Notes on the Theory of Centrifugal Pumps.

Paper read by Mr. Geo. Higgins, September 7th, 1900.

The writer does not propose to give an exhaustive investigation into the theory of centrifugal pumps. Many points which are dealt with in text-books will be omitted. Possibly, indeed, all that is here stated may be quite familiar to some who are present. Nevertheless, in view of the variations in practice amongst the manufacturers of centrifugal pumps, and in view of the different explanations given of their action by various modern authors, it is thought that some advantage may result from bringing the matter before the Institute; indeed, it is hoped that the reading of these notes will accomplish three objects, viz.:

1. To direct attention to the most important points connected with the design of centrifugal pumps, eliminating those of minor importance, so as to avoid confusion.

2. To indicate what experiments are needed for the elucidation of some doubtful points, and

3. To rouse such criticism and discussion as may have the result of placing the theory on a firmer basis than on that which it has hitherto rested.

Without much error, the runner of a centrifugal pump may be considered as creating what Rankine calls a "forced vortex" within it. Rankine, however, dealt only with runners having vanes placed with their outer tips radial; but as runners are frequently made with vanes having their outer tips inclined to the circumfer-
ence at various angles, we must consider the tangential velocity of the water and deal with it as distinct from the velocity of the runner.

Consider a runner, full of water, discharging W lbs. of water per second from its periphery.

Let \( \omega \) be the angular velocity of the runner.

- \( R \), its radius.
- \( V (\text{=} R\omega) \), the velocity of the periphery of the runner.
- \( v \), the velocity of the water at the periphery.
- \( v_r \), the velocity of the water relatively to the runner at the periphery. The direction of \( v_r \) is assumed to be that of the outer tips of the vanes.

\[ 45^\circ \] the angle which the outer tips of the vanes make with the tangents to the circumference.

- \( u \), the radial resolved part of \( v \).
- \( v_r \), the tangential resolved part of \( v \).

\[ v_r = V - u \cot \phi = V - v_r \cos \phi \]
The change of angular momentum per second is \( \frac{WR}{g} \).

The corresponding angular impulse per second is \( Pr \), where \( P \) is a certain force acting in a circle at a certain radius \( r \); \( P \) being that part of the total driving force which changes the angular momentum of the water.

Then \( Pr = \frac{WR}{g} \).

The work done by \( P \) per second is \( Prw = \frac{WwR}{g} \) = energy imparted to the water per second.

The corresponding head is \( \frac{wR}{g} = \frac{wV}{g} \).

This expression, \( \frac{wV}{g} \), represents the total head due to the accelerating turning force \( P \). This head is represented partly by the pressure existing at the periphery of the runner and partly by the velocity head \( \frac{v^2}{2g} \), corresponding to the velocity \( v \) with which the water leaves the runner. The pressure referred to is due to centrifugal force; its amount is found as follows:

The surfaces of equal pressure being concentric cylinders, consider a small portion of an annular element contained between two concentric cylinders at radius \( x \) separated by a very small distance \( \delta x \). Let \( a \) be the area of the portion considered, so that its volume = \( a.\delta x \); its mass = \( \frac{G}{g} a.\delta x \), where \( G \) is the weight of a cubic foot of water.

Let \( y \) denote the tangential velocity at radius \( x \), then

\[
y = \frac{w}{R} x\]

The centrifugal force of the element considered is

\[
\frac{G}{g} a.\delta x \cdot \frac{y^2}{R^2} = \frac{G}{g} a.\delta x \cdot \frac{x^2}{R^2} \delta x.
\]

If \( \delta h \) represent the variation of head of pressure in the distance \( \delta x \), then \( G.a.\delta h = \frac{G.a.\delta x.\delta x}{gR^2} \).

\[
\therefore \frac{\delta h}{\delta x} = \frac{w^2 x}{gR^2}
\]

\[
\vdots \quad \text{head of pressure at periphery } h_c = \frac{w^2}{2g}
\]

It is important to notice that the velocity head \( \frac{v^2}{2g} \) is greater than the head due to centrifugal force, viz., \( \frac{w^2}{2g} \). Certain pumps are so constructed that the energy due to \( v \) is nearly all dissipated in eddies and friction.
In choosing an engine to drive a centrifugal pump, it must be borne in mind that, over and above the power required to do the work represented by \( \frac{WwV}{g} \), power is further required to overcome frictional resistances, including the frictional resistance to the motion of the water through the runner and also that due to the rotation of the cheeks of the runner in water, as well as the friction in bearings, etc.

Suppose that different runners are experimented with, their speeds being so adjusted that in the case of each of them the total head \( \frac{wV}{g} \) is the same; that is to say, \( w \) and \( V \) are caused to vary inversely to each other so that their product is constant. It is evident that the way to have \( w \) small and \( V \) large is to construct a runner with its vanes much recurved; that is to say, with the angle small, and \textit{vice versa}. Now, if the energy corresponding to \( w \) is to be wasted, it is manifestly desirable to make \( w \) as small as possible, other things being equal. We are prevented, however, from carrying this beyond a certain limit, because, if \( V \) exceeds a certain amount, the energy lost in friction exceeds that saved by diminishing \( w \).

How can the energy corresponding to the velocity which the water possesses just before leaving the runner be saved? Rankine discusses the conditions existing in a free vortex—spiral or circular—in which the particles are free to pass, with a slow radial movement, from one circular current to another; and in doing so assume the velocities proper to the currents entered by them without the action of any force but weight and fluid pressure, the total head remaining constant, i.e., if \( h_1 = \text{total head in a free vortex} \) and \( \nu \) and \( h \) the velocity and pressure at any point in it,

\[
h' = \frac{\nu^2}{2g} + h.
\]

If it is possible for such a vortex to exist, it is easily proved, as explained by Rankine, that the velocity at any point in it is inversely as the distance from the axis.

Assuming the existence of such a free vortex in the casing outside the runner, the way to proceed to find the suitable dimensions of the casing, near the point where the water leaves it, is as follows:

Let \( v_3 \) represent the velocity which is required in the discharge pipe, and suppose that it is found convenient, having regard to considerations of space and ease of construction, to gradually taper the discharge pipe near where it joins the casing so that a velocity \( v_1 \) in the casing may be gradually, and without loss of head, changed to \( v_3 \). Now, if the casing be so shaped that the water circulating in it circulates as a free vortex, the mean radius of the whirling space near where the discharge pipe is joined to it
to be made = $R^{20} \over \nu s$.

Experiments are needed to determine to what extent we are justified in assuming the possibility of the existence of a free vortex in the casing outside the runner. Simple preliminary experiments could be performed with a casing designed in the manner above indicated by inserting a number of pressure gauges at various parts of the whirling space, extending in rows from the periphery of the runner onwards. These gauges would indicate to what extent the pressures vary with the radii, and, as the relation between velocity and pressure is a simple one, we could ascertain how nearly the actual whirling approximates to a free vortex. Unwin recommends that a spiral discharge chamber should be employed in addition to a moderately large whirlpool chamber, adding that “The passage between the whirlpool and discharge chamber should be so accurately proportioned as to secure as far as possible a uniform flow all round from the whirlpool chamber into the discharge chamber. If this passage is of varying section, the regular motion in the whirlpool chamber would be more or less broken up.” He does not give a drawing illustrating what he means. Professor James Thomson, who was one of the first to demonstrate the importance of providing whirling space, causes the water on leaving the whirling space to change the direction of its radial velocity by 180 deg. This is probably done in order to gain compactness, and, as the radial velocity is usually not great, the effect will not usually be very injurious.

If a series of experiments with differently shaped casings enabled us to determine the form most suitable for one set of conditions, such information would not be of direct service in enabling us to determine the best form for a different set of conditions; but a series of independent investigations for different sets of conditions would certainly facilitate the designing of centrifugal pumps in general. First and foremost, it is necessary to ascertain whether the velocity which the water possesses at the moment it leaves the runner can be so gradually reduced in a whirling space as to avoid loss of head, or, if some loss of head must ensue, to ascertain the extent of such loss.

Experiments on existing pumps with ordinary spiral discharge chambers would show how the velocity varies in such chambers. It is certain that the velocity will diminish towards the outside; but it remains to be seen whether the pressure increases accordingly.

If it is proved that real advantage is derived from the employment of sufficient whirling space, then it will be evident that a casing should be specially designed for each particular set of conditions. Thus, for a high lift, the product $wV$ must be large.

The larger we make \( w \), the larger must we make the whirling chamber in order to get the necessary reduction of velocity from \( w \) to \( v_s \). If the discharge pipe is so high that the water simply stands in it without flowing—in other words, if the revolving water in the runner simply supports a column of water in the discharge pipe—then the casing would require to be of an impossibly large size in order to reduce \( w \) to \( o \).

Generally speaking, for convenience of construction the vanes should be re-curved more for high lifts than for low ones.

If the wasting of the energy corresponding to the velocity \( v \) can be avoided, then it is a matter of no importance as regards efficiency what angle the outer ends of the vanes make with the circumference. In some places one's choice of engines may be limited. For instance, an engine may be procurable which will develop the necessary power when running at high speed, while it might not be possible to procure an engine which would develop the same power at low speed, or vice versa. In the former case we should then have to decide whether a small runner with radial-tipped vanes should be employed or a larger runner with backward-curved vanes. What is essential is to get the product \( wV \) to equal a certain quantity, the angular velocity \( w \) being the same in each case. The larger the runner the greater the moment of the frictional resistances. Hence, if it can be managed, by employing a sufficiently large whirling chamber, to prevent the dissipation of the energy corresponding to \( w \), preference may be given to a radial-tipped runner.

Neglecting the thickness of the vanes, the radial velocity of flow at the periphery, viz., \( u \), is the same, for a given discharge and given diameter of runner, no matter what the shape of the vanes may be. If the change in radial velocity between entry and outflow is so sudden as to cause loss of head, the passages through the runner may be lengthened by curving the vanes backward, even if, ultimately, they again curve round to the radial direction, as shown in Rankine's sketch.

It has frequently been observed that when a centrifugal pump has been discharging through a very long pipe, and the pipe is suddenly broken close to the pump, the engine at once slows down. The explanation of this is that the quantity of water which has its angular momentum changed in a given time is greater in the case of the short than in that of the long pipe, and thus, when the pipe breaks, more work is thrown upon the engine.

Most writers on the subject of centrifugal pumps show how to calculate the effect of abrupt changes of velocity during the passage of the water through the runner. It is hardly necessary to say that abrupt changes of velocity should be avoided as far as possible, and common sense will generally enable a designer to reduce losses from such causes to a minimum. Practically all
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Mr. W. Stone (the President) regretted that the paper had not

When the subject show how to design the inner tips of the vanes so that they shall cleave the water, the angles these tips make with the radii having reference to the relative velocities of the water and the runner. Unwin shows how the ordinary principles of hydraulics can be applied to determine the loss of head when the vanes do not make the proper angles with the radii.

Authorities differ, however, in their suppositions as to what takes place at the periphery. Rankine, who pre-supposes the existence of a suitable whirling chamber, assumes that the velocity $v$ which the water has just before leaving the runner is gradually reduced to a smaller velocity of whirl as it moves further from the axis, and that pressure is gained corresponding to the lessening of the velocity. Unwin, although he deals with whirlpool chambers at the end of his paper, yet, in his general treatment of the subject, as well as in his treatment of a pump with a spiral discharge chamber, assumes that there is an abrupt change from $v$ to a lesser velocity of whirl $v_s$. He calculates that the gain of pressure due to this lessening of velocity is a maximum when $v_s = \frac{1}{2} v$. But he altogether neglects the velocity head $\frac{v_s^2}{2g}$ Seeing that $v_s$ actually exists, the corresponding head should not be neglected in summing up the various results of the change of angular momentum.

With reference to the assumption made at the beginning of these notes, that the water in the runner may be treated as a forced vortex with peripheral velocity $w$, it is submitted that the error resulting from this assumption is very much smaller than that which may possibly arise from even a slight disproportion in the size and form of the pump-casing. More elaborate calculations may be made at leisure. In the meantime, the simplification of formulæ enables attention to be directed to the points which are of principal importance.

Summing up.—The energy required to impart the angular momentum $WwR$ to $W$ lbs. of water by means of a centrifugal pump is $\frac{WwV}{g}$

The theoretical lift for any given centrifugal pump will be that corresponding to the pressure existing in the discharge chamber just at the mouth of the discharge pipe, plus the velocity head corresponding to the actual mean velocity of the water at that place, minus the velocity head corresponding to the mean velocity of the water in the discharge pipe.

Discussion, October 3rd, 1900.

Mr. W. Stone (the President) regretted that the paper had not
been printed in time for the meeting. This was a very important subject, and he trusted when the paper was in the hands of members, it would be thoroughly discussed. Mr. B. Smith, a visitor, was present, and he had much pleasure in asking him to take part in the discussion.

Mr. B. A. Smith said that he had been approaching the subject from the opposite direction to Mr. George Higgins—viz., from the point of view of a maker of turbines, in which the water flows from a region of high pressure into the runner and thence into a region of low pressure.

If friction is neglected, and we reverse the direction of flow of the water and of rotation of the runner, then, provided that such reversal is possible—which is usually tacitly assumed, though experimental evidence is wanting to support this assumption—it is a matter of indifference whether we look on the machine as a turbine or a pump.

For the sake of simplicity, we may confine our attention to the case where the flow is in "two dimensions," as in a radial flow turbine or a pump with parallel cheeks, between which the guide vanes are placed.

There are two fundamental relations between the velocity of the water and the rate of change of the velocity as we proceed along the stream line and at right angles to it respectively, which must be satisfied at all points within the fluid.

The first expresses the condition of continuity of flow, i.e., if we consider a tube of flow bounded by two stream lines $A, B,$ and $A', B'$ as much water flows in at $A, A'$ per unit time as flows out at $B, B'$ per unit time. This is expressed by the relation—

$$\frac{\delta u}{\delta t} = \lambda u$$

where $u$ is the velocity of the water at any point $P,$ $\lambda$ is convergence of the stream lines at that point, $x$ is width of tube at $P,$ $\delta u$ is rate of change of velocity as we proceed along the stream lines.

The second expresses the condition that the flow is "irrotational"; i.e., if we imagine a small sphere of the water to become solidified, it will continue to move without rotation. This is clear if we consider that since there is no friction all the forces acting on the sphere pass through its centre, and, therefore, can have no tendency to turn it.

This relation takes different forms, according as to whether the
region within which the fluid is moving is at rest or is rotating (as in the runner of a turbine).

If the region is at rest we have \( \frac{du}{d\sigma} = \kappa u \quad (2) \)

Where \( u \) is the velocity of the fluid at the given point relative to the boundaries of the region—in this case it is the absolute velocity of the fluid—\( \kappa \) is the curvature of the streamline, \( \frac{dQ}{d\sigma} \) is the rate of increase of the velocity per unit distance towards the centre of the curvature of the streamline.

If the region is rotating with angular velocity \( \omega \) (positive when the rotation of the runner is opposite to the direction of motion of the hands of a watch) we have \( \frac{du}{d\sigma} = \kappa u + 2\omega \quad (3) \)

The \( u \) in this case is the velocity relative to the runner.

If, then, we consider a tube of the fluid, which is narrow at
every point, in comparison with the radius of curvature at that
point, we can, by means of (1) and (2) for a fixed tube or (1) and
(3) for a rotating tube, determine the velocity of the water at every
point in the tube.

In order that the circumstances of the motion may be the same,
whatever may be the position of the runner, we require that the
velocity should be constant, and that the angle which the stream
lines make with the radius should be the same at all points of the
inlet and outlet of the runner.

This condition is fulfilled above the runner if in this region the
water moves as a "free vortex," i.e., the motion is such that the
radial and tangential components of the velocity both vary
inversely as the distance from the axis. The stream lines in this
case are equiangular spirals, and the conditions (1) and (2) are
satisfied at all points of the region.

In order to approximate to this guide vanes are inserted, each
formed to the same equiangular spiral and placed at equal intervals
round the circumference.

Now, considering the moving region, and for simplicity sup-
pose the flow in the runner is radial at the inlet; since the
velocity is constant and radial at all points, the curve $\sigma$ coincides
with the inlet boundary, and we have along the inlet (since $u$ is
constant $\frac{du}{d\sigma} = 0$) $ku + 2w = \sigma$—(4).

i.e., the curvature of the stream lines in the runner is known near
the inlet.

Similarly the curvature of the stream lines in the runner near the
outlet is determined, but the result is rather more complex.

Again, since the velocity and the convergence of the stream lines
at the inlet and at the outlet are known, it follows from (1) that
$\frac{du}{ds}$, i.e., the change of velocity per unit distance along the stream line
is known at the inlet and outlet.

In choosing the form of vane we may therefore proceed as
follows:—

1. Lay out portion of the central stream line (between two of
the runner vanes) near the inlet.

2. Lay out portion of the same line near the outlet and plot
this portion in several positions, one beside the other.

3. Lay out a curve of known curvature at every point to join
the portion of the curve near the inlet and one of the
portions near the outlet (the elastica is a convenient curve
for this purpose).

4. Lay out a velocity curve, giving the velocity as a function
of the radius. This curve must pass through two given
points (corresponding to the known velocity at the inlet
and outlet) and have given directions at these points
(corresponding to the known rates of change of the
velocity at the inlet and outlet).
5. Calculate the width of the portion of the tube of flow on each side of the central stream line, using the velocity at each radius from the velocity curve, and the rate of change of the velocity at right angles to the stream line by means of equation (3) above.

6. Having the velocity at every point of the tube of flow we can calculate the pressure at the corresponding points by means of Bernoulli's equation—

\[ \frac{1}{2} \rho (u^2 - \omega^2 r^2) + \frac{p}{\rho g} = \text{constant} \quad (5). \]

It is to be observed that this equation only holds when the velocity is constant over the inlet and outlet.

The accompanying diagram represents one of the water passages in the runner of a radial flow turbine 3 feet external diameter and 2 feet internal diameter, to work under a head of 20 feet, running at 180 revolutions per minute.

The large amount of metal in the vanes is to be noted.

A certain amount of latitude is possible in fixing the widths of the tube, e.g., we might vary the velocities slightly, so as to increase the width of the narrower portions of the tube, but care must be exercised in doing this if we desire to avoid back flow in the runner.

The rapid variation of velocity across the section of the tube is remarkable.

The distribution of the pressure as indicated by the dotted lines of equal pressure on the diagram is interesting and points to the necessity of exercising caution in making assumptions as to the distribution of pressure in centrifugal pumps.

Mr. G. Higgins said this question had attracted a good deal of attention from a number of persons lately, and some of the latest text books were wrong, in his opinion. Before Professor Kernot left for England, he had given him quite a list of the more glaring errors in a work recently published by a Professor of Engineering in England to hand to the gentleman concerned. He also intended to send this paper to that Professor. Every effort should be made generally to elucidate this problem.

Mr. Stone proposed a vote of thanks to Mr. B. Smith for his contribution to the discussion. This question was a very difficult one, and also an important one, and it was only by those possessing the necessary ability, the time, and the opportunities to put their theories into practice, looking into these matters, that they could hope to arrive at a satisfactory solution of these questions. He hoped Mr. Smith would also try and make his explanation as simple as possible.

The centrifugal pump, if used on a high lift, was an excessively fine machine for dissipating energy, also the turbine, and they wanted to get the water through these machines without producing eddies.
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The vote of thanks to Mr. Smith was carried by acclamation. Mr. Smith briefly acknowledged the compliment, and stated he felt it a pleasure to come before the Institute. When a man took an interest in his work, he was only too glad to tell about it to an appreciative audience.
CENTRIFUGAL PUMPS.

Discussion on Papers by Messrs. George Higgins and E. Seitz.

MR. STONE noticed that in Mr. Seitz's paper, no attempt had been made to analyse the loss. The centrifugal pump was generally put down as inefficient for high lifts. If possible, a "balance sheet" should be shown giving the loss due to radiation, internal friction, etc.

MR. A. H. MERRIN said the mathematical treatment in Mr. Higgins' paper seemed to be on the most approved lines. The sectional flow throughout the runner was given as uniform; but he was doubtful as to the accuracy of that; and if the sectional flow was not uniform, the mathematical treatment would be completely upset. With water flowing in a converging pipe, the gain in velocity head would represent a loss in static head; but these two would not apply in the opposite direction in a diverging pipe. Some of the text books assumed that this was so. Should the vane start near the centre of the runner so as to avoid the shock of the water, and continue to re-curve until the tips became radial, it would completely upset the mathematical treatment. He agreed with Mr. Higgins' idea of a whirling chamber. He thought a complete series of experiments was needed in connection with centrifugal pumps. He had recently to make a few observations of a centrifugal pump. The ordinary engineering formula was adopted, viz., $V$ of the periphery was equal to

$$\sqrt[2]{gh},$$

which was nearer the truth for high lifts. The shape of this runner was a cross between Nos. 21 and 22 of Mr. Seitz's paper, with side discs, with pump case like No. 24. Size of the pump was 3ft. 6in. diameter; suction pipe 12in., with discharge of 40ft.

The first experiment was a lift of 25 feet from the top of the discharge pipe to the level of the water on the suction side. The revolutions of pump were 180 per minute. The water was merely held in the inclined pipe. The result was

$$V = 0.82 \sqrt{2gh},$$

Second case was: Lift of 29ft. 3in.; discharge, 400 cubic feet per minute. Vacuum gauge on the suction pipe to show the actual loss in velocity and friction head. Allowing for friction up the pipe, the total lift was 33'34 feet; revolutions of pump 229 per minute; mean of three observations gave

$$V = 0.9 \sqrt{2gh},$$
Third experiment: Pump discharging water; lift 31 ft. 6 in.; vacuum, 21 1/4 in.; 40 ft. of 1 1/2 in. suction pipe; 14 ft. 6 in. discharge pipe. Discharge was 570 cubic feet per minute; revolutions of pump, 245 1/2; total lift, 37 ft. Result:

\[ V = \frac{92}{\sqrt{2gh}}. \]

The last experiment was a discharge of 450 cubic feet per minute, and an unknown quantity of solid material. Vacuum, 24 in.; number of pump revolutions, 243 per minute; lift, 34 ft. 5 in. to 39 ft., taking velocity and friction heads into account. The result was:

\[ V = \frac{102}{\sqrt{2gh}}. \]

The incline of the vanes at the ends was 27°, which worked out pretty closely to the mathematical result that would be expected from such a runner.

Centrifugal pumps in mines, they would have to be made to accommodate themselves to varying loads, and a pump designed for a heavy load did not give the same efficiency when working with a light load.

Mr. J. T. N. Anderson said:—Regarding centrifugal pumps for mining. He had noticed a description of them in use in California, four or five pumps in a series. His own experience was that the efficiency of centrifugal pumps fell as the head increased. One noticeable feature about centrifugal pumps is that if the pump be run at a speed but slightly exceeding the speed which gives a fair discharge, the increase in the discharge is entirely beyond what would be expected from any theoretical consideration. Thus an increase from about \( 1 \frac{1}{2} \) revolutions per minute faster than the speed which gives the static head without discharge, will give perhaps four times the consumption of power, and an increase of a few more revolutions will give the most economical speed, viz., the speed at which the ratio of work done to power consumed is a maximum. There does not seem to be much difficulty in maintaining this speed, as the centrifugal pump has a wonderful power of accommodating itself to its work. The number of revolutions showing a very slight change with the rise or fall of the steam pressure. Speaking on the question of the forms of vanes. He thought that it is quite probable that the time will come when power can be transmitted directly from compressed steam or air to water, and so raise it, as the wind raises a water spout, entirely without the intervention of machinery.

Prof. Kernot said that all they wanted now was for some competent engineer with proper appliances, ample time, and unlimited means, to make a complete series of scientific and practical tests of all the different pumps which had been brought before the Institute, and give them the results. The mathematics on the subject were very conflicting and bewildering. Mr. Seitz had not been able to give them any apparently conclusive tests. The question as a whole was one of vast importance, and if the centrifugal pumps could be made to give an efficiency of 60 or 70 per cent, under ordinary conditions, it seemed to
him that it should replace pumps of all other kinds. As to the actual
efficiency obtained, a great many statements were made, but very little
proof was shown. Valuable tests had been made in America and else-
where with turbines of considerable size: could they not get such tests
with the centrifugal pump? He did not see why the centrifugal pumps
should not rival the results obtained by turbines, and would very
much like to see a practical experiment made. During his recent
trip he saw in a large dock at Glasgow some centrifugal pumps,
with high speed engines lifting the water up to a height of
30 feet; while at the southern outfall of the London sewerage scheme
he saw very slow moving plunger pumps lifting the sewage 25 feet.
These pumps had a great knack of wrecking their valves, and he saw a
large heap of wrecked ones close by, which had not taken long to
gather. He would like to ask if Mr. Seitz knew of any instance of large
sewerage pumps of low lifts working centrifugally?

MR. W. FYVIE stated that his experience of the centrifugal pump
was that it was anything but an efficient appliance. He agreed with
Prof. Kernot, that a series of experiments would be very valuable. The
centrifugal pumps should, in his opinion, be constructed for every 10 feet
or so to suit the conditions of pumping. He understood in the test
made by Mr. Seitz the delivery was 1,200 gals. He would like to know
the delivery at 270 feet. He did not think it would be practicable to
obtain a "balance sheet" of the actual results as suggested by Mr. Stone.
In looking over a number of tests in "Engineering," &c., he found
the best results given seemed to be 65% of i.h.p., and in no case was
the lift beyond 20 feet. This was with an engine of 400 horse-power.
The actual efficiency of this pump with a lift of 10 feet, was about 70%.
If these were the best results with a large pump, he did not think very
good results would be obtained with smaller ones.

MR. J. T. N. ANDERSON (President) considered it was a subject of the
greatest importance and very great economical interest. To his knowledge
more than £100,000 had been spent in Victoria alone during the last
couple of years on large mining pumps. If centrifugal pumps had been
used the cost would have been only one-fourth of the above amount. The
first cost was a very important point in mining work; sometimes a great
dead of efficiency had to be sacrificed to it. There was no certainty
about mining, and machinery might only be required for two or three
years. Different types of Cornish pumps were generally used for
mining purposes; but the valves had to be very accessible in case of a
sudden inrush of grit and sand. Centrifugal pumps, if properly designed
and put down in pairs in the shafts, seemed to be the ideal machinery
for mines; more especially where modern methods of working under-
ground by electricity obtained. By using centrifugal pumps they saved
the necessity for changing from a rotative form of using the energy, to a
rectilinear form. There was a great saving in gearing. He mentioned
the above to emphasise the points brought out by Prof. Kernot. The
mathematics in connection with the centrifugal pump after they left the
rudimentary facts, were largely speculative, depending upon hypotheses
which were not thoroughly worked out. There seemed to him to be no reason why the centrifugal pump should be less economical than any other pump. Up to a 25 ft. head the centrifugal pump should give as good an economy as any reciprocating pump. From actual trials he had got over 60% efficiency with centrifugal pumps, and the average returns they would get from reciprocating pumps at 25 ft. to 30 ft. head would not be so good. The easiest way to express the efficiency was on the duty. The Mildura pumps varied from 70,000,000 to 80,000,000 foot lbs. duty. At Rotterdam, with the biggest pump there (triple expansion Worthington), a test recorded by Simpson worked out to 82,000,000 foot lbs. With the large centrifugal pump at the Billabong Station, at Mildura, he had obtained a result up to 80,000,000 foot lbs. The Rotterdam pumps cost about three times as much as those at Mildura, and the difference between the efficiency was not more than 5%. The boiler pressure of the Rotterdam pumps was 90 lbs., and that of the Mildura pumps 130 lbs. To have used a high steam pressure in the former case with the low speed of the pump would not have given a much increased efficiency.

Mr. Seitz in reply apologised for not giving more tests and figures, but he was now making a series of tests, and hoped to be able to submit the results shortly. He wanted to give as correct information as possible. Re Mr. Merrin's contribution. He had not yet the exact data to go thoroughly into his figures, but would do so later on. As far as the formula \( \frac{1}{2} gh \) was concerned, he was of opinion that further investigations into centrifugal pumps would show that the above formula was the correct one. A number of both theoretical and practical points in connection with the centrifugal pump required to be solved. Professor Zeuner, of Zurich, the great authority on pumps and turbines, had admitted that the questions of proper velocities, proper proportions, and proper laws governing centrifugal forces in the whirling chamber and the runner were not yet finally settled. With Mr. Higgins' formula of \( \sqrt{gh} \) it would be found by experience that they could get results apparently confirming it, but in the case where a head was obtained with a tangential velocity less than calculated by the formula \( \sqrt{\frac{1}{2} gh} \) the energy was lost in the whirling chamber in the same way as in a hydraulic ram, where water can be raised to a much greater height than the velocity of a stream would indicate, as it is only obtained by utilizing a shock in the ram chamber and sacrificing efficiency to obtain simplicity. If he used a pump to lift water 100 ft. without discharging any water, the pump would give no efficiency at all, but still would require a great amount of work expended on it. If it discharged \( \frac{1}{4} \) of its rated quantity the efficiency would rise to perhaps 7 or 10%; if it discharged \( \frac{3}{4} \) it would rise to about 20%, and so on till with its full quantity it would give the maximum efficiency, which should remain almost constant from that to a velocity 2 or 3 times higher, leaving out the friction in the pipe itself. He thoroughly agreed with Prof. Kernot in all he said, and was sanguine as to the ultimate success of the centrifugal pump. Firms like De Laval, Sulzer, Gwynne and others would contract to supply plants to pump 1,000 ft. with good efficiency. He thought they
would in time get as good results from Centrifugal Pumps with high lifts as with other pumps. It was principally a matter of being able to make the velocities correspond with the conditions of the delivery head. In reply to Mr. Fyvie, he said the discharge at 270 ft. with 120 lbs. pressure was 40 to 50 gals. per minute; the nozzle was a \( \frac{3}{8} \) in. pipe. He would bring his books at next meeting to show the results obtained with \( \frac{3}{8} \) in., \( \frac{3}{4} \) in., 1 in. and \( 1\frac{1}{2} \) in., 3 in. and 4 in. pipes. The efficiency with the \( \frac{3}{8} \) in. discharge was 10 to 15%. He had results recorded where efficiencies of 70, 75, 83 and 87\% respectively were shown, but in one case where he got 93\% he put it down to an error in recording. In his trials the indicated horse-power shown by the engine running empty at the same revolution was deducted from the I.H.P. of full work, and the result was divided by the total water horse-power. In reply to Prof. Kernot, he said there were some large centrifugal sewerage pumps used in Holland and in Germany, of which he would give particulars when convenient to members at a future meeting.