archimedian screw pump in the empty condition, the loading will still be acceptable if the motor and gearing are designed only for the calculated nominal load on the screw shaft. It is therefore not necessary to add any margins to the dimensions of these parts for safety reasons beyond the addition of 10 % recommended in Chapter 2.16 " Calculation of Power required to Drive Pump. " If the selected motor is more powerful than is needed to cope with the screw capacity, this will have no adverse effect on the reduction gearing. As investigations carried out on an installation provided with too powerful a motor have shown (Fig. 30c), with a rated motor current of 99A and a service current of 57A, the ratio of starting current to rated motor current drops to the value 2 : 1.

A comparison of Fig. 30a with 30c leads to the conclusion that, whilst for motors with higher service speeds the starting current is maintained for a slightly longer time than with slower speed motors, there is no change in the ratio of starting current to service current. There is no difference, or at least no significant difference, inthe starting conditions of small and large plants. With properly maintained switchgear and the correct adjustment of the starting time delay on the changeover relay there are also no load peaks when changing over from star to delta operation. The changeover peak, which usually lasts for only a few cycles, is mainly produced by the switchgear, as shown by the no-load starting measurements on a motor that are given in Fig. 30b. Re-starting a screw pump, even if it is equipped with a non-reversing lock, shortly after it has been stopped, and while it is still full of water, must in all circumstances be avoided, since in this case it will be started under load. Screw pumps that are not equipped with a non-reversing lock could suffer damage if they are re-started shortly after being switched off, since in that case the screw, and hence the drive, could be rotating in the reverse direction while draining takes place. The motor and gearing would be particularly susceptible to damage under those conditions.

It will be unusual for the current observed on the ammeter of a screw drive circuit to maintain a steady value. This may be due to the blades striking the water surface in the supply or the discharge troughs, or both. Out-of-balance forces or excessive deflection of the screw shaft can, however, also be the cause of such current fluctuations which occur while the pump is running steadily. As the delivery head increases the ammeter readings will be smoothed out unless the fluctuations are due to out-of-balance forces or excessive shaft deflection.

3.12 Switchgear and Control Equipment for Screw-Pump Pumping Stations

In VDE 0660 switchgear is defined as equipment which connects, interrupts, or isolates current paths. If these appliances are provided with auxiliary devices either built in or built on they can also undertake protective duties, and will then switch-off all phases in the event of a fault by means of a thermal or magnetic excess-current trip. Switches are used for switching current paths on or off, or for changeover purposes, and can be operated either under human control or automatically; usually they form part of an item of switch-gear.

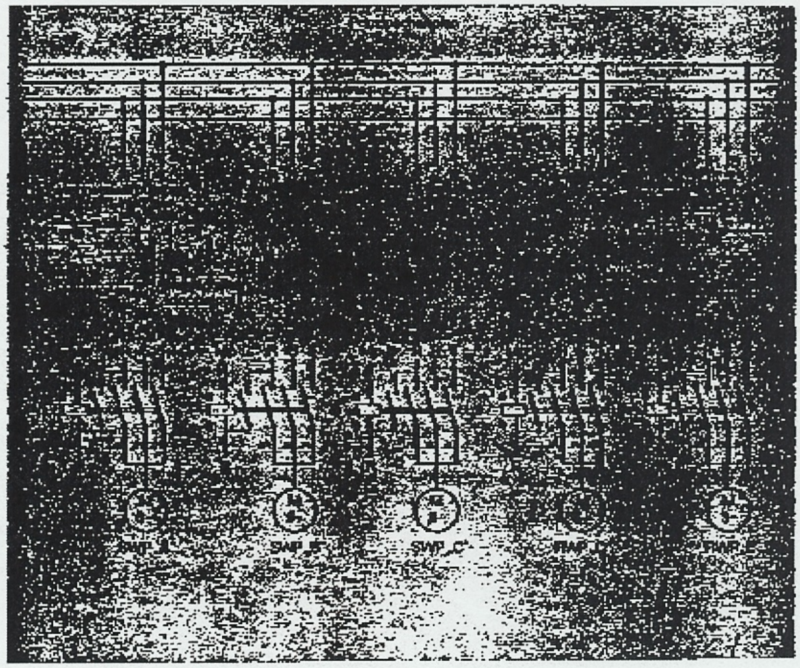
Starters bring electrical appliances, and in particular motors, into the running condition, while controllers can vary the running condition.

Another item included in the electrical equipment of pumping stations is the fuse which will open a circuit automatically in the event of an overload or a short circuit due to the melting of the fusible link through which the current passes.

In addition to this electrical equipment there are the measuring instruments. Even the smaller plants should be provided with a volt-meter, a phase selector, a frequency meter and ammeters measuring the currents in all phases. It is essential that an electrical elapsed-time counter be fitted to each screw pump that is installed, since the records these instruments provide can be extremely useful. The elapsed-time records give an indication of the quantities of water pumped, and enable the approximate efficiency of an installation to be calculated; this is done by calculating the quantity of water pumped during the time the pump has been running, and dividing this by the recorded power consumption in kWh and so obtaining very simply the power required to pump 1 m^3 of water.

When the correct choice of switchgear and measuring instruments has been made, depending on the purpose of the plant, the amount of work it is to perform and the mains supply, these are mounted in suitable switch cabinets or, where very damp conditions prevail, in cast-iron boxes, and then are wired neatly and clearly. A circuit diagram will prove of great assistance when tracing faults, and should be provided for every installation as a matter of course. Fig. 31 shows an example of such a circuit diagram in the form of a schematic diagram for a pumping station containing five Archimedian screw pumps which is, in fact, the station to illustrate the allocation of the pump capacities. Water-level switches of various types are used almost exclusively for the automatic starting and stopping of Archimedian screw pumps. Reference is therefore made to water-level regulation by means of water-level controllers even though these act, not in a loop, but in an open chain so that, strictly speaking, it is a straight, water-level control system.

Prinzipschaltplan	=	Schematic circuit diagram
Schmutzwasserpumpen	=	Effluent pumps
Regenwasserpumpen	=	Rainwater pumps
Wahlschlater (doppelpol)	=	Selector switch (two-pole)
SWP	=	Eff. P.
RWP	=	Rw. P.



31 Schematic circuit diagram of a pumping station with five pump units.

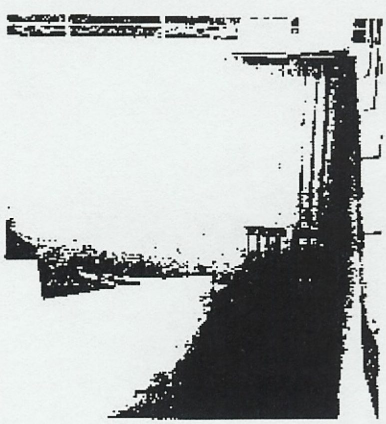
Suitable switches for starting and stopping screw pumps with variance in the water level include float-operated switches, immersed-contact switches, electrode switches, float-operated tilting switches and, more recently, an air-pressure system employing diaphragm type pressure pick-ups (manufacturers: Rittmaier/Schwyz and Huber/Bad Tolz). With a small number of pump units the electrode switch can conveniently be accommodated in the supply basin in the form of rigid elements whilst, where a large number of pump units are involved, the air-pressure method mentioned above has proved satisfactory. Where electrodes are used it is most strongly recommended that they be protected against any clogging material that could interfere with their operation, since this could short-circuit the electrodes and so keep the pumps running continuously. If the level of water in the supply basin is also to be indicated or recorded, it is advisable to use a float-operated switbh using a resistance, current, or voltage balancing method. The float is most conveniently arranged in a protective tube with a screen at the bottom.

In general, the control unit should be backed-up with an amplifier to amplify the signal from the control unit to a level suitable for operating the actuator concerned.

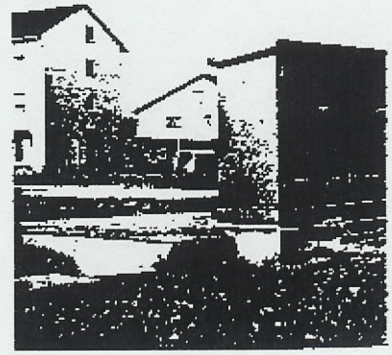
One of the methods of arranging electrode-type water level indicators is shown in Fig. 32.

It is not possible to make any general recommendations regarding the

32 Water-level control of Archimedian screw pumps by means of an electrode system



33 Open-air distribution cabinet for housing the controls and instruments for an underground intermediate pumping station.



housing of the electrical equipment for Archimedian screw pumps. It will be necessary to decide, in each individual case whether a modern distribution cabinet, a weather-proof hut (which can also be used to accommodate the motors and gearing) or a simple open-air distribution system as shown in Fig. 33 should be used.

3. 13 Emergency Power Supply in the Event of Power Failure

If there is a power failure in the mains supply such that the pumping stations cannot operate, considerable difficulties can be caused in the drainage system due to the resultant build-up of water and flooding. The fact that heavy rain and power failures due to lightning strikes often coincide it makes it essential to provide an emergency power supply for drainage pumping stations. Industry has developed fully-automatic emergency generator sets for this type of work; these require no supervision or manual intervention, and are capable of supplying current to drive the pumps within a few seconds of the failure of the mains supply, and to keep the pumps working until the mains supply is restored in all phases, when they will automatically switch themselves off. Since it is undesirable for the pumps to be started whilst the emergency generator is starting and running up to speed, a delay timer is fitted.

The automatic control gear will start the emergency generator even if there is a failure in only one phase of the mains supply, and will keep it running until the mains supply is fully restored. In the case of the larger units, a Diesel engine is almost invariably used to drive the direct-coupled generator, this type of engine having proved reliable, economic, and capable of maintaining the frequency constant under a wide range of con ditions. Engines rated at up to about 250 h.p. are started by a battery, whilst larger engines are started by means of compressed air.

Air-cooling is normally preferred because of the absence of maintenance. It is only when the power ratings exceed about 200 h.p. that air cooling proves inadequate, and then liquid-cooled engines are employed. Fig. 34 shows the appearance of a modern fully-automatic emergency generator unit for a pumping station.

The supply of Diesel fuel stored on site should be enough to keep the plant running for at least three days in an emergency.

Since the fuel consumption of a Diesel engine is about 180-200 g/h.p.-h, this means that the daily fuel consumption will amount to about 5 litres/h.p. of installed power, or 15 litres/h.p. to provide a 3-day emergency supply as recommended above. For a medium-sized emergency generator set of 100 h.p., which is a size frequently installed in pumping stations, a total bunker capacity of 1500 litres of Diesel fuel would therefore be required.

To ensure that the emergency generator set is maintained in good working order it should be started every two weeks and run for at least 15 minutes under no-load conditions, and subjected to a trial run on load once every two months. It is then advisable to start the set by simulating a power failure, e.g. by removing a fuse from the main power supply, etc. This will then, at the same time act as a check on the automatic starting mechanism. Although the voltage, current, and frequency can be measured using the same instruments as when the pumps are driven from mains supply, the emergency generator should be equipped with a separate elapsed-time counter. This can then be used to check that the trial starts have been performed correctly.

It will be necessary to investigate separately, in each particular case, the extent to which the emergency generators can be used economically to cover peak power demands on the plant, thereby reducing the standing charge raised by the power company. It will, in any event, prove necessary to negotiate with the power company since their approval is needed if an emergency generator or private power supply is to be installed.

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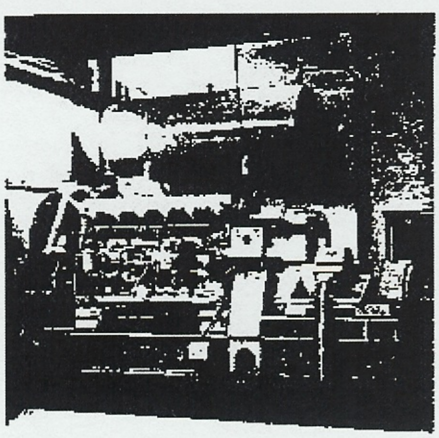


34 Fully-automatic emergency Diesel-generator set for a pumping station.

3.14 Measures for Obtaining Satisfactory Gap Values

The various methods of producing the screw trough have already been discussed in Chapter 3.3. This also included details as to the way in which the correct gap can be obtained.

As is well know, it is almost inevitable that a structure will undergo a certain amount of settlement after it has been completed, and in the case of Archimedian screw pumping stations this may make it necessary to modify the screw trough. This can also be the case if t..e surface produced is too rough or irregular. In this connection it should be pointed out that the gap width and the roughness of the trough surface, and in particular lack of uniformity between the outer edges of the screw blades and the trough can have a major influence on the efficiency that is attained.

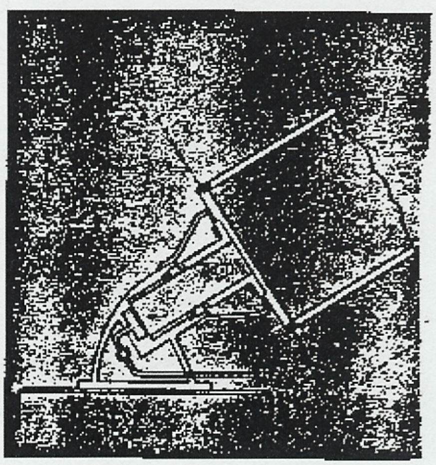
There are nowdays many plastics products available which appear suitable for lining the trough. Many of these, however, must be rejected because of the hardness which they develop after processing, and the consequent brittleness and lack of resistance to impact loads. In addition to good adherence to the base material, e.g. concrete, a material for lining the troughs must have a degree of stretch and flexibility, and must also be resistant to wear and to damage by dilute acids, alkalis, and biological attack etc., since any of these may be encountered in the effl. Even though there is no evidence to show that it is desirable to line screw troughs completely, it appears advisable to find a suitable material that can be used for repairing rough and damaged sections of troughs. One possibility would be the application of self-vulcanising anti-corrosive coatings based on neoprene or hypalon. These materials are resistant to wear and to mechanical impact, they adhere well to concrete, wood, or metal sufaces, and are not attacked by mildew, acids, alkalis, salts, or chemicals of various types and are also resistant to weathering. If applied in large areas the film, which can be built up to any desired thickness, will also show no ageing cracks even after a lenghty period. Another advantage of these preparations is the ease with which they can be handled on site. Their relatively high price will, however, make them unsuitable for general use for lining troughs as a matter of routine. The extent to which the lining of troughs would prove to be an economical process, because of the increase in useful life that results cannot be determined with certainty until considerable service experience has been gained. The effects of temperature on the diameter and the gap have bee mentioned in Chapter 3/3.2.

3.15 Lubrication of Screw Bearings

Even though the lower bearing of the screw, which is the most heavily loaded is nowadays almost invariably of enclosed water-tight design, it will still require special care and attention. This lower bearing is constantly immersed in contaminated liquid, which would damage the bearing surfaces if it could penetrate to them. The enclosure of the bearing, which is adopted primarily with this end in view, serves the additional purpose of keeping clogging materials away from the short length of screw shaft between the screw and the bearing. In the case of the smaller screws, any material that becomes wound around the shaft can considerably restrict the inflow of water to the screw, and its removal is a task which not only takes considerable time and effort on the part of the staff, but one which also has to be repeated at frequent intervals.

If grease is supplied to the lower screw bearing, by means of an automatic grease gun secured to the gearbox whilst the pump is operating, the enclosed space will fill with grease and this will provide adequate protection against contaminated water and other materials penetrating to the bearing.

A grease line carries the lubricant from the grease gun to the lower bearing (which is less highly loaded and less liable to suffer damage)



35 Illustration of an enclosed lower screw bearing.

is not connected to the grease gun as a matter of policy, but instead is lubricated at regular intervals by turning the hand lubricator during normal inspection. This prevents the greater part of the grease being supplied to the upper bearing, which is closer to the grease gun, to the detriment of the lower and more heavily loaded bearing. This arrangement has proved successful in recent years, so that there is no reason to make any change. Fig. 35 shows a sketch of an enclosed lower screw bearing.

Details of the type and size of the bearings will not be discussed here. It should be pointed out, however, that the values given in the tables enable the loads to which the bearings will be subjected to be predicted without difficulty. Since these tables also give the maximum speeds of rotation of the screws, it should be possible to determine the sizes and types of bearings required from this information.

3.16 Corrosion and Its Prevention

One of the subjects that arises most frequently, when the advantages and disadvantages of Archidedian screw pumps are discussed, is the question whether the metallic parts of the pumps require special protection against corrosion. The fact that domestic wastes usually have a fairly high fat content suggests that no special anti-corrosion measures are necessary. The practice of galvanising the whole of the surface of the screw, which has been derived from the experiences gained in pumping waters from land reclamation schemes, which have a high humic acid content, and salt seawater, has not been found absolutely essential when handling normal effluents. The non-porous zinc layer, which is applied by means of a special process, would of course act as a sacrificial metal and hence protect the underlying steel surface, just as the galvanised pipe has proved effective in household water systems. However, once the zinc coating has suffered damage, an electro-element is formed, and the resultant electrolysis will cause the screw components to be damaged. Experience shows that it is hopeless to attempt to repait damage of this type, since the corrosion immediately spreads under the adjacent zinc layer and it continues its destruction unchecked. In view of this one can conclude that it is, in general, not worth while to undergo the comparatively heavy expense of zinc-coating the metallic parts of an Archimedian screw pump if this is to be used for handling normal municipal effluents.

The application of an anti-corrosive coating to the metallic parts of the screw pump is adopted more for the sake of improving the appearance than for increasing the durability and life. It does, however, have the major advantage that any damage to the doating can be repaired without difficulty by the normal maintenance personnel, so that a reasonable life can be achieved if the paint is correctly selected. In this connection special attention must be paid to achieving good adherence to the metal surface, and also to ensuring that the paint retains its toughness after curing. Resistance to sunlight and to damage by impact is also of importance. It will not be possible to make any final recommendations on this matter until further experience has been gained over the course of a number of years. Further extensive investigations are also needed as to whether coating the screw surface with rubber will prove effective both in reducing corrosion and also improving the operating characteristics. If the screws are painted, light colours should be chosen, since these reduce the thermal effects of the sun shining on the screw.

3.17 Abrasive Action and Damage due to Solid Matter

Whereas, in the caseof centrifugal pumps, sand and other solid materials often cause breakdowns and rapid wear these difficulties are unknown with Archimedian screw pumps, as is cavitation. As a result of their low weight when submerged in water, sand and stones are largely held in suspension in the flow and so will damage neither the screw blades nor the trough.

If pieces of wood or other bulky items, which are larger than the space between the screw blades, enter the supply basin, the low speed of the screw causes this flotsam to be pushed back by the leading edges of the blades, and this is unlikely to cause any appreciable damage. It is important, however, to ensure that the bottom edges of the blades are flush with the end of the trough, to prevent any danger of any material becoming jammed between the two.

If the level of water in the supply basin is high, however, so that the screw is deeply immersed, there is a danger that floating objects, and especially long pieces of wood such as roofing laths, etc., can become jammed between the outer edges of the rotating screw blades and the trough. This difficulty is usually avoided by fitting deflection plates. An example of such plates is shown in Fig. 36. The extensions to the sides of the troughs seen on the left-hand side of the screw pumps increase the amount of water drawn into the chambers formed between the screw blades, while the deflection plates are visible on the right.

Experience in recent years with a large number of pumping stations has shown that if Archimedian screw pumps are carefully designed and correctly installed, taking the various points mentioned into account, they will give reliable and economical service for many years.

36 Extensions to sides of troughs.
Plates for deflecting solid material liable to cause jamming on the right, and plates to increase volume in chambers on the left.

3.18 Maintenance of Screw Pump Installations and Auxiliaries

The fact that Archimedian screw pumps require very little maintenance has been mentioned on several occasions. Where it proves necessary to undertake frequent maintenance the cause is almost always found to be a design or a constructional error, such as the need to remove material which has become wound around the exposed screw shaft between the ends of the blades and the bearings, and which can easily be avoided. If major sludge deposits form in the supply basin, the bottom of the basin is not correctly shaped. Another frequent cause of trouble, which could be eliminated at small expense, occurs when materials wind themselves round the control electrodes, as a result of which the pumps are kept running permanently.

If repairs are needed, the bearings can be removed without it being necessary to remove the heavy screw itself, provided they are designed and fitted in accordance with the latest proposals. It is then merely necessary to raise the screw slightly and to support it.

The supply of lubricant to the screw shaft bearings makes no particular demands on the frequency at which maintenance is performed. Weekly inspection and servicing of intermediate pumping stations is regarded as adequate. Mention has already been made of the need to ensure that the emergency generating plant is at all times ready for use.

The main factor governing the serviceability and reliability of the pumping plant is undoubtedly the electrical installation. It is most unlikely that the plant attendants will be skilled electricians, and so it is advisable to arrange a maintenance contract with a firm of electrical contractors who have had experience in this field. This should provide for a careful check being made on the correct operation of the switchgear, controls, and instruments, as well as the electric motors, the generator, and the automatic starting equipment for the emergency generator at intervals of not more than a year.

If the attendant enters all the operational data (running times, electricity meter readings) and miscellaneous matter (trials, faults, causes of breakdowns, etc) into his log at least once a week, when he makes his tour of inspection, one can be sure that the plant will run satisfactorily for long periods.

3.19 Cost Estimates, Forecast of Working Life, and Depreciation Data

The following assumptions can be made when estimating the costs of Archimedian screw pumps of medium size:

Cost of structure, including trough	= 300,- DM/m^3 of encompassed space
Screw pump with bearings	= 2.- DM/kg weight
Drive motor	= 80.- DM/h.p.
Gearing and transmission	= 60.- DM/h.p.
Switchgear and controls	= 100.- DM/h.p.
Emergency generator and fuel storage	= 500.- DM/h.p. output.

These standard prices are based on 1965 price levels, and should be modified where necessary by the current building cost index. For high-capacity plants the standard costs quoted will be reduced by up to about 30 %, whilst for small units it will be necessary to calculate the costs in each particular case, and the cost levels are likely to be appreciably higher.

The prices quoted will also vary according to local conditions and so can be regarded as only very approximate. They should be used only for producing an initial estimate of the probable cost of erecting a pumping plant equipped with Archimedian screw pumps.

If the average service lives for water supply equipment as laid down by the Magdeburg Rules are applied to Archimedian screw pumping installations, making allowance for the special features of sewage pumping service, the following assumed lives and depreciation rates will be obtained.

	Life in yeats	Depreciation rate in %
Buildings and structural part of pumping installation	40 - 50	2.0 - 2.5
Connections to drainage system, supply basin, and discharge connections	30 - 40	2.6 - 3.3
Screw pump and mechanical components including gearing	15 - 25	4.0 - 6.6
Motors and switchgear	15 - 25	4.0 - 6.6
Emergency generator and accessories	25 - 30	3.3 - 4.0
Other equipment, tools, and auxiliaries	10 - 20	5.0 -10.0

Maintenance costs will depend largely on the frequency with which repairs are required. In view of the comparatively robust machinery parts, and the use of well-tried auxiliary equipment it may be assumed thet a yearly sum of 0.8 % of the initial cost of the machinery and electrical equipment will be more than adequate to cover the maintenance costs.

The actual running costs, which are not identical with the commercial definition of "running expenses", but cover merely the actual outlays on items such as electric power and consumable stores can be based, for a rough estimate, on the figure of 0.005 kWh/m^3 + m, per m^3 of water pumped and per m delivery head. If, for example in the case considered earlier, the town of 6000 inhabitants requires 36 000 m^3 of effluent to be pumped per month over a height of 5.0 m, the electrical power consumed in this period will be:

$$36\,000 \text{ m}^3 \;.\; 5.0 \text{ m} \;.\; 005 \text{ kWh/m}^3 + \text{m} = 900 \text{ kWh.}$$

In addition, there will be the standing charge (which varies from district to district), reactive power costs and meter rents, a figure of 15.- DM per kW of installed power representing a reasonable average for the total of all these items.

Lubricants, cleaning materials, and other consumable stores will be covered by a figure of 2.- DM per m^3 of effluent that is pumped.

By summing the relevant individual figures for the various items refferred to above it is possible to calculate the direct costs of a pumping station, to which a sufficient margin will have to be added to cover the user's own costs.

4.1 The significance of Measurements on Archimedian Screw Pumps

The operational economy of a pumping plant depends upon the initial cost, maintenance and repair costs, reliability and, in particular, upon the efficiency of the pump itself.

When acquiring a pumping installation the user will expect the manufacturer to provide him not only with an estimate of the costs, but also with the efficiency that the plant is expected to attain. The efficiency of a pump is the ratio of the power delivered to the pump to the work performed in raising the water, which is the product of the delivery rate and the head.

The overall efficiency η_{ges} for an Archimedian screw pump, given in Chapter 2.15 " Determination of Efficiency " allows for all losses that occur after the drive shaft coupling, and thus also takes into account those losses due to the essential reduction in speed by means of the V-belt transmission and the reduction gearing. It thus corresponds to the pump efficiency η_p, normally quoted for other types of pump.

Any guarantee of performance is valueless unless the latter can be checked. Unfortunately it is not practicable for the manufacturer of an Archimedian screw pump to make the necessary measurements on a test bed and so, with such pumps (unlike centrifugal pumps) it is necessary to rely solely on measurements made when the plant is finally installed. It is only if such measurements are made that it is possible to produce an adequate counter to the claims for excessively high efficiencies that sometimes form the basis of unfair competition.

To indicate the economic importance of efficiency claims, and the necessity of checking them by making the appropriate measurements it may be pointed out that, for the efficiency range of 70 to 80 %, which is normal for Archimedian screw pumps, a change in efficiency of 1 % will on average, require about 0.05 kW_h – increase or decrease – for a delivery of 1000 m^3 of water and an effective delivery head of 1 m.

Reverting to the example in Chapter 3/3.4, dealing with economic calculations, the additional power required per day for the delivery of 3450 m^3/day at a head of 4.0 metres would amount, for a reduction in efficiency of 1.7 %, to:

$$\frac{3450 \, . \, 4.0 \, . \, 0.05}{1000} \, . \, 1.7 = 1.20 \text{ kWh/day.}$$

This corresponds with the increase in power quoted in the example which was, however, determined in a different way.

4.2 Determination of Delivery

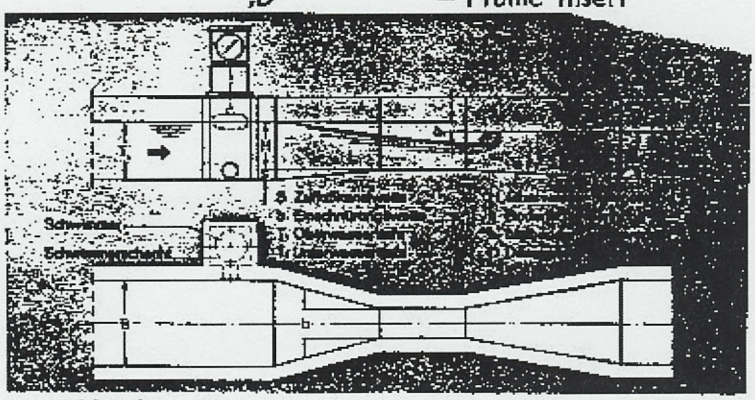
Whereas the installation of reliable flow meters such as water meters or venturi meters is taken as a matter of course, when fresh water is being handled, there are still various difficulties encountered in the measurement of effluent flows. Because of the danger of contamination and blockage it is not practicable to use venturi tubes, inserts, or orifice plates, at least with untreated effluent, unless the measuring section is to be flushed out continuously with water under pressure, to prevent the settlement of foreign bodies. Since it is forbidden, for hygienic reasons, to connect a water mains to the effluent-measuring instruments, the flushing arrangments involve a considerable outlay. Where it is necessary to measure the flow of effluents delivered by an Archimedian screw pump, a measuring instrument of this type is therefore not a practical proposition, especially since there are other reasons why it is possible to dispense not only with the need to clean the effluent being measured, but even with computing devices.

For a permanent flow-measuring instrument used in municipal effluent systems the only suitable type is therefore the venturi flume, or the PARSHAL flow metering channel, which works onthe same principle. This method of measuring the flow of liquids in open channels (free-surface channels) depeds, in all ofthe known variants, on introducing a construction into the channel at a given point, and thereby increasing the flow velocity at the point. This change in velocity is, in turn, accompanied by a change in the level of the fluid. The measuring section comprises the approach section, the inflow, the constriction, and the discharge section. If a reliable measurement is to be obtained it is essential that the depth of water on the inflow side depends only on the quantity of water flowing through the instrument, and not on how high or low the water level is behind the constriction, this latter depending on the discharge conditions, and not on the measurement itself. The relationship between the depth of water upstream of the flume, H, and the flow, Q, is then a parabola, the equation for which is $Q = C \cdot H^{2/3}$, where C is a coefficient that depends on the form and size of the channel concerned. The value of this coefficient is allowed for when the instruments are being calibrated.

One of the main advantages of this method of measurment is that any impurities in the flow, including any solid bodies, will have no adverse effect on the reliabity of the measurements. Also, apart from a relatively short approach section, the fluid can be led to, and discharged from, the measuring section via a closed pipe. An arrangement for a channel flow measuring installation is hown in Fig. 37.

The PARSHAL flow metering channel can be regarded as a geometrically improved form of the conventional venturi flume. It offers a number of advantages. For example, it is of particularly short length, is very simple install, and its range of measurement

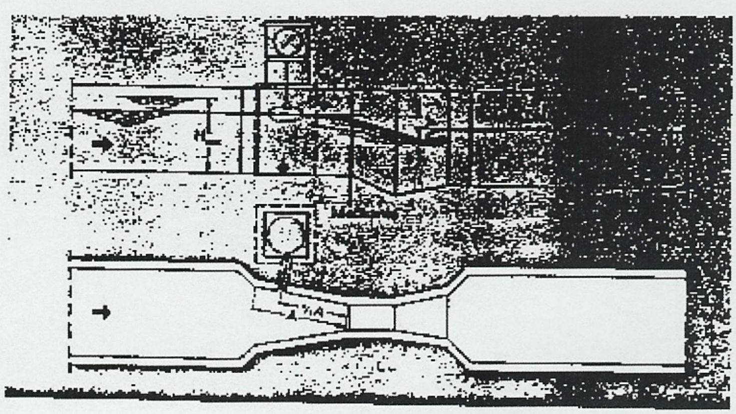
Schwimmer	= Float
Schwimmerschacht	= Float shaft
B	= Breadth of approach channel
b	= breadth of constriction
T_1	= Upstream depth
T_2	= Downstream depth
H	= Upstream depth above step
s	= Height of step
h_v	= Permanent loss of head
D_v	= Flume insert



Messkante = Edge of measuring section

37 General arrangement of channel flow measuring installation

38 Arrangement of a PARSHAL flow metering channel



(together with the ease with which this can be varied and the high accuracy of the results obtained) are of advantage in comparison with a channel flow-measuring installation. The bottom slopes downward at the constriction, and this ensures that the flow velocity is supercritical even at very low rates of flow. As a result, this is the only known instrument of this type which enables linear flow measurements to be made down to flow rates of less than 1 % of the maximum flow.

Standardisation has now progressed to the stage where prefabricated measuring sections made of glass-fibre reinforced polyester are available for certain channel sizes. Recent water-conservation regulations make it, in many cases, necessary to measure the effluent flows as a basis for further calculations. At present the regulatory requirements are usually restrictred to knowledge of the quantities discharged to the main drain, and the measurement of the quantities passing through treatment plants. In general, it is therefore extremely unlikely that a flowmeter will be permantently installed either immediately before or immediately after an Archimedian screw effluent pump. It will then be necessary to attempt to measure the delivery rate for the screw pump in a different way.

Very often the supply channel will be of suffieient length to make it possible to build a temporary dam in which either a rectangular or triangular weir can be fitted, depending on the rate of flow. Provision should be made for this, during the construction of the supply channel, in the form of grooves provided in the walls into which the dam section can leter be slid. If this is not done, it is possible to secure the dam by means of angle iron.

Since it is necessary to smooth any excessive turbulence between the supply pipe and the weir, this method of measurement is suitable for flows of up to about 50 litres/s where a triangular weir is used, or up to about 800 litres/s where a rectangular weir is used. For thigher flow rates it would be necessary to adopt the bottom discharge method of measurement. Reference should be made to the appropriate DIN Standards for the shapes of the weirs and the design of the edges. Part 5 Tables and Diagrams includes flow rate diagrams for both types of weir. (Diagrams 2 and 3)

If the provision of even a temporary measuring device such as this is not possible, or is too expensive, it is at least possible to obtain approximate results from the depths of water or the flow rates in the channels, provided that the depth of water and the slope are known, or that it is possible to measure the water velocity between two points either by timing the passage of loats, or by injecting a dye into the water. For preference, both methods should be used and the results compared.

For both methods of measurements: $Q = v \cdot F$

the velocity v being either the measured velocity or the calculated value

$$v = \frac{100 \cdot \sqrt{R}}{b + \sqrt{R}} \cdot \sqrt{R \cdot J} \; (m/s)$$

$$\text{Here } R = \frac{F}{U} = \frac{\text{cross-section of water } (m^2)}{\text{wetted circumference } (m)}$$

$$\text{and } J = \frac{h}{l} = \frac{\text{Difference in height of water level (m)}}{\text{Length of measuring section (m)}}$$

while, according to Kutter, the roughness coefficient is b = 0.35.

It is also often possible to determine the quantity of water flowing to the pump by measuring the rise in the water level in the supply basing over a given time, during which the pump is not running, provided a valve is fitted near the inlet, and the supply pipe has sufficient capacity to keep the pressure constant over the time that the measurements are taken.

Another possible method for obtainig at least an approximate method for the efficiency of the pump installation is to shut the inlet valves to a pipe, which is full and whose capacity is accurately known, and to measure the time required to pump asway this fluid and the electrical energy consumed. An example of this method is shown in Section 4.6.

4.3 Establishing the Effective Delivery Head

To enable the value H of the delivery head of an Archimedian screw pump to be determined for the delivery caclulation, 1the water level at the bottom of the screw should be compared with the water level above the spill point. If the heights of these two levels are known, it is merely necessary to subtract the former from the latter as to obtain the effective delivery head.

s_{sp} = 2,5 to 3,5 mm (measured values), and with the screws rotating at 70 rev/min, the blading extending over a length L = 8,60 metres. The driving motor is a squirrel-cage, three-phase type with an output of 3,5 kW at 1440 rev/min. Speed reduction is effected by a V-belt transmission of ratio 540/240 = 2,25 : 1, and a reduction gear having a ratio of 640/70 = 9,1 : 1. The screw is fitted with a non-reversing lock, and the trough is made of concrete.

In the absence of a permanently installed flowmeter, a dam was installed in the supply basin , and this was equipped with a triangular weir (Thompson weir). Since a long supply pipe could be used as a preliminary storage area, and a valve which could be used both as a stop valve and throttling valve was installed in the pipe immediately before it emptied into the basin, there was no difficulty in regulating the inflow to any value required within the capacity range of the pumps.

It was first necessary to establish the reference heights for the delivery head. The walls surrounding the underground pumping station are located at 105,32 m above datum, while the centre of the screw at the bottom end of the blading was found to be at 100,18 m above datum, and the spill point at 104,11 m above datum. The intersection of the water level in the supply basin with the reference line was measured by a light beam, while the height of the water level in the discharge basin,

Table 1 Measured results and calculated values from investigations on Archimedian screw pumps for effluents.

Unterwasserspiegel = Supply basin level (above datum)
Oberwasserspiegel = Discharge basin level (above datum)
U/min = rev/min

Unter-wasser-spiegel (+ NN)	Ober-wasser-spiegel (+ NN)	H_{eff} (m)	Q (l/s)	n_M/n_S (U/min)	N_e (kW)	V	A	N_M (kW)	η_M (%)	η_{S+G} (%)
100,31	104,33	4,02	58,5	1449/72	2,70	395	9,4	4,31	85	73,0
100,28	104,32	4,08	54,1	1448/72	2,15	395	7,9	3,57	84	71,6
100,22	104,32	4,10	41,6	1450/72	1,67	395	6,4	2,87	83	70,1
100,19	104,31	4,12	32,0	1451/72	1,30	395	5,3	2,32	82	68,4
100,15	104,30	4,15	24,5	1452/72	1,00	395	4,5	1,90	81	[illegible]
[illegible]	104,30	4,19	17,6	1453/72	0,73	395	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	4,20	[illegible]	[illegible]	0,46	395	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	4,22	[illegible]	[illegible]	[illegible]	395	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

above the spill point, was determined by means of a measuring rod. The wattmeter was connected to all three phases, and a multiple instrument used for measuring the currents and voltages was connected to one phase when used as an ammeter, the voltages being measured between phases. As a check on the power consumption, the armature revolutions of the meter installed by the power company were measured. The results obtained are listed in Table 1.

To check the measured values and the values of the efficiency that had been calculated from them, a further experiment was performed in which the supply pipe, which had a length of 3428 metres, was filled with water. Since thispipe was of circular section with an internal diameter of 1100 mm, the volume of water involved was 3257 m^3.

Using the valve to control the flow, an attempt was then made to regulate the inflow so that the immersion of the bottom end of the screw swas kept at its optimum value; because of the reduction in pressure in the pipe this was, however, not possible for the whole of the period that the pump was operating. It may be assumed, however, that the conditions approximated to those obtained in practice, although the pump is run continuously rather than cycled. The screws were sometimes too deeply immersed.

If the mean delivery head is taken (from the results obtained in the earlier tests) as 3,95 metres, the results of this test become:

Meter reading after pumping basin empty	= 1 791,93 kWh
Meter reading when pumping commenced	= 1 739,52 kWh
Electric power consumed	= 62,41 kWh
or 62,41 . 367,2 mt/kWh	= 22,910 mt

Effective work performed in raising water from volume and delivery head:

= 3257 . 3,95 = 12 865 mt.

Making allowance for motor efficiency it is possible to calculate an average efficiency for a screw pump from these figures, as follows:

$$\eta_{ges} = \frac{\text{Work performed in raising water}}{\text{Electric power consumed} \cdot \eta_{Motor}} = \frac{12865}{22910 \;.\; 0,82} = 0,684 = 68,4\ \%$$

The following conclusions can be drawn from the measurements and observations performed on this plant:

1. Even if it is run dry for several hours, the screw pump suffers no damage from idling.

2. The power consumption of the whole plant, when idling, amounts to the very low figure of 0,92 kW and, even after an experimental dry run of 4 hours duration this did not increase. There was no temperature rise in the bearings.

3. The lubricant consumption was checked over a lengthy period of operation. It amounted to 290 grams over a period of 210 hours of running, or 1,4 g/h.

4. Leakage losses were checked by determining the time for the pump to empty after it was stopped. The theoretical capacity of the chamber formed by two adjacent blades is 20,62 litres. Over the bladed length of the screw of 8,62 metres there are forty-three such chambers, so that the total capacity amounts to about 888 litres. After the screw was stopped the time taken for it to empty completeley amounted to 212,4 s. From this the leakage flow can be calculated as

$$Q_l = \frac{888\ \mathrm{l}}{212{,}4\ \mathrm{s}} = 4{,}19\ \text{litres/s}\ (= 6\ \%\ \text{of delivery rate}).$$

The theoretical value for the ideal gap width of 3,0 mm is 3,79 litres/s (= 5,4 %). The measured value can therefore be considered satisfactory.

5. The time taken to empty the entire contents of the supply pipe using both pumps amounted to 7 h 3 min 20 s. From this the delivery rate of the individual screws is calculated to be 64,2 litres/s, compared with the design value of 70 litres/s.

6. The results obtained during the course of the performance tests justify the assumption that the efficiency of an Archimedian screw pump remains high over a wide performance range.

7. At the time the tests were made the plant had been in operation for more than two years. There were no indications of abrasive wear or corrosion.

8. Since the pumps are frequently operated only at part load, the motors draw a high reactive current. The mean power factor (cos phi), measured over a lengthy period of service amounted to 0,68. It is recommended that power factor correctors be fitted, since there is more than one reactive kWh for every active kWh. If reactive power costs are to be avoided, efforts should be made to correct the power factor to cos phi = 0,9.

9. The medium being pumped consists of medium to heavily contaminated effluents taken from a separate sewerage system. No screens or sand traps are fitted. Despite this, however, no difficulties have been encountered to date. Difficulties due to materials becoming wound round the screw have been avoided by fitting suitable casings. The design of the supply basin has proved to be such that there is a free flow from the supply pipe to the screws, so that there are only minor sludge deposits in the stagnant corners.

A pumping station designed for handling rainwater or effluent diluted with five times its volume is equippedwith three similar screw pumps having the following dimensions:

$Q = 3960\ m^3/h = 1100$ litres/s over 4,9 m = 35°, D = 2,00 m, d = 0,95 m, three-start

L = 10 356 mm, S = D, n = 33 rev/min.

These are driven by 90 kW, three-phase motors running at 1465 rev/min, via a V-belt transmission with a 355/505 mm diameter reduction and a reduction gearing reducing from 1045 to 33 rev/min. The troughs are made of concrete and have gaps which range from 6 to 10 mm and which, on average, exceed the theoretical design value of 6,5 mm.

The centre of the lower end of the screw, at the end of the blading, is located at a height of + 2,21 m relative to a reference point on the structure; the spill point is at a height of + 7,12 m, and the contact point is at + 1,37 m. The optimum immersion is at r cos β above the centre of the lower end of the screw, i.e., with a water level in the supply basin of + 2,61 m.

For determining the delivery rate use is made of a rainwater storage basin having a surface area of 47,10 x 18,00 m = 852.75 m^2. Each rainwater pump has a separate discharge with a non-return flap. The results of measurements made on one of the three pumps are given in Table 2.

Table 2 Measured results and calculated values from investigations on an Archimedian screw rainwater pump.

Versuch Nr. = Test No.
Unterwasserspiegel = Supply basin level
Oberwasserspiegel = Discharge basin level
U/min = rev/min

Tabelle 2 Meßwerte und Auswertung von Untersuchungen an einer Regenwasserförderschnecke

Versuch Nr.	Unterwasserspiegel	Oberwasserspiegel	$H_{ges.}$ (m)	Q (l/s)	n_M/n_S (U/min)	N_e (kW)	V	A	N_K (kW)	η_K	$\eta_{ges.}$
[illegible]	+ 2,60	+ 7,55	5,35	890	1471/33	[illegible]	[illegible]	[illegible]	[illegible]	82,5	57,4
[illegible]	+ 2,70	+ 7,67	5,27	905	1467/33	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	60,2
[illegible]	[illegible]	+ 7,60	4,18	1000	1463/33	[illegible]	[illegible]	[illegible]	[illegible]	82,5	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	1150	1461/33	[illegible]	[illegible]	[illegible]	[illegible]	83,0	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	1170	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

As a general conclusion it may be stated that the efficiencies achieved at part load and at full delivery are not entirely satisfactory for this rainwater pump. Nevertheless, they are not very much below the theoretical value of 80.2 % - if the full circumstances of the pump tested are taken into account.

Apart from the exessive gap widths, a factor that tends to reduce the efficiency is the heavy turbulence where the medium enters the pump, which results in the water flooding over the lower portions of the screws. This, together with the excessive gap widths, causes the efficiency to drop 4.5 % below the maximum value anticipated at full load.

Photographs of the three Archimedian screw rainwater pumps will be found in the illustrations section. Careful inspection of these will reveal the heavy turbulence where the water flows into the pump screw.

PART 5 TABLES AND DIAGRAMS

Table 3 q-Values for Determination of Delivery of three-Start Archimedian Screw Pumps

For screws having different number of starts, see data and conversion factors in Part 2/2.8

Table 4 Table for Selection of Steel Pipe for Use in Screw Pump Shafts from DIN 2458

Außen-Ø	=	Ext. dia.
Wanddicke	=	Wall thickness
Gewicht (kg/lfdm)	=	Weight (kg/metre run)
Widerstandsmoment	=	Section Modulus
Trägheitsmoment	=	Moment of Inertia

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[Ta]belle 3 q-Werte zur Bestimmung der Fördermenge dreigängiger Wasserförderschnecken

[illegible]	$\beta = 22°$	$\beta = 26°$	$\beta = 30°$	$\beta = 33°$	$\beta = 35°$	$\beta = 37°$	$\beta = 40°$
[illegible]	0,00507	0,00460	0,00393	0,00354	0,00324	0,00295	0,00247
[illegible]	0,00503	0,00460	0,00405	0,00365	0,00334	0,0030[illegible]	0,0025[illegible]
[illegible]	0,00500	0,00460	0,00417	0,00376	0,00343	0,0031[illegible]	0,0026[illegible]
[illegible]	0,00479	0,00436	0,00406	0,00366	0,00335	[illegible]	0,0025[illegible]
[illegible]	0,00457	0,00411	0,00395	0,00356	0,00326	[illegible]	0,0025[illegible]
[illegible]	[illegible]	[illegible]	0,00383	0,00347	[illegible]	[illegible]	[illegible]

[illegible]

[illegible] DIN 2458 für die Herstellung von [illegible]

[illegible] (mm)	Wanddicke [illegible]	Gewicht (kg/lfdm)	Widerstands-moment [illegible]	[illegible] moment [illegible]
[illegible]	7,1	41,2	4 720	394,6
[illegible]	12,6	72,0	6 190	500,8
273,0	7,1	46,7	5 720	419,1
273,0	12,6	80,9	11 670	855,0
[illegible]	[illegible]	61,2	[illegible]	[illegible]
[illegible]	[illegible]	107,0	[illegible]	[illegible]
[illegible]	[illegible]	123,0	[illegible]	[illegible]
[illegible]	[illegible]	154,0	[illegible]	[illegible]
[illegible]	10,0	173,0	[illegible]	[illegible]
[illegible]	[illegible]	223,0	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	334,2	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	12,0	498,4	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

* Die auszugsweise Wiedergabe erfolgt mit Genehmigung des Deutschen Normenausschusses

Table 5 Performance Data for RITZ Archimedian Screw Pumps of the Series 11 for various angles of inclination and numbers of starts and at different speeds.

(Note: The figures following the Type No. 11 are the rounded-up values of the external diameter in decimetres.)

Typ = Type

Aufstellwinkel = Angle of inclination

Gangzahl = No. of starts

n (U/min) = n (rev/min)

	Typ	1104	1104	1104	1104	1104	1104	1104	1105	1105	1105	1105	1106	110[illegible]
Aufstell-Winkel [illegible]	Gangzahl	1	1	1	2	2	3	3	1	1	2	2	2	3
	Q (l/s)	10	14	17	18	21	22	27	[illegible]	29	30	[illegible]	[illegible]	[illegible]
	n (U/min)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	(U/min)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Aufstell-Winkel [illegible]	Q (l/s)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	n (U/min)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Aufstell-Winkel [illegible]	Q (l/s)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	n (U/min)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	67.6	75.6	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Aufstell-Winkel [illegible]	Q (l/s)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	n (U/min)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

[illegible]	Typ	106	106	106	108	108	108	107	107	107	107	107	10[illegible]
Aufstell-Winkel $\beta = 30^\circ$	Gangzahl	1	1	2	2	3	3	[illegible]	[illegible]	[illegible]	2	3	3
	Q (l/s)	40	44	45	55	58	69	80	86	70	83	84	104
	n (u/min)	64	70	57	70	57	70	57	63	63	63	51	64
	η_{res} (%)	72,1	73,1	75,2	76,4	76,3	78,5	73,2	73,2	76,4	76,6	78,6	78,9
Aufstell-Winkel [illegible]	Q (l/s)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	n (u/min)	[illegible]	70	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	η_{res} (%)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Aufstell-Winkel [illegible]	Q (l/s)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	n (u/min)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	η_{res} (%)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Aufstell-Winkel [illegible]	Q (l/s)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	n (u/min)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	η_{res} (%)			75,0	76,2	76,1	76,0	[illegible]	[illegible]	[illegible]	76,7	[illegible]	[illegible]
Aufstell-Winkel $\beta = 40^\circ$	Q (l/s)			29	35	38	43			40	52	53	65
	n (u/min)			59	70	58	70			48	63	51	63
	η_{res} (%)			74,8	76,1	76,3	77,6			73,2	76,7	76,5	77,9

	Typ	1108	1108	1108	1108	1108	1108	1109	1109	1109	1109	[illegible]	[illegible]
Aufstell.-Winkel $\gamma = 30°$	Gangzahl	1	1	2	2	3	3	1	1	2	2	[illegible]	[illegible]
	Q (l/s)	80	88	90	111	115	139	110	116	120	148	155	183
	n (u/min)	53	58	47	58	48	58	51	54	44	54	46	54
	η_{ges} (%)	73,6	74,1	76,1	77,0	79,1	79,1	74,0	74,5	77,8	77,8	79,5	79,5
Aufstell.-Winkel $\gamma = 33°$	Q (l/s)	70	80	82	99	100	125	97	105	110	132	140	165
	n (u/min)	51	58	50	58	47	58	50	54	45	54	46	54
	η_{ges} (%)	72,5	73,9	76,5	77,0	78,7	79,0	73,8	74,3	76,9	[illegible]	[illegible]	[illegible]
Aufstell.-Winkel $\gamma = 35°$	Q (l/s)	68	73	75	91	95	114	80	96	105	121	130	151
	n (u/min)	54	58	48	58	48	58	51	54	47	54	47	54
	η_{ges} (%)	73,0	73,6	76,0	77,0	78,6	79,0	73,5	74,0	76,9	77,2	79,0	79,2
Aufstell.-Winkel $\gamma = 37°$	Q (l/s)			75	83	95	104	[illegible]	[illegible]	90	109	[illegible]	137
	n (u/min)			52	58	47	[illegible]	[illegible]	[illegible]	[illegible]	54	[illegible]	[illegible]
	η_{ges} (%)			76,5	76,9	77,7	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Aufstell.-Winkel $\gamma = 40°$	Q (l/s)	60	69	70	87	[illegible]	[illegible]	80	92	95	[illegible]	[illegible]	[illegible]
	n (u/min)			50	58	47	58	[illegible]	[illegible]	47	54	46	54
	η_{ges} (%)			76,1	76,9	76,8	78,3	[illegible]	[illegible]	76,6	77,1	77,5	78,5

	Typ	1110	1110	1110	1110	1110	1110	1112	1112	1112	1112	1112	1112
Aufstell.-Winkel $\gamma = 30°$	Gangzahl	1	1	2	2	3	3	1	1	2	2	3	3
	Q (l/s)	145	153	165	192	210	240	200	219	240	275	310	344
	n (u/min)	47	50	43	50	44	50	40	44	38	44	39	44
	η_{ges} (%)	74,6	74,8	77,3	77,7	79,7	79,5	75,1	75,4	77,8	78,1	80,1	80,0
Aufstell.-Winkel $\gamma = 33°$	Q (l/s)	130	138	150	173	185	216	190	197	220	248	270	310
	n (u/min)	47	50	43	50	43	50	42	44	39	44	38	44
	η_{ges} (%)	74,2	74,6	77,3	77,7	79,1	79,3	75,1	75,2	77,8	78,1	80,0	80,0
Aufstell.-Winkel $\gamma = 35°$	Q (l/s)	115	126	135	158	170	198	172	181	195	227	245	284
	n (u/min)	46	50	43	50	43	50	42	44	38	44	38	44
	η_{ges} (%)	73,6	74,3	76,8	77,7	79,1	79,3	74,5	75,1	77,4	78,0	79,6	79,8
Aufstell.-Winkel $\gamma = 37°$	Q (l/s)			130	144	155	180			175	207	225	259
	n (u/min)			45	50	43	50			37	44	38	44
	η_{ges} (%)			77,2	77,6	78,6	79,1			77,7	78,0	79,2	79,8
Aufstell.-Winkel $\gamma = 40°$	Q (l/s)			105	121	130	151			145	173	185	217
	n (u/min)			43	50	43	50			37	44	37	44
	η_{ges} (%)			76,6	77,5	78,0	78,7			77,4	77,9	78,7	79,2

Aufstell.-Winkel	Typ	1114	1114	1114	1114	1114	1114	1116	1116	1116	1116	1116	1116
Aufstell.-Winkel $\beta = 30°$	Gangzahl	1	1	2	2	3	3	1	1	2	2	3	3
	Q (l/s)	315	336	350	422	450	526	420	451	510	566	650	706
	n (u/min)	38	40	33	40	34	40	35	37	33	37	34	37
	η_{ges} (%)	76,1	76,1	78,5	78,8	80,0	80,4	75,4	75,5	78,1	78,0	80,1	79,8
Aufstell.-Winkel $\beta = 33°$	Q (l/s)	280	303	340	380	420	474	370	407	460	510	580	637
	n (u/min)	37	40	36	40	35	40	34	37	34	37	34	37
	η_{ges} (%)	75,9	76,1	78,7	78,8	80,6	80,4	75,3	75,5	78,1	78,0	80,0	79,8
Aufstell.-Winkel $\beta = 35°$	Q (l/s)	250	276	300	347	380	434	340	372	410	466	520	583
	n (u/min)	36	40	35	40	35	40	34	37	33	37	33	37
	η_{ges} (%)	75,4	75,9	78,5	78,8	80,5	80,4	74,8	75,2	77,8	78,0	79,8	79,8
Aufstell.-Winkel $\beta = 37°$	Q (l/s)			270	317	345	395			370	424	470	531
	n (u/min)			34	40	35	40			32	37	33	37
	η_{ges} (%)			78,6	78,5	80,0	80,2			77,6	78,0	79,4	79,4
Aufstell.-Winkel $\beta = 40°$	Q (l/s)			220	266	290	331			310	354	410	455
	n (u/min)			33	40	35	40			33	37	34	37
	η_{ges} (%)			77,8	78,2	79,4	79,8			77,5	77,8	79,0	79,2

	Typ	1118	1118	1118	1118	1118	1118	1120	1120	1120	1120	1120	1120
Aufstell.-Winkel $\vartheta = 30°$	Gangzahl	1	1	2	2	3	3	1	1	2	2	3	3
	Q (l/s)	540	605	700	760	890	948	700	760	880	954	1100	1195
	n (u/min)	30	34	31	34	32	34	29	32	29	32	29	32
	η_{ges} (%)	76,0	76,1	78,8	78,7	80,5	80,3	76,3	76,4	78,8	78,7	80,6	80,3
Aufstell.-Winkel $\vartheta = 33°$	Q (l/s)	495	546	620	684	790	855	620	688	790	860	1000	1076
	n (u/min)	31	34	31	34	31	34	29	32	29	32	30	32
	η_{ges} (%)	75,7	76,0	78,8	78,7	80,5	80,3	76,0	76,2	78,7	78,7	80,5	80,3
Aufstell.-Winkel $\vartheta = 35°$	Q (l/s)	440	498	565	625	720	782	565	628	700	780	920	1013
	n (u/min)	30	34	31	34	31	34	29	32	29	32	30	32
	η_{ges} (%)	75,2	75,9	78,5	78,7	80,4	80,3	75,5	76,0	78,6	78,7	80,4	80,3
Aufstell.-Winkel $\vartheta = 37°$	Q (l/s)			510	570	650	713			590	718	810	[illegible]
	n (u/min)			30	34	31	34			29	32	29	32
	η_{ges} (%)			78,4	78,6	79,9	80,1			78,5	78,8	80,0	80,[illegible]
Aufstell.-Winkel $\vartheta = 40°$	Q (l/s)			420	480	540	588			540	586	680	752
	n (u/min)			30	34	31	34			29	32	29	32
	η_{ges} (%)			78,1	78,5	79,4	79,7			78,4	78,5	79,5	79,8

	Typ	1123	1123	1123	1123	1123	1123	1126	1126	1126	1126	1126	1126
Aufstell.-Winkel $\beta = 30°$	Gangzahl	1	1	2	2	3	3	1	1	2	2	[illegible]	3
	Q (l/s)	945	1045	1210	1312	1500	1640	1300	1436	1600	1800	2000	2240
	n (u/min)	26	29	27	29	27	29	24	27	24	27	[illegible]	27
	η_{ges} (%)	78,0	78,0	79,5	79,0	80,8	80,9	76,7	77,0	79,2	[illegible]	[illegible]	80,5
Aufstell.-Winkel $\beta = 33°$	Q (l/s)	840	944	1070	1182	1360	1478	1090	1283	1410	1610	[illegible]	2020
	n (u/min)	26	29	26	29	27	29	23	27	24	27	[illegible]	27
	η_{ges} (%)	77,1	77,0	79,1	79,0	80,7	80,6	76,1	76,9	79,1	[illegible]	[illegible]	80,5
Aufstell.-Winkel $\beta = 35°$	Q (l/s)	760	883	980	1082	1140	1353	1000	1179	1300	1400	[illegible]	[illegible]
	n (u/min)	26	29	26	29	25	29	23	27	24	[illegible]	[illegible]	27
	η_{ges} (%)	75,8	76,5	78,9	79,0	80,5	80,5	76,0	76,7	78,9	[illegible]	[illegible]	80,5
Aufstell.-Winkel $\beta = 37°$	Q (l/s)			785	885	1120	1232			1150	[illegible]	[illegible]	[illegible]
	n (u/min)			26	29	26	29			23	[illegible]	[illegible]	27
	η_{ges} (%)			78,6	79,0	80,2	80,2			78,6	79,0	80,1	80,3
Aufstell.-Winkel $\beta = 40°$	Q (l/s)			730	834	930	1033			940	1136	1200	1414
	n (u/min)			26	29	26	29			22	27	29	27
	η_{ges} (%)			78,6	78,9	79,4	80,0			78,4	78,9	79,6	80,1

	Typ	1130	1130	1130	1130	1130	1130	1136	1136	1136	1136	1136	1136
Aufstell.-Winkel $\beta = 30°$	Gangzahl	1	1	2	2	3	3	1	1	2	2	3	3
	Q (l/s)	1750	1963	2250	2465	2850	3065	2680	2910	3420	3650	4330	4560
	n (u/min)	22	24	22	24	22	24	20	21	20	21	20	21
	η_{ges} (%)	76,9	77,1	79,4	79,3	81,0	80,6	77,8	77,3	79,5	79,4	80,9	80,8
Aufstell.-Winkel $\beta = 33°$	Q (l/s)	1500	1710	2000	2210	2550	2760	2450	2685	3050	3280	3900	4120
	n (u/min)	21	24	22	24	22	24	20	21	20	21	20	21
	η_{ges} (%)	76,5	77,0	79,3	79,3	80,9	80,6	76,9	77,2	79,5	79,4	80,9	80,8
Aufstell.-Winkel $\beta = 35°$	Q (l/s)	1400	1612	1800	2020	2300	2524	2180	2400	2800	3010	3550	3780
	n (u/min)	21	24	21	24	22	24	19	21	20	21	20	21
	η_{ges} (%)	76,1	76,7	79,2	79,3	80,7	80,5	76,6	76,8	79,3	79,4	80,8	80,8
Aufstell.-Winkel $\beta = 37°$	Q (l/s)			1600	1842	2100	2308			2520	2740	3200	3425
	n (u/min)			21	24	22	24			19	21	20	21
	η_{ges} (%)			79,0	79,2	80,2	80,3			79,2	79,3	80,3	80,5
Aufstell.-Winkel $\beta = 40°$	Q (l/s)			1350	1552	1720	1930			2100	2300	2650	2875
	n (u/min)			21	24	21	24			19	21	19	21
	η_{ges} (%)			78,8	79,2	79,9	79,9			79,1	79,2	80,2	80,2

	Typ	1145	1145	1145	1145	1145	1145
Aufstell.-Winkel $\gamma = 30°$	Gangzahl	1	1	2	2	3	3
	Q (l/s)	4600	4870	5850	6120	7400	7640
	n (u/min)	17	18	17	18	17	18
	η_{ges} (%)	77,4	77,5	79,7	79,5	80,9	80,7
Aufstell.-Winkel $\gamma = 33°$	Q (l/s)	4170	4400	5270	5510	6500	6890
	n (u/min)	17	18	17	18	17	18
	η_{ges} (%)	77,3	77,3	79,5	79,5	80,9	80,7
Aufstell.-Winkel $\gamma = 35°$	Q (l/s)	3800	4020	4800	5040	6050	6300
	n (u/min)	17	18	17	18	17	18
	η_{ges} (%)	77,0	77,1	79,3	79,5	80,8	80,7
Aufstell.-Winkel $\gamma = 37°$	Q (l/s)			4350	4585	5500	5740
	n (u/min)			17	18	17	18
	η_{ges} (%)			79,3	79,4	80,5	80,5
Aufstell.-Winkel $\gamma = 40°$	Q (l/s)			3610	3850	4600	4820
	n (u/min)			17	18	17	18
	η_{ges} (%)			79,2	79,3	80,1	80,2

Table 6 Dimensions with Weight and Water Content Data for Archimedian Screw Pumps

Type u. Nr. = Type and No.

Nenndurchmesser = Nominal dia.

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

für Typ 1104 .... = For Type 1104, G is reduced by 30,3 kg/m and L_{max} to 6,5 – 7,3 m if s_w = 7,1 mm; f taken equal to s_{sp}

für Typ 1105 .... = For Type 1105, G is reduced by 34,2 kg/m and L_{max} to 6,7 – 7,6 m if s_w = 7,1 mm; f taken equal to s_{sp}

Gangzahl .... = No. of starts a (for a = 3)

- 121 -
- 123 -
- 125 -
- 127 -
- 128 -

Feste Werte .... = Fixed values: $S/D = 1,0$; $n = \frac{50}{\sqrt[3]{L^2}}$

$s_{sp} = 0,0045\sqrt{D}$; $\delta = \frac{d}{D}$; J = water content/metre;

L_{max} at $f \leqslant s_{sp}$;

$Q = 1,15 \cdot n \cdot q \cdot D^3$; nominal dia. = $D + 2\,s_{sp}$.

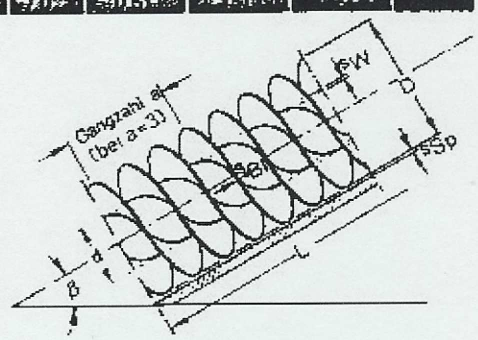
Für Typ 1106 .... = For Type 1106, $\sum G$ is reduced by 44,8 kg/m and L_{max} to 8,0 - 9,1 m, if s_w = 7,1 mm; f taken equal to s_{sp}.

Für Typ 1107 .... = For Type 1107, $\sum G$ is reduced by 43,8 kg/m and L_{max} to 7,6 - 7,8 m, if s_w = 7,1 mm; f taken equal to s_{sp}.

—

Feste Werte: $\frac{S}{D} = 1{,}0$; $n = \frac{50}{\sqrt[3]{D^2}}$; $s_{sp} = 0{,}0045\sqrt{D}$; $\delta = \frac{d}{D}$; J = Wasserinhalt/m; L_{max} bei $f \leq s_{sp}$;
$Q = 1{,}15 \cdot n \cdot q \cdot D^3$; Nenndurchmesser = $D + 2\,s_{sp}$

Typ u. Nr.	Nenndurchmesser	D (mm)	d/s_w (mm)	s_{sp} (mm)	s_{BL} (mm)	δ	ΣG (kg/m)	J (l/m)	(m)
106 – 30/3	600	593,0	355,6/12,5	3,5	6,0	0,60	149,8	98,9	[illegible]
106 – 30/2	600	593,0	355,6/12,5	3,5	6,0	0,60	135,2	79,2	[illegible]
106 – 30/1	600	593,0	355,6/12,5	3,5	6,0	0,50	120,6	63,2	[illegible]
106 – 33/3	600	593,0	355,6/12,5	3,5	6,0	0,60	149,8	88,2	[illegible]
106 – 33/2	600	593,0	355,6/12,5	3,5	6,0	0,60	135,2	[illegible]	[illegible]
106 – 33/1	600	593,0	355,6/12,5	3,5	6,0	0,60	120,6	[illegible]	[illegible]
106 – 35/3	600	593,0	355,6/12,5	3,5	6,0	0,60	149,8	[illegible]	[illegible]
106 – 35/2	600	593,0	355,6/12,5	3,5	6,0	0,60	135,2	[illegible]	[illegible]
106 – 35/1	600	593,0	355,6/12,5	3,5	6,0	0,60	120,6	[illegible]	[illegible]
106 – 37/3	600	593,0	355,6/12,5	3,5	6,0	0,60	149,8	74,4	[illegible]
106 – 37/2	600	593,0	355,6/12,5	3,5	6,0	0,60	135,2	59,6	[illegible]
106 – 40/3	600	593,0	355,6/12,5	3,5	6,0	0,60	149,8	62,3	[illegible]
106 – 40/2	600	593,0	355,6/12,5	3,5	6,0	0,60	135,2	49,8	[illegible]
107 – 30/3	700	692,4	355,6/12,5	3,8	7,0	0,51	173,4	142,8	[illegible]
107 – 30/2	700	692,4	355,6/12,5	3,8	7,0	0,51	151,6	114,3	[illegible]
107 – 30/1	700	692,4	355,6/12,5	3,8	7,0	0,51	128,8	90,9	[illegible]
107 – 33/3	700	692,4	355,6/12,5	3,8	7,0	0,51	173,4	126,6	[illegible]
107 – 33/2	700	692,4	355,6/12,5	3,8	7,0	0,51	151,6	103,0	[illegible]
107 – 33/1	700	692,4	355,6/12,5	3,8	7,0	0,51	128,8	[illegible]	[illegible]
107 – 35/3	700	692,4	355,6/12,5	3,8	7,0	0,51	173,4	[illegible]	[illegible]
107 – 35/2	700	692,4	355,6/12,5	3,8	7,0	0,51	151,6	91,0	[illegible]
107 – 35/1	700	692,4	355,6/12,5	3,8	7,0	0,51	128,4	74,8	[illegible]
107 – 37/3	700	692,4	355,6/12,5	3,8	7,0	0,51	173,4	107,0	[illegible]
107 – 37/2	700	692,4	355,6/12,5	3,8	7,0	0,51	151,6	85,7	[illegible]
107 – 40/3	700	692,4	355,6/12,5	3,8	7,0	0,51	173,4	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	355,6/12,5	3,8	7,0	0,51	[illegible]	[illegible]	[illegible]



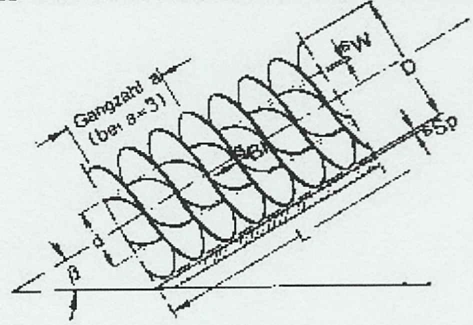
für Typ 1106 vermindert sich bei $s_w = 7{,}1$ mm ΣG um 44,8 kg/m und L_{max} auf 8,0—9,1 m; $f = s_{sp}$ angesetzt.

für Typ 1107 vermindert sich ΣG bei $s_w = 7{,}1$ mm um 43,8 kg/m und L_{max} auf 7,6—8,7 m; $f = s_{sp}$ angesetzt.

Für Typ 1108 .... = For Type 1108, Σ G is reduced by 43,8 kg/m and L_{max} to 7,2 - 8,4 m, if s_w = 7,1 mm; f taken equal to s_{sp}.

Für Typ 1109 .... = For Type 1109, Σ G is reduced by 31,0 kg/m and L_{max} to 9,8 - 11,2 m, if s_w = 10,0 mm; f taken equal to s_{sp}.

Typ u. Nr. = β/a	Nenndurchmesser	D (mm)	d/s_w (mm)	s_{sp} (mm)	s_{SL} (mm)	[illegible]	ΣG (kg/m)	J (l/m)	L_{max} (m)
1108 – 30/3	800	792,0	355,6/12,5	4,0	8,0	0,45	203,8	179,5	8,0
1108 – 30/2	800	792,0	355,6/12,5	4,0	8,0	0,45	171,2	143,7	8,4
1108 – 30/1	800	792,0	355,6/12,5	4,0	8,0	0,45	138,6	114,6	8,6
1108 – 33/3	800	792,0	355,6/12,5	4,0	8,0	0,45	203,8	161,9	8,2
1108 – 33/2	800	792,0	355,6/12,5	4,0	8,0	0,45	171,2	129,5	8,6
1108 – 33/1	800	792,0	355,6/12,5	4,0	8,0	0,45	138,6	103,4	9,0
1108 – 35/3	800	792,0	355,6/12,5	4,0	8,0	0,45	203,8	148,2	8,3
1108 – 35/2	800	792,0	355,6/12,5	4,0	8,0	0,45	171,2	118,3	8,7
1108 – 35/1	800	792,0	355,6/12,5	4,0	8,0	0,45	138,6	94,4	9,2
1108 – 37/3	800	792,0	355,6/12,5	4,0	8,0	0,45	203,8	134,8	8,4
1108 – 37/2	800	792,0	355,6/12,5	4,0	8,0	0,45	171,2	107,7	8,9
1108 – 40/3	800	792,0	355,6/12,5	4,0	8,0	0,45	203,8	113,1	8,6
1108 – 40/2	800	792,0	355,6/12,5	4,0	8,0	0,45	171,2	90,0	9,1
1109 – 30/3	900	891,4	508/12,5	4,3	8,0	0,57	245,8	227,5	10,3
1109 – 30/2	900	891,4	508/12,5	4,3	8,0	0,57	215,2	181,8	10,7
1109 – 30/1	900	891,4	508/12,5	4,3	8,0	0,57	184,6	144,8	11,2
1109 – 33/3	900	891,4	508/12,5	4,3	8,0	0,57	245,8	204,5	10,5
1109 – 33/2	900	891,4	508/12,5	4,3	8,0	0,57	215,2	164,0	10,8
1109 – 33/1	900	891,4	508/12,5	4,3	8,0	0,57	184,6	130,8	[illegible]
1109 – 35/3	900	891,4	508/12,5	4,3	8,0	0,57	245,8	187,6	[illegible]
1109 – 35/2	900	891,4	508/12,5	4,3	8,0	0,57	215,2	149,8	[illegible]
1109 – 35/1	900	891,4	508/12,5	4,3	8,0	0,57	184,6	119,4	11,6
1109 – 37/3	900	891,4	508/12,5	4,3	8,0	0,57	245,8	170,7	10,8
1109 – 37/2	900	891,4	508/12,5	4,3	8,0	0,57	215,2	135,8	11,3
1109 – 40/3	900	891,4	508/12,5	4,3	8,0	0,57	245,8	143,0	11,1
1109 – 40/2	900	891,4	508/12,5	4,3	8,0	0,57	215,2	113,7	11,6



für Typ 1108 vermindert sich bei s_w = 7,1 mm
ΣG um 43,8 kg/m und L_{max} auf 7,2—8,4 m;
f = s_{sp} angesetzt.

für Typ 1109 vermindert sich bei s_w = 10,0 mm
ΣG um 31,0 kg/m und L_{max} auf 9,8—11,2 m;
f = s_{sp} angesetzt.

Typ u. Nr. – β/a	Nenndurchmesser	D (mm)	d/s_w (mm)	s_{sp} (mm)	s_{Bl} (mm)	δ	ΣG (kg/m)	I (l/m)	L_{max} (m)
110 – 30/3	1000	991,0	508/12,5	4,5	8,0	0,51	270,2	287,5	10,0
110 – 30/2	1000	991,0	508/12,5	4,5	8,0	0,51	228,8	230,5	10,5
110 – 30/1	1000	991,0	508/12,5	4,5	8,0	0,51	191,4	183,5	11,1
110 – 33/3	1000	991,0	508/12,5	4,5	8,0	0,51	270,2	259,0	10,2
110 – 33/2	1000	991,0	508/12,5	4,5	8,0	0,51	228,8	207,5	10,7
110 – 33/1	1000	991,0	508/12,5	4,5	8,0	0,51	191,4	165,5	11,2
110 – 35/3	1000	991,0	508/12,5	4,5	8,0	0,51	270,2	237,0	10,4
110 – 35/2	1000	991,0	508/12,5	4,5	8,0	0,51	228,8	189,5	10,9
110 – 35/1	1000	991,0	508/12,5	4,5	8,0	0,51	191,4	151,0	11,4
110 – 37/3	1000	991,0	508/12,5	4,5	8,0	0,51	270,2	216,0	10,6
110 – 37/2	1000	991,0	508/12,5	4,5	8,0	0,51	228,8	173,0	11,1
110 – 40/3	1000	991,0	508/12,5	4,5	8,0	0,51	270,2	181,5	10,9
110 – 40/2	1000	991,0	508/12,5	4,5	8,0	0,51	228,8	145,5	11,4
112 – 30/3	1200	1190	711,2/10,0	5,0	8,0	0,60	291,5	396	11,2
112 – 30/2	1200	1190	711,2/10,0	5,0	6,0	0,60	252,0	317	11,6
112 – 30/1	1200	1190	711,2/10,0	5,0	8,0	0,60	212,5	253	12,0
112 – 33/3	1200	1190	711,2/10,0	5,0	8,0	0,60	291,5	357	11,3
112 – 33/2	1200	1190	711,2/10,0	5,0	8,0	0,60	252,0	286	11,7
112 – 33/1	1200	1190	711,2/10,0	5,0	8,0	0,60	212,5	227	12,[illegible]
112 – 35/3	1200	1190	711,2/10,0	5,0	8,0	[illegible]	291,5	327	11,[illegible]
112 – 35/2	1200	1190	711,2/10,0	5,0	8,0	0,60	252,0	262	11,7
112 – 35/1	1200	1190	711,2/10,0	5,0	6,0	0,60	212,5	209	12,1
112 – 37/3	1200	1190	711,2/10,0	5,0	8,0	0,60	291,5	298	11,5
112 – 37/2	1200	1190	711,2/10,0	5,0	8,0	0,60	252,0	239	11,9
112 – 40/3	1200	1190	711,2/10,0	5,0	8,0	0,60	291,5	250	11,6
112 – 40/2	1200	1190	711,2/10,0	5,0	8,0	0,60	252,0	199	12,0

Für Typ 1110 .... = For Type 1110, ΣG is reduced by 35,o kg/m and L_{max} to 9,5 – 11,0 m, if s_w = 10.0 mm; f taken equal to s_{sp}.

f taken as 4,5 mm

f taken as 5,0 mm

f taken as 5,0 mm

- 125 -

f taken as 5,0 mm

f taken as 5,0 mm

- 126 -

f taken as 6,0 mm

f taken as 6,0 mm

- 127 -

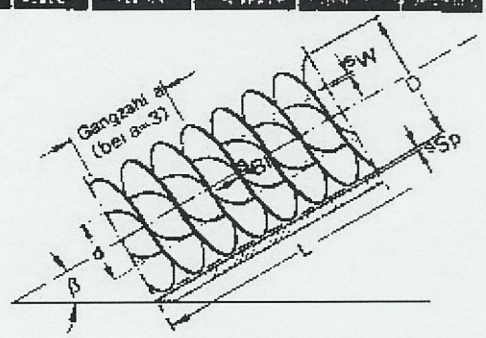
f taken as 7,0 mm

f taken as 7,5 mm

- 128 -

f taken as 8,0 mm

[illegible]	Nenn-durch-messer	(mm)	d/s (mm)	(mm)	(mm)	[illegible]	(kg/m)	(t/m)	(m)
114 – 30/3	1400	1389,4	711,2/10,0	5,3	8,0	0,51	330,5	568	10,5
114 – 30/2	1400	1389,4	711,2/10,0	5,3	8,0	0,51	278,0	454	11,0
114 – 30/1	1400	1389,4	711,2/10,0	5,3	8,0	0,51	225,5	[illegible]	11,0
114 – 33/3	1400	1389,4	711,2/10,0	5,3	8,0	0,51	330,5	[illegible]	10,6
114 – 33/2	1400	1389,4	711,2/10,0	5,3	8,0	0,51	278,0	409	[illegible]
114 – 33/1	1400	1389,4	711,2/10,0	5,3	8,0	0,51	225,5	[illegible]	[illegible]
114 – 35/3	1400	[illegible]	711,2/10,0	5,3	8,0	0,51	330,5	[illegible]	10,7
114 – 35/2	1400	[illegible]	711,2/10,0	5,3	8,0	0,51	278,0	[illegible]	[illegible]
114 – 36/1	1400	[illegible]	711,2/10,0	[illegible]	8,0	0,51	225,5	[illegible]	11,8
114 – 37/3	1400	1389,4	711,2/10,0	5,3	8,0	0,51	330,5	[illegible]	11,0
114 – 37/2	1400	1389,4	711,2/10,0	5,3	8,0	0,51	278,0	[illegible]	11,5
114 – 40/3	1400	1389,4	711,2/10,0	5,3	8,0	0,51	330,5	[illegible]	11,3
114 – 40/2	1400	1389,4	711,2/10,0	5,3	8,0	0,51	278,0	[illegible]	11,8
116 – 30/3	1600	1588,6	711,2/10,0	5,7	8,0	0,45	387,4	[illegible]	10,1
116 – 30/2	1600	1588,6	711,2/10,0	5,7	8,0	0,45	302,6	[illegible]	10,6
116 – 30/1	1600	1588,6	711,2/10,0	5,7	8,0	0,45	237,8	[illegible]	11,2
116 – 33/3	1600	[illegible]	711,2/10,0	5,7	8,0	0,45	[illegible]	[illegible]	10,2
[illegible]	1600	[illegible]	711,2/10,0	5,7	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	1600	[illegible]	711,2/10,0	5,7	[illegible]	[illegible]	237,8	[illegible]	[illegible]
[illegible]	1600	[illegible]	711,2/10,0	5,7	8,0	0,45	[illegible]	[illegible]	10,1
[illegible]	1600	1588,6	711,2/10,0	5,7	8,0	[illegible]	302,6	[illegible]	10,6
[illegible]	1600	1588,6	711,2/10,0	5,7	8,0	0,45	237,8	[illegible]	11,5
116 – 37/3	1600	1588,6	711,2/10,0	5,7	8,0	0,45	387,4	536	10,6
116 – 37/2	1600	1588,6	711,2/10,0	5,7	8,0	0,45	302,6	428	11,3
116 – 40/3	1600	1588,6	711,2/10,0	5,7	8,0	0,45	387,4	459	11,1
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]



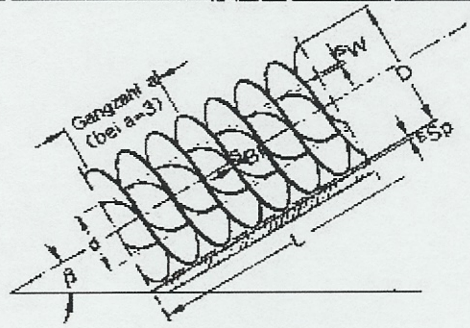
f = 5,0 mm angesetzt.

f = 5,0 mm angesetzt.

Feste Werte: $\frac{S}{D} = 1{,}0$; $n = \frac{50}{\sqrt[3]{D^2}}$; $s_{sp} = 0{,}0045\sqrt{D}$; $\delta = \frac{d}{D}$; J = Wasserinhalt/m; L_{max} bei $f \leq s_{sp}$;

$Q = 1{,}15 \cdot n \cdot q \cdot D^3$; Nenndurchmesser $= D - 2\,s_{sp}$

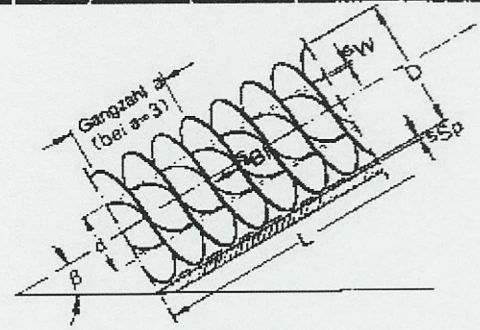
[illegible]	Nenn-durch-messer	[illegible] (mm)	[illegible] (mm)	[illegible] (mm)	[illegible] (mm)	[illegible]	[illegible] (kg/m)	[illegible] (l/m)	[illegible] (m)
118 – 30/3	1800	1788	914,4/10,0	6,0	8,0	0,51	426,1	924	11,6
118 – 30/2	1800	1788	914,4/10,0	6,0	8,0	0,51	358,4	741	12,5
118 – 30/1	1800	1788	914,4/10,0	6,0	8,0	0,51	290,7	590	13,2
118 – 33/3	1800	1788	914,4/10,0	6,0	8,0	0,51	426,1	834	11,8
118 – 33/2	1800	1788	914,4/10,0	6,0	8,0	0,51	358,4	668	12,7
118 – 33/1	1800	1788	914,4/10,0	6,0	8,0	0,51	290,7	532	13,3
118 – 35/3	1800	1788	914,4/10,0	6,0	8,0	0,51	426,1	783	12,0
118 – 35/2	1800	1788	914,4/10,0	6,0	8,0	0,51	358,4	610	12,8
118 – 35/1	1800	1788	914,4/10,0	6,0	8,0	0,51	290,7	484	13,0
118 – 37/3	1800	1788	914,4/10,0	6,0	8,0	0,51	426,1	693	12,6
118 – 37/2	1800	1788	914,4/10,0	6,0	8,0	0,51	358,4	556	13,0
118 – 40/3	1800	1788	914,4/10,0	6,0	8,0	0,51	426,1	583	12,8
118 – 40/2	1800	1788	914,4/10,0	6,0	8,0	0,51	358,4	466	13,2
120 – 30/3	2000	1987,2	914,4/10,0	6,4	10,0	0,46	523	1116	11,8
120 – 30/2	2000	1987,2	914,4/10,0	6,4	10,0	0,46	423	896	12,6
120 – 30/1	2000	1987,2	914,4/10,0	6,4	10,0	[illegible]	[illegible]	714	[illegible]
120 – 33/3	2000	[illegible]	[illegible]	6,4	10,0	[illegible]	[illegible]	1011	[illegible]
120 – 33/2	2000	[illegible]	[illegible]	6,4	10,0	[illegible]	[illegible]	806	[illegible]
[illegible]	2000	1987,2	[illegible]	[illegible]	10,0	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	2000	[illegible]	[illegible]	[illegible]	[illegible]	0,46	523	924	[illegible]
120 – 35/2	2000	1987,2	[illegible]	6,4	10,0	[illegible]	423	740	12,2
120 – 35/1	2000	1987,2	914,4/10,0	6,4	10,0	0,46	323	591	12,8
120 – 37/3	2000	1987,2	914,4/10,0	6,4	10,0	0,46	523	842	12,1
120 – 37/2	2000	1987,2	914,4/10,0	6,4	10,0	0,46	423	674	12,8
120 – 40/3	2000	1987,2	914,4/10,0	6,4	10,0	0,46	523	707	12,5
120 – 40/2	2000	1987,2	914,4/10,0	6,4	10,0	0,46	423	563	13,5



f = 5,0 mm angesetzt.

f = 6,0 mm angesetzt.

Typ u. Nr. – β/e	Nenn-durch-messer	D (mm)	d/s_w (mm)	s_{Sp} (mm)	s_{BL} (mm)	[illegible]	ΣG (kg/m)	J (l/m)	L_{max} (m)
123 – 30/3	2300	2286,4	1016/10	6,8	10	0,45	600,5	1478	12,2
123 – 30/2	2300	2286,4	1016/10	6,8	10	0,45	483	1182	12,8
123 – 30/1	2300	2286,4	1016/10	6,8	10	0,45	365,5	942	13,6
123 – 33/3	2300	2286,4	1016/10	6,8	10	0,45	600,5	1298	12,4
123 – 33/2	2300	2286,4	1016/10	6,8	10	0,45	483	1064	12,9
123 – 33/1	2300	2286,4	1016/10	6,8	10	0,45	365,5	849	13,7
123 – 35/3	2300	2286,4	1016/10	6,8	10	0,45	600,5	1217	12,5
123 – 35/2	2300	2286,4	1016/10	6,8	10	0,45	483	974	13,0
123 – 35/1	2300	2286,4	1016/10	6,8	10	0,45	365,5	774	13,8
123 – 37/3	2300	2286,4	1016/10	6,8	10	0,45	600,5	1108	13,1
123 – 37/2	2300	2286,4	1016/10	6,8	10	0,45	483	886	13,8
123 – 40/3	2300	2286,4	1016/10	6,8	10	0,45	600,5	929	13,5
123 – 40/2	2300	2286,4	1016/10	6,8	10	0,45	483	742	14,1
126 – 30/3	2600	2585,4	1220/11	7,3	10	0,475	721,2	1920	13,2
126 – 30/2	2600	2585,4	1220/11	7,3	10	0,475	592,2	1541	13,7
126 – 30/1	2600	2585,4	1220/11	7,3	10	0,475	463,2	1228	14,8
126 – 33/3	2600	2585,4	1220/11	7,3	10	[illegible]	721,2	1728	13,3
126 – 33/2	2600	2585,4	1220/11	7,3	10	[illegible]	592,2	1362	13,9
126 – 33/1	2600	2585,4	1220/11	7,3	10	0,475	463,2	1107	15,0
126 – 35/3	2600	2585,4	1220/11	7,3	10	0,475	721,2	1578	13,5
126 – 35/2	2600	2585,4	1220/11	7,3	10	0,475	592,2	1265	14,3
126 – 35/1	2600	2585,4	1220/11	7,3	10	0,475	463,2	1008	15,1
126 – 37/3	2600	2585,4	1220/11	7,3	10	0,475	721,2	1443	14,1
126 – 37/2	2600	2585,4	1220/11	7,3	10	0,475	552,2	1154	15,0
126 – 40/3	2600	2585,4	1220/11	7,3	10	0,475	721,2	1210	15,0
126 – 40/2	2600	2585,4	1220/11	7,3	10	0,475	592,2	972	15,4



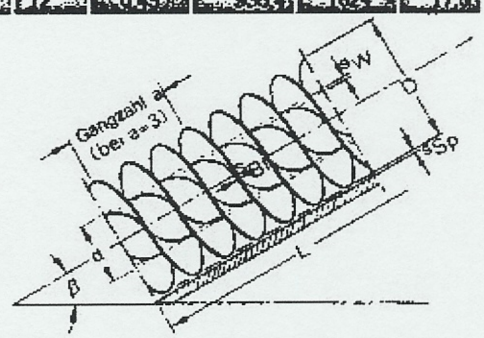
f = 6,0 mm angesetzt.

f = 6,0 mm angesetzt

Feste Werte: $\frac{S}{D} = 1{,}0$; $n = \frac{50}{\sqrt[3]{D^2}}$; $s_{sp} = 0{,}0045\sqrt{D}$; $\delta = \frac{d}{D}$; J … Wasserinhalt/m; L_{max} bei $f \leq s_{sp}$;

$Q = 1{,}15 \cdot n \cdot q \cdot D^3$; Nenndurchmesser = $D + 2\, s_{sp}$

Typ u. Nr. – β/a	Nenn-durch-messer	D (mm)	d/s_w (mm)	s_{sp} (mm)	s_{Bl} (mm)	δ	ΣG (kg/m)	J (l/m)	L_{max} (m)
130 – 30/3	3000	2984,4	1420/12	7,8	10	0,475	671,3	2550	15,0
130 – 30/2	3000	2984,4	1420/12	7,8	10	0,475	723,8	2045	15,6
130 – 30/1	3000	2984,4	1420/12	7,8	10	0,475	576,3	1632	16,2
130 – 33/3	3000	2984,4	1420/12	7,8	10	0,475	671,3	2295	15,1
130 – 33/2	3000	2984,4	1420/12	7,8	10	0,475	723,8	1837	15,7
130 – 33/1	3000	2984,4	1420/12	7,8	10	0,475	576,3	1470	16,3
130 – 35/3	3000	2984,4	1420/12	7,8	10	0,475	671,3	2100	15,2
130 – 35/2	3000	2984,4	1420/12	7,8	10	0,475	723,8	1682	15,6
130 – 35/1	3000	2984,4	1420/12	7,8	10	0,475	576,3	1341	16,4
130 – 37/3	3000	2984,4	1420/12	7,8	10	0,475	671,3	1917	15,5
130 – 37/2	3000	2984,4	1420/12	7,8	10	0,475	723,8	1532	16,0
130 – 40/3	3000	2984,4	1420/12	7,8	10	0,475	671,3	1808	15,8
130 – 40/2	3000	2984,4	1420/12	7,8	10	0,475	723,8	1291	16,3
136 – 30/3	3600	3583	1620/12	8,5	12	0,45	1155,4	3625	15,5
136 – 30/2	3600	3583	1620/12	8,5	12	0,45	936,4	2900	16,4
136 – 30/1	3600	3583	1620/12	8,5	12	0,45	717,4	2310	17,2
136 – 33/3	3600	3583	1620/12	8,5	12	0,45	1155,4	3270	15,6
136 – 33/2	3600	3583	1620/12	8,5	12	0,45	936,4	2610	16,0
136 – 33/1	3600	3583	1620/12	8,5	12	0,45	717,4	2085	17,3
136 – 35/3	3600	3583	1620/12	8,5	12	0,45	1155,4	2990	15,7
136 – 35/2	3600	3583	1620/12	8,5	12	0,45	936,4	2390	16,6
136 – 35/1	3600	3583	1620/12	8,5	12	0,45	717,4	1906	17,4
136 – 37/3	3600	3583	1620/12	8,5	12	0,45	1155,4	2720	16,4
136 – 37/2	3600	3583	1620/12	8,5	12	0,45	936,4	2170	17,2
136 – 40/3	3600	3583	1620/12	8,5	12	0,45	1155,4	2280	16,7
136 – 40/2	3600	3583	1620/12	8,5	12	0,45	936,4	1824	17,5



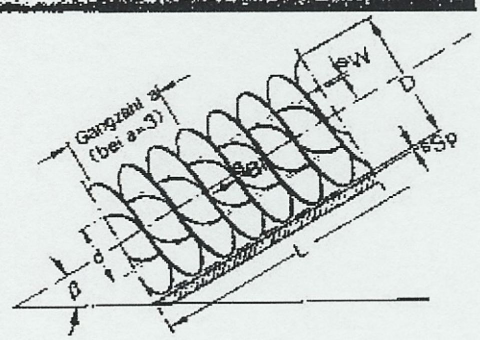
f = 7,0 mm angesetzt.

f = 7,5 mm angesetzt.

Feste Werte: $\frac{S}{D} = 1{,}0$; $n = \frac{50}{\sqrt[3]{D^2}}$; $s_{sp} = 0{,}0045 \sqrt{D}$; $\delta = \frac{d}{D}$; I = Wasserinhalt/m; L_{max} bei $f \leq s_{sp}$;

$Q = 1{,}15 \cdot n \cdot q \cdot D^3$; Nenndurchmesser = D − 2 s_{sp}

Typ u. Nr. – p/a	Nenn-durch-messer	[illegible] (mm)	[illegible] (mm)	s_{sp} (mm)	s_{Bl} (mm)	[illegible]	[illegible] (kg/m)	[illegible] (l/m)	[illegible] (m)
145 – 30/3	4500	4481	2020/14	9,5	12	0,45	1550	5650	17,1
145 – 30/2	4500	4481	2020/14	9,5	12	0,45	1278	4530	[illegible]
145 – 30/1	4500	4481	2020/14	9,5	12	0,45	1006	3600	18,0
145 – 33/3	4500	4481	2020/14	9,5	12	0,45	1550	5090	17,2
145 – 33/2	4500	4481	2020/14	9,5	12	0,45	1278	4075	[illegible]
145 – 33/1	4500	4481	2020/14	9,5	12	0,45	1006	3255	[illegible]
145 – 35/3	4500	4481	2020/14	9,5	12	0,45	1550	4660	[illegible]
145 – 35/2	4500	4481	2020/14	9,5	12	0,45	1278	3725	[illegible]
145 – 35/1	4500	4481	2020/14	9,5	12	0,45	1006	2970	[illegible]
145 – 37/3	4500	4481	2020/14	9,5	12	0,45	1550	4240	[illegible]
145 – 37/2	4500	4481	2020/14	9,5	12	0,45	1278	3390	[illegible]
145 – 40/3	4500	4481	2020/14	9,5	12	0,45	1550	3560	[illegible]
145 – 40/2	4500	4481	2020/14	9,5	12	0,45	1278	2845	[illegible]



f = 8,0 mm angesetzt.

Table 7 Water Consumption and Effluent Figures (8)

Type of consumer	Water consumption and effluent
Drinking, cooking, cleaning	25 – 40 litres/person/day
Washing	10 - 15 litres/person/day
Toilets	10 - 20 litres/flush
Bath	200 – 250 litres/bath
Shower bath	50 - 80 litres/bath
Car wash	80 – 100 litres
Lorry wash	200 – 300 litres
Market gardener +)	2,5 – 3,5 litres/day/m^2 cultivated area
Garden spray, lawn +) (private)	2 - 3 l/m^2 + rainfall
Garden spray, vegetables +) (private)	5 – 10 l/m^2 + rainfall
Cattle	30 – 40 litres
Sheep, etc.	8 – 10 litres
Schools	3 – 5 litres/scholar/school/day
Barracks	100 - 150 litres/person/day
Hospitals	400 - 600 litres/bed/day
Boarding houses, hotels	100 – 130 litres/guest/day
Restaurants	20 – 30 litres/meal
Swimming pools	500 litres/day/m^3 pool capacity
Slaughterhouses	300 – 400 litres/animal
Laundries	40 – 60 litres/kg laundry
Bakers	140 litres/day/employee
Hairdressers	170 litres/day/employee
Photographers	300 litres/day/employee
Dairies	4 - 5 litres/litre milk
Breweries	14 – 20 litres/litre beer
Paper mills	1500 – 3000 litres/kg paper
Cellulose factories	80 – 90 litres/kg cellulose
Sugar refineries	1800 litres/100 kg beet
Mining	1000 – 3000 litres/ton produced
Steelworks	7000 - 12000 litres/ton pig iron

+) Deduct when calculating effluent flow

Table 8 Rainwater precipitation rates and impermeability factors for 15-minute rainfall periods together with water flows for land drainage (9)

Rainwater precipitation :	
North-West Germany	85 litres/s/hectare
North-East and Central Germany	95 litres/s/hectare
West Germany	96 litres/s/hectare
Saxony and Silesia	106 litres/s/hectare
South-West Germany	119 litres/s/hectare

Corresponding impermeability factors :	
Very heavily built-up areas	o,7 - o,9
Heavily built-up areas	o,5 - o,7
Continuously built-up areas	o,3 - 6,5
Dispersed buildings	o,2 - o,3
Open country	o,1 - o,2
Sports grounds and parks	o,05 - o,1

Land drainage :	
Small polders	o,9 litres/s/hectare
Large polders	o,7 litres/s/hectare
With heavy addition of drainage water	1,7 - 2,0 litres/s/hectare

Abstand .... = Distance 1 between bearings (m)

Belastbarkeit .... = Load-carrying capacity P for f = s_{sp}

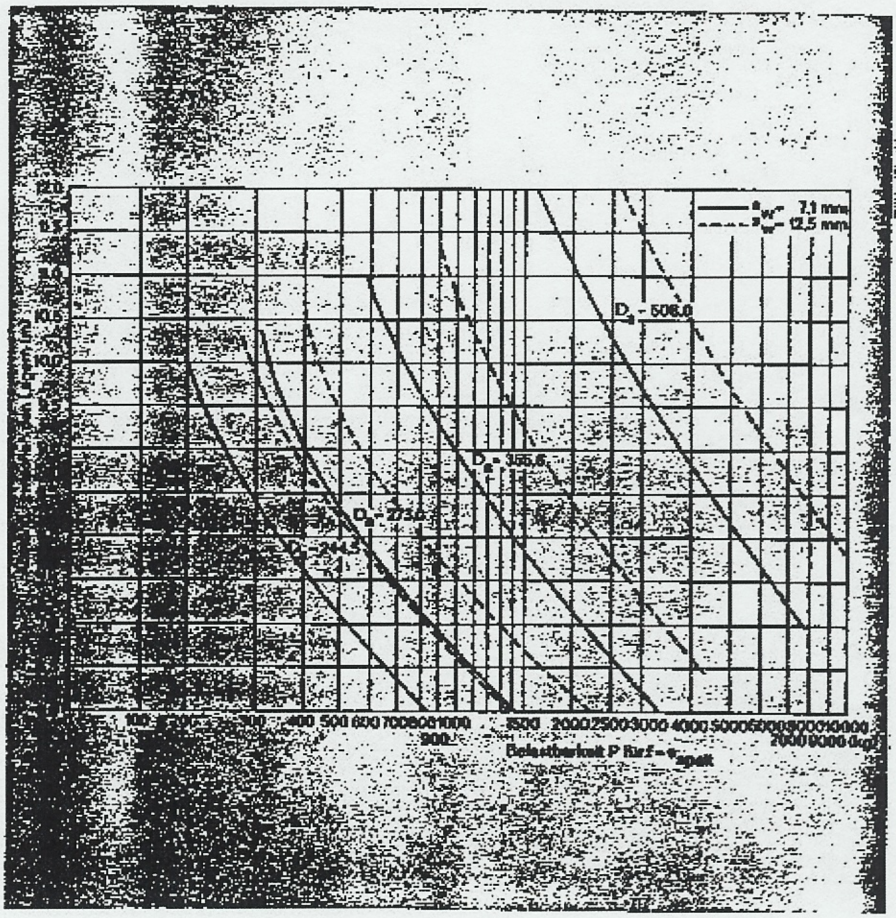


Diagram 1

Load-carrying capacity of shaft tubes as a function of bearing spacing

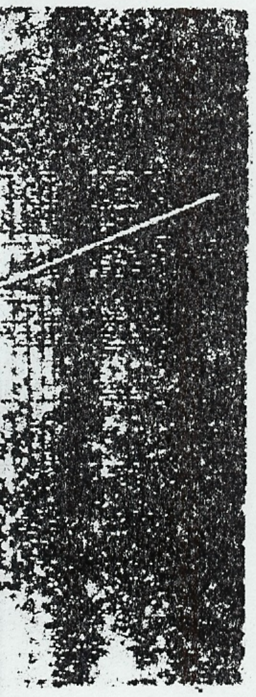
Überfall höhe = Crest height h in cm

Wassermenge ...... = Water flow in m^3/h

$$Q = 8/15\,\mu\,.\ \tan\ \alpha/2\ \sqrt{2g}\ \ h^{5/2}$$

where $\mu = 0{,}565 + 0{,}0087\ h^{1/2}$

(h in metres)



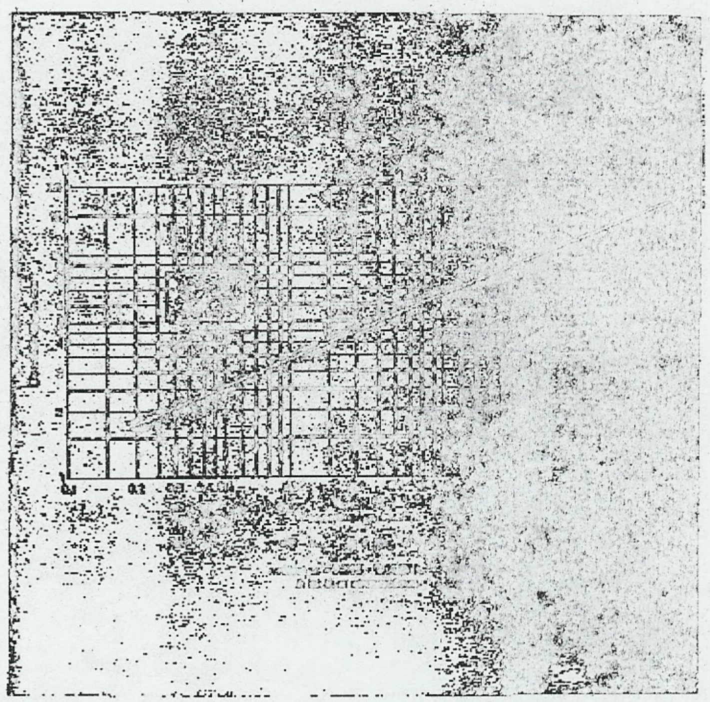


Diagram 2

Rate of flow over a right-angle triangular weir for small flows (10)

h_e in cm = h_e in cm

Wassermenge .... = Water flow Q in m^3/h

$$\boxed{Q = 2/3 \mu b \sqrt{2g}\, h_e^{3/2}}$$

where $h_e = h_o + 0{,}0011\,m$ = Equivalent crest height

$\mu = 0{,}6035 + 0{,}0813\, h_e/p$

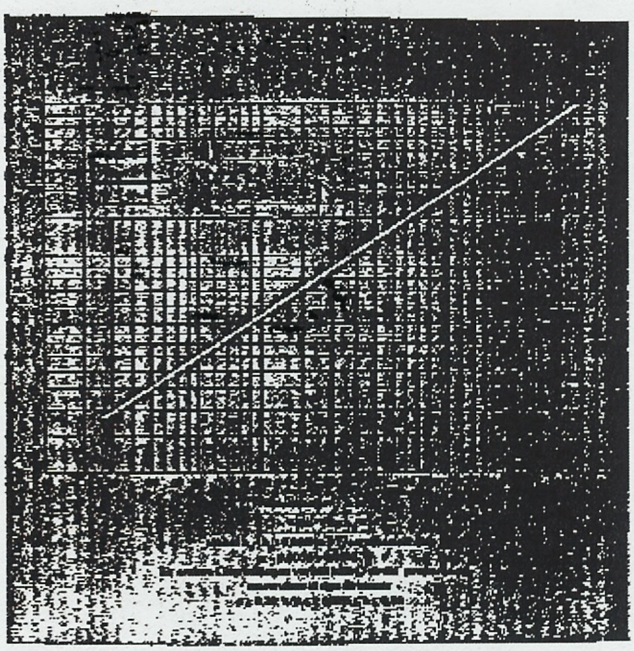
Q will be directly proportinal to weir length.

Applicable within limits $p \geq 0{,}30$; $h/p \leq 1$;

$0{,}025 \leq h \leq 0{,}80$

Diagram 3

Rate of flow over a sharp-edged rectangular weir for b = 1,0 m without end contraction (10)



Abflußmenge = Flow rate
Gefälle = Gradient
Fülltiefe = Depth of water

1-2=3 read across horizontally to chosen breadth B to give H. H in cm

Beispiel = Example
given

Q = 500 litres/s
i = 1 : 600
B = 1,0 m
find H.
H = 43,5 cm

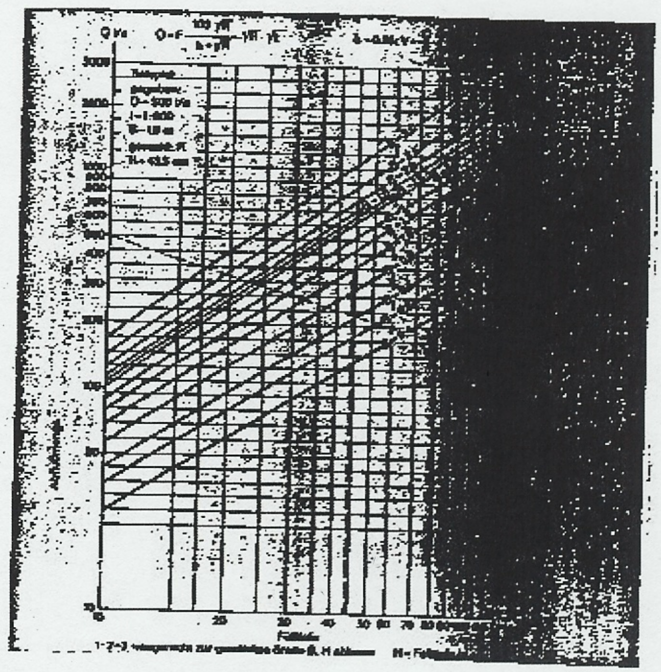


Diagram 4

Rate of flow through a rectangular channel from the depth of water (10)

Gefälle = Slope

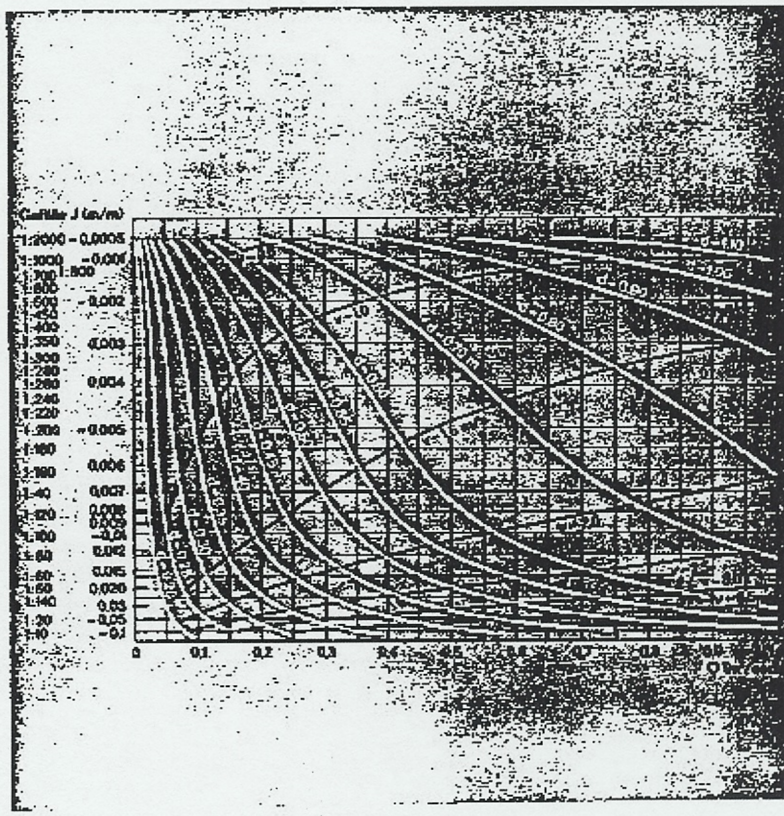


Diagram 5

Rates of flow and corresponding speeds
in circular-section drain pipes (9)

Leis..ngsdiagramm ... = Performance diagram for Archimedian screw pumps of Series 11 (without speed corrections)

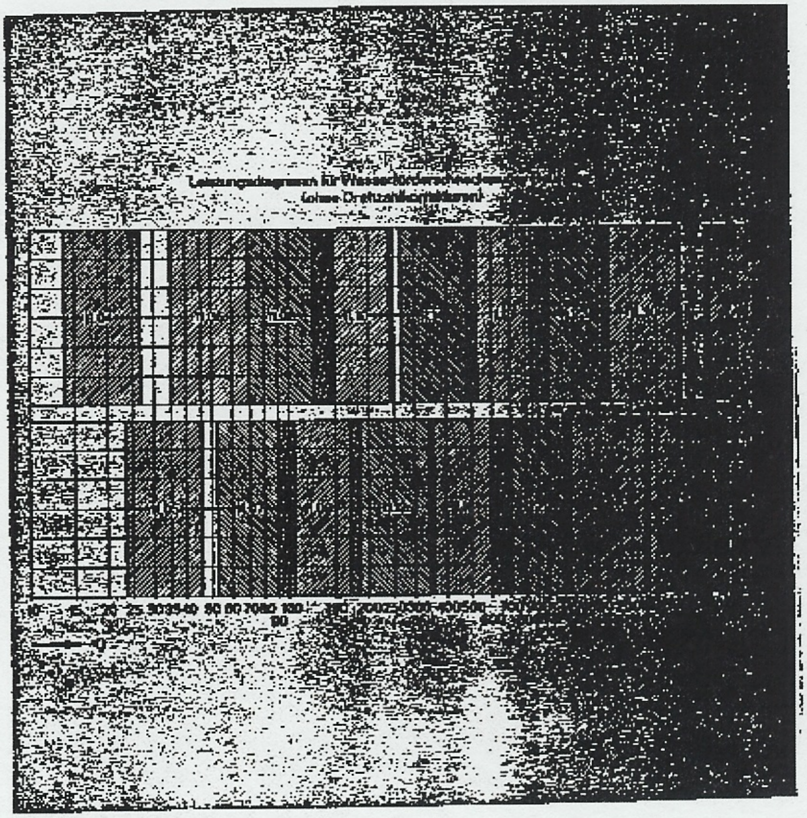


Diagram 6

Performance diagram for Archimedian screw pumps of Series 11 at n_{max}

PART 6 References and LBibliography

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(2) Horch, I.G. — Die Wasserförderschnecke (The Archimedian screw pump) "De Ingenieur" (1916) pp. 945 et seq.

(3) Bekkering — Fliessgeschweindigkeiten und Drehzahlen an Wasserförderschnecken (Rates of flow and speeds of Archimedian screw pumps), referred to in (1).

(4) Delprat — "Werktuigkunde" (Mechanical engineering) KMA, 3rd Edition, 1961, pp. 242 et seq.

(5) Nagel, G. — Die LWasserförderschnecke als Vorflut- und Abwasserpumpe (The Archimedian screw pump as a drainage and effluent pump) "[illegible]" Vol. 1[illegible] (1959), No. 52. Berechnungsgrundlagen für Wasserförderschnecken (Basis for the calculation of Archimedian screw pumps).
Die Archimedische Schnecke als Vorflut- und Abwasserpumpe (The Archimedian screw pump as a drainage and effluent pump) "Die Umschau" (1960), No. 16, p.476.

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(7) Storm, Buyzing — "Werktuig und Scheepsbow" (1851) Modellversuche an Wasserförderschnecken (M[illegible] tests on Archimedian screw pumps).

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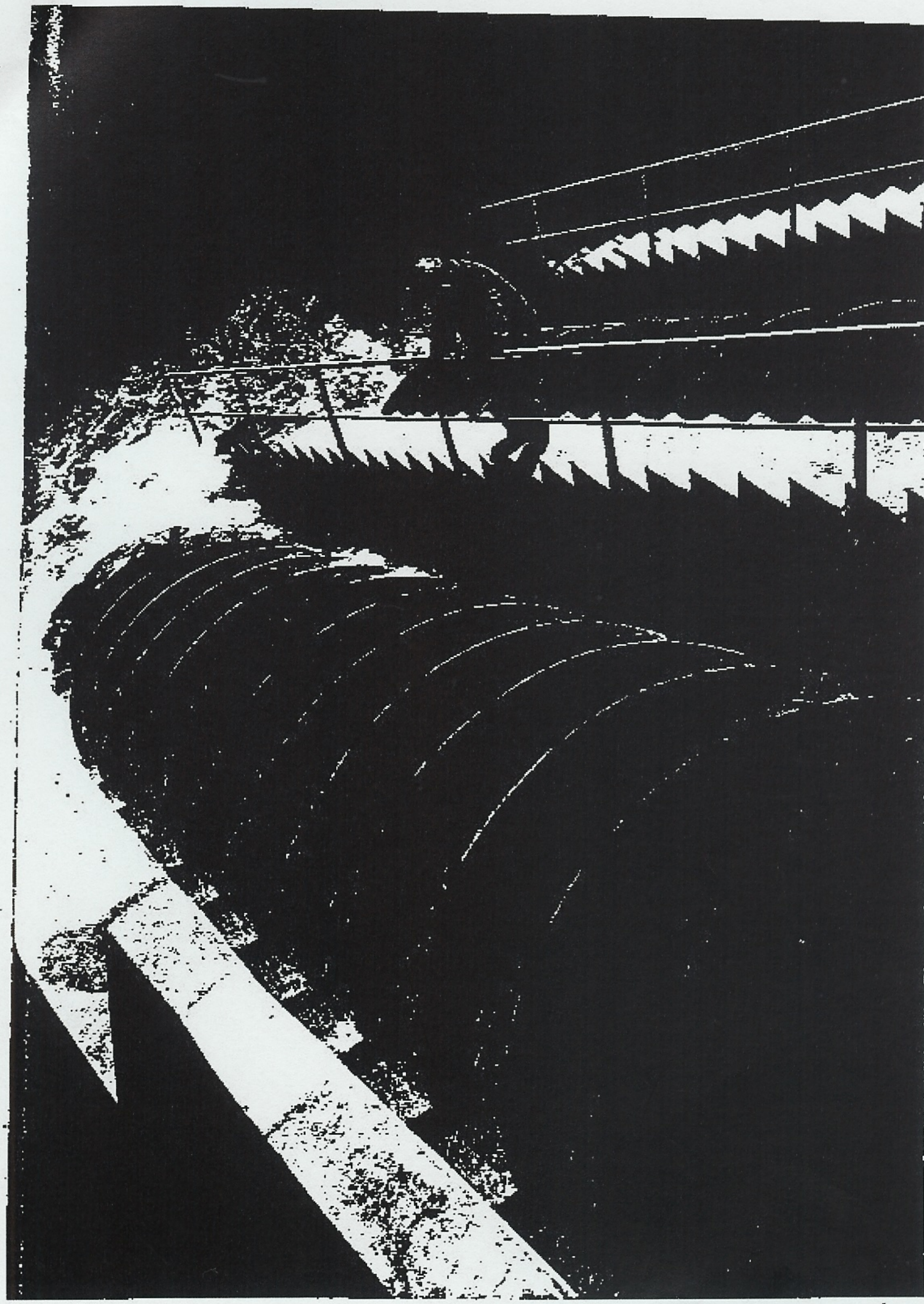
PART 7 ALPHABETICAL SUBJECT INDEX

PICTURES AND DRAWINGS

TABLES

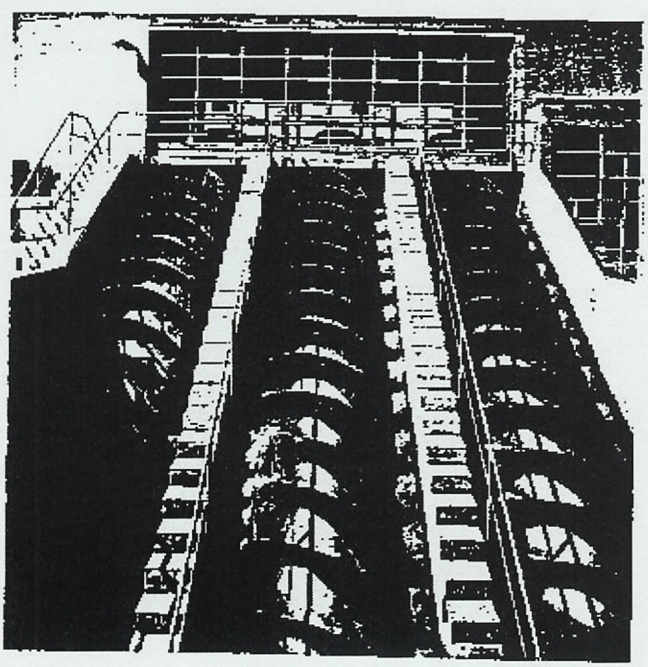
DIAGRAMS

Immersion above the filling point
of a Type 1120 rainwater screw pump,
three-start, 2000 mm dia.,
length of blade 10 360 mm,
installed in an Archimedian screw
pumping station at Ludwigshafen a. Rh.;

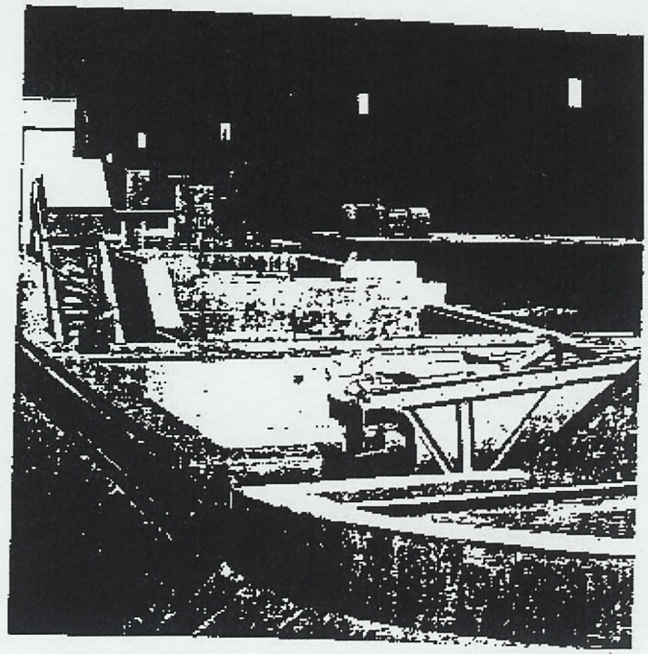


Manufacture of screws
-welding blade onto shaft tube.

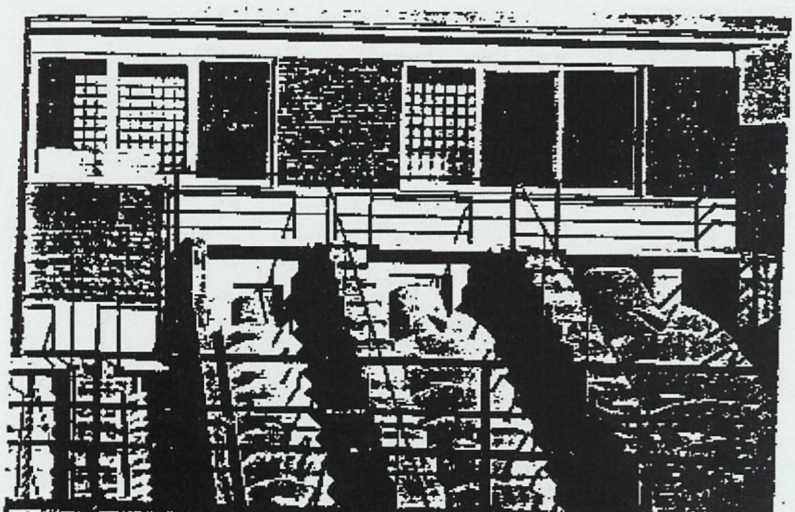
Combined effluent and rainwater pumping station
:: Ludwigshafen a. Rh., equipped with
two effluent screw pumps,
Types 1105 and 1106,
delivery rates 40 and 80 litres/s,
delivery head 4.9 and 4.75 m,
and with three rainwater pumps,
delivery rate 1100 litres/s each,
delivery head 4.9 m.



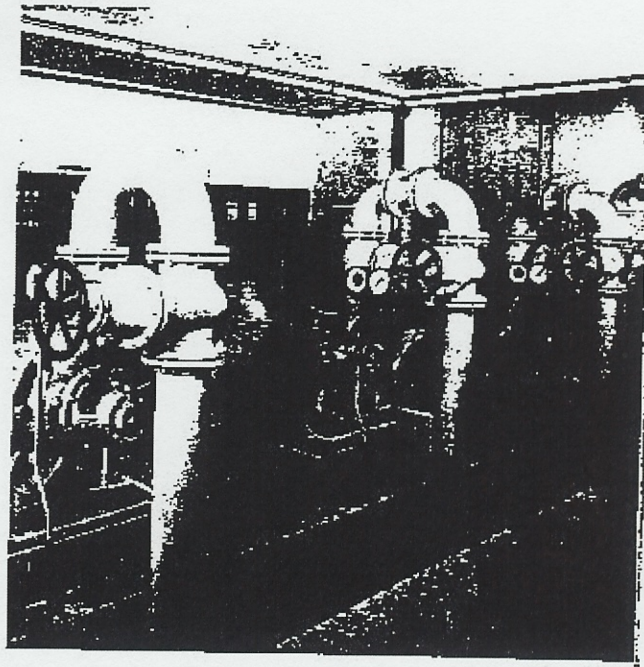
Portable screw pumps mounted on self-contained steel frames with drive unit.
Delivery rate variable from 1.5 to 8 litres/sec,
delivery head variable from 0.25 to 1.28 m.
Installed in research treatment plant of the Stuttgart Technical University at Stuttgart-Büsnau.



Archimedian screw pumping station Hirschau with two effluent pumps,
Types 1104 and 1105,
delivery rates 35 and 65 litres/s,
delivery haed 3.85 and 3.60 m,
and with three rainwater pumps,
Types 1112, 1116, and 1120,
delivery rates 350, 780, and 1270 litres/s,
delivery head 3.61, 3.20 and 2.83 m.



Pressure boosting station for the water supply system at Langen with four multi-stage, high pressure centrifugal pumps.
Total delivery rate about 800 m3/h, delivery head about 50 meters head of water.



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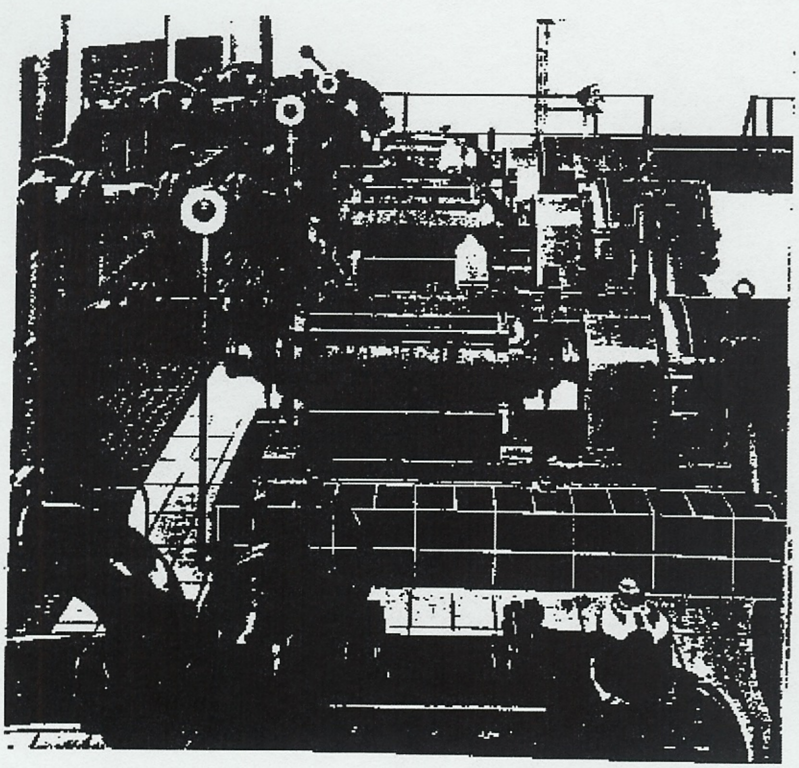
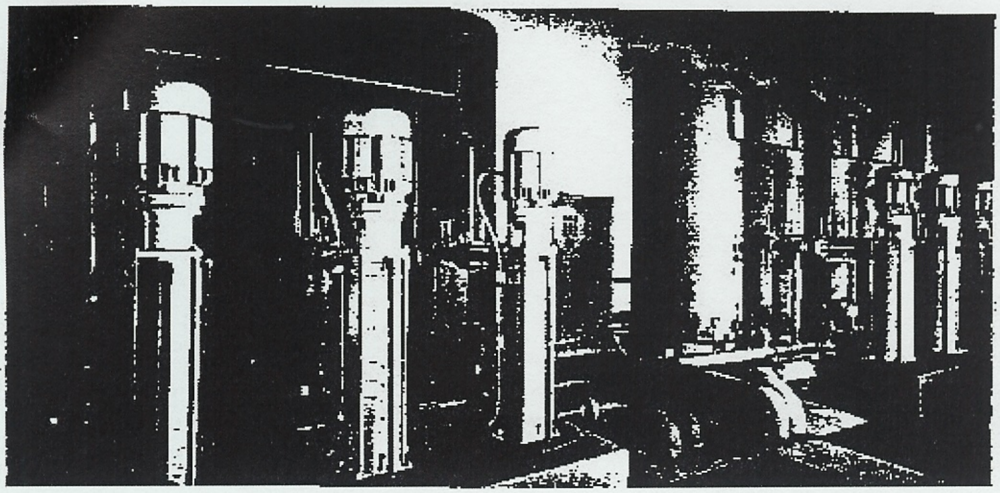
New waterworks for the municipality of Freising/Obb.,
Pump group "Hochzoll-Weihenstephan".
Delivery rate total 1300 m^3/h approx., delivery head 45 or 49 metres head of water.



Pressure boosting station with
six vertical, high pressure centrifugal pumps,
Type O, 10-stage, for the central
water supply of a high-rise office block
at Bordeaux/France.



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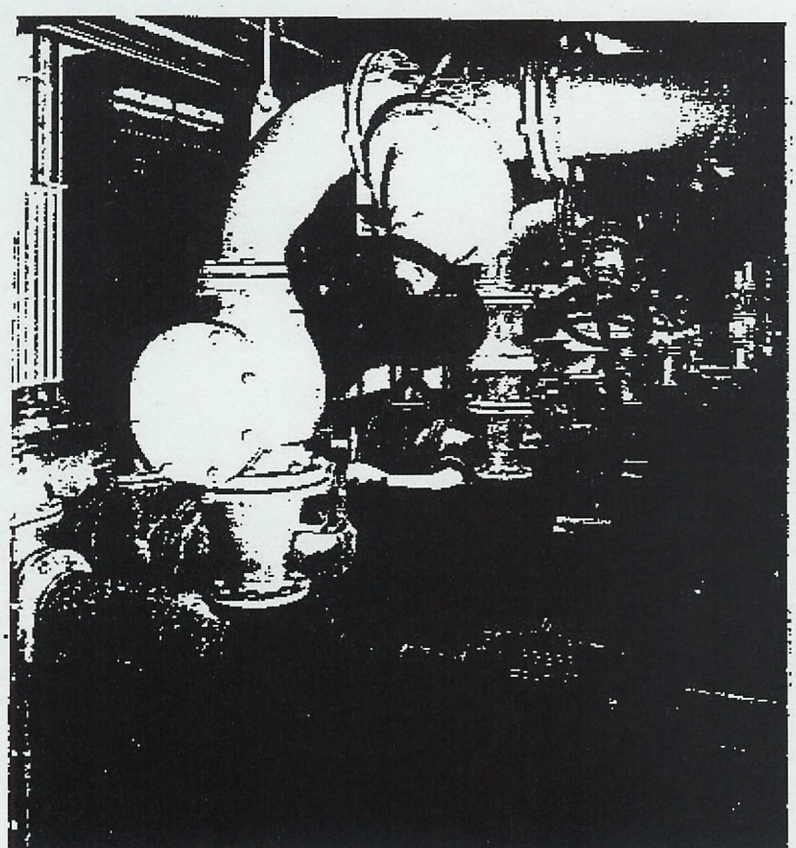
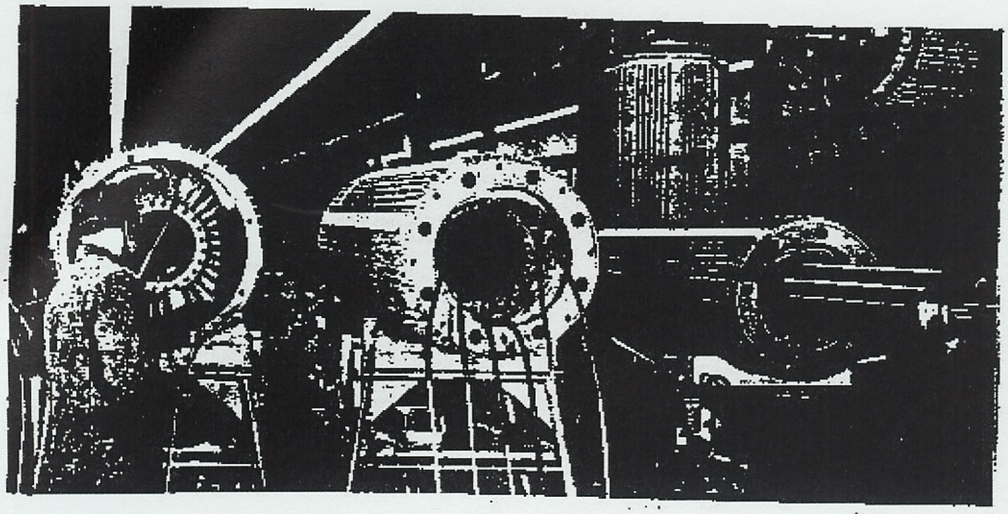
Air-conditioning plant with
high pressure and low pressure
centrifugal pumps in a large paper mill.



Manufacture of a batch of RITZ high-voltage submersible motors, each rated at 1360 h.p.;



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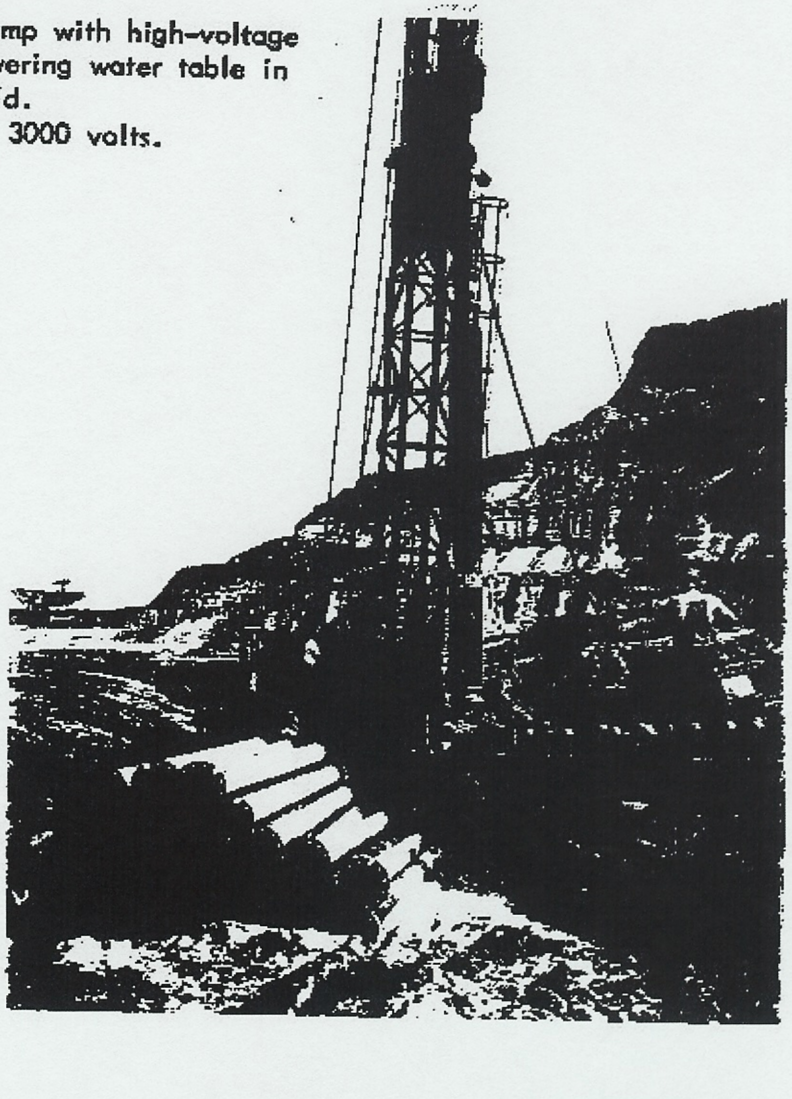
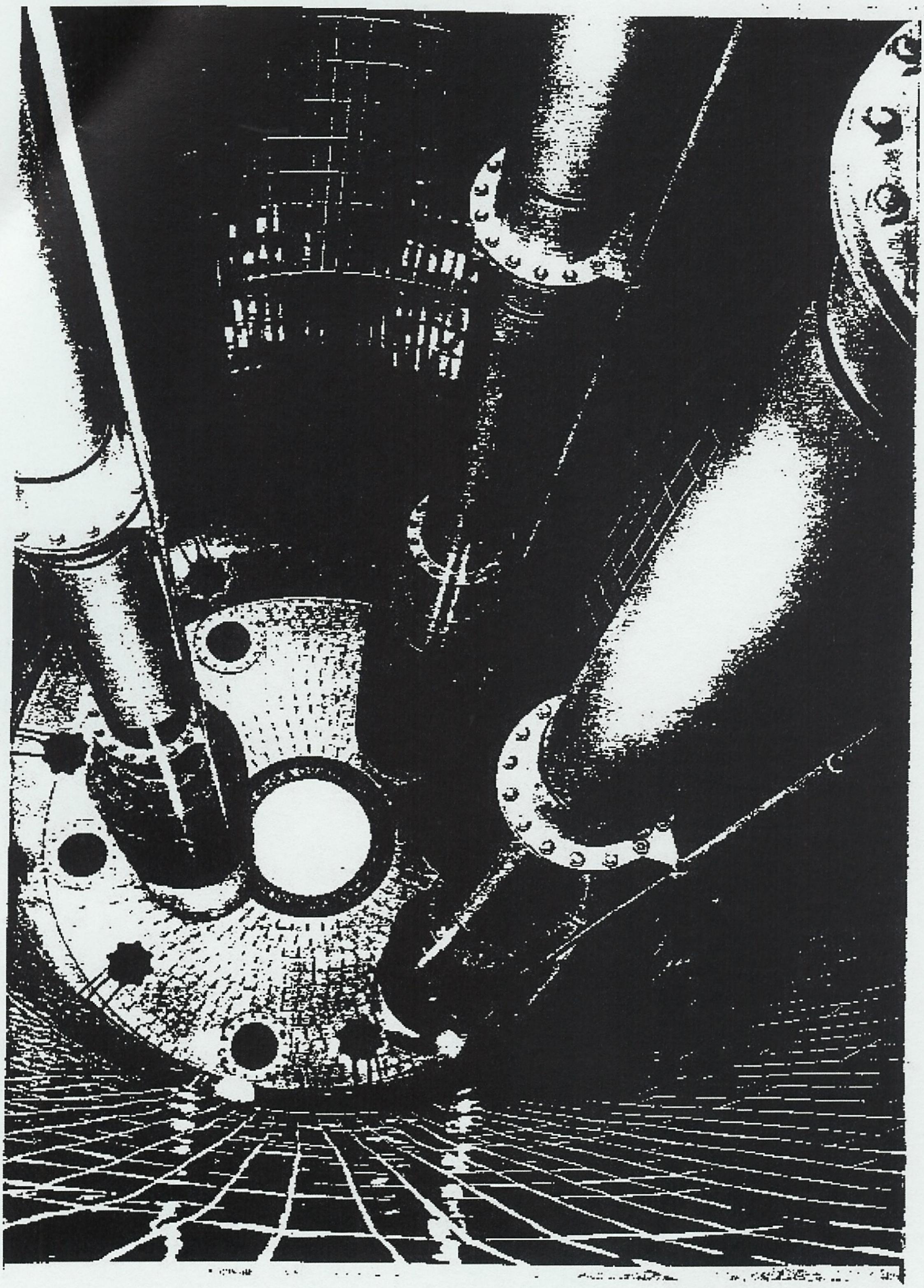
Installation of a RITZ pump with high-voltage submersible motor for lowering water table in the Rhineland lignite field.
Motor rating 1130 h.p., 3000 volts.



Horizontal well for the Munich municipal
water system with three double-volute
RITZ submersible motor pumps.
Total motor rating 1170 h.p., 3000 volts.



THE "RITZ" PROGRAMME

For clean water

NORMA Range -
Single-stage volute-casing pumps
to DIN 24 255 with bearing brackets,
pedestal bearings, and in monobloc form.

Multi-stage high-pressure centrifugal pumps
of horizontal and vertical type

Multi-stage high-pressure centrifugal pumps
of compact, upright type with flanged-on
motors

Volute-casing pumps for large delivery rates,
single and double-volute form,
with horizontally or vertically split casing

Submersible motor pumps,
single and double-volute

For contaminated water and effluents

Screw pumps of horizontal, vertical,
and upright type with flanged-on motor

Tubular screw pumps

Enclosed-impeller pumps with single and
double impellers, of horizontal, vertical,
and upright type with flanged-on motor

Contractors' submersible pumps

Floodable cellar drainage pumps

Archimedian screw effluent pumps