

April 3, 1928.

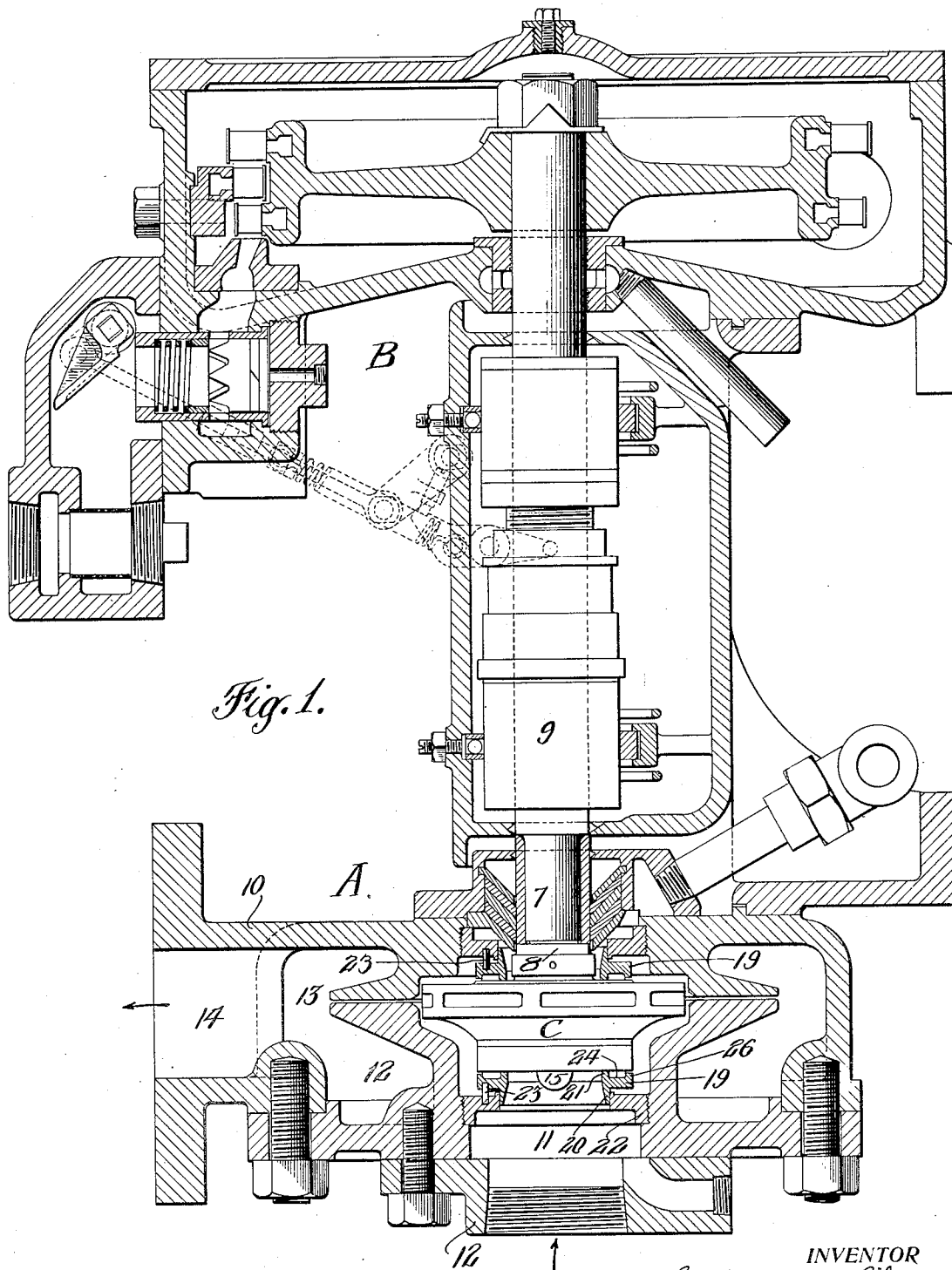
1,664,488

C. A. SCHELLENS

CENTRIFUGAL PUMP

Filed July 1, 1924

3 Sheets-Sheet 1



INVENTOR
Christopher A. Schellens
BY
Symmes & Lechner
ATTORNEYS

April 3, 1928.

1,664,488

C. A. SCHELLENS

CENTRIFUGAL PUMP

Filed July 1, 1924

3 Sheets-Sheet 2

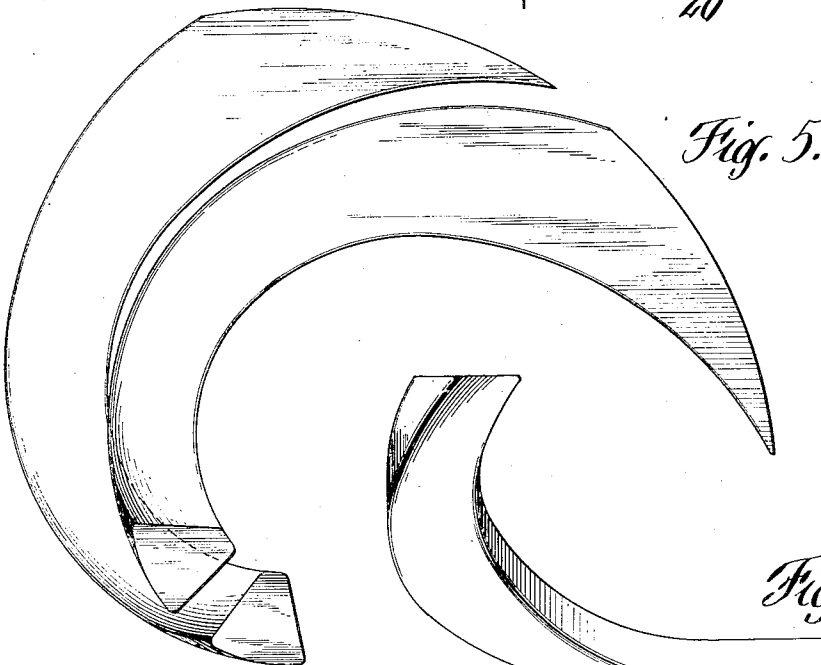
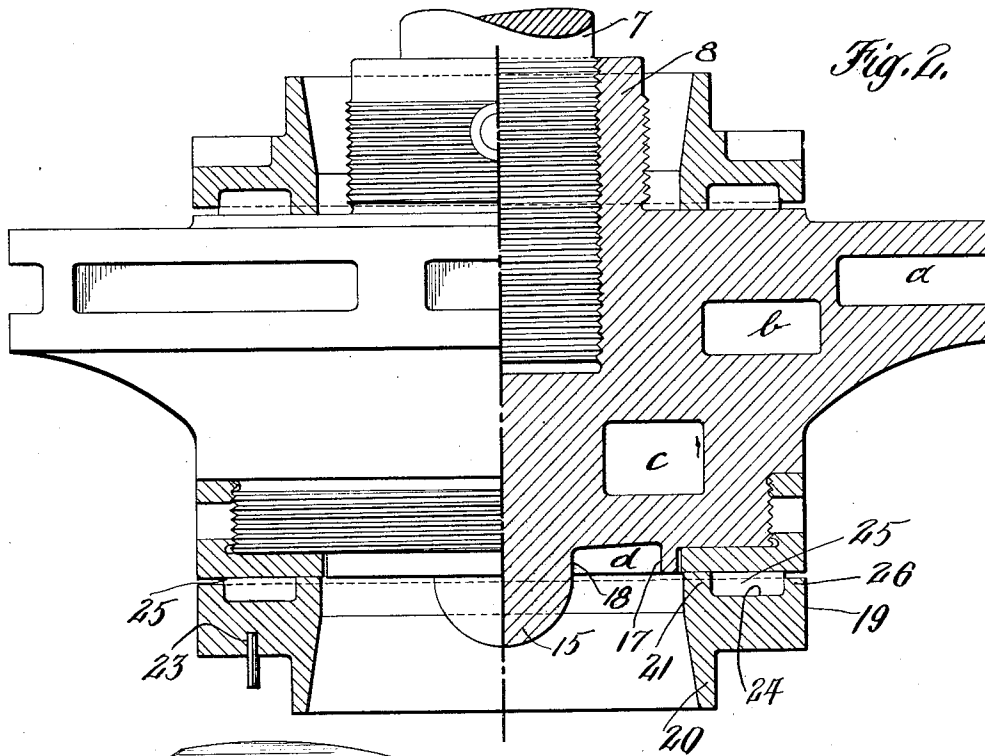


Fig. 6.

INVENTOR

Christopher A. Schellens
BY
Symmetrict & Lechner
ATTORNEYS

April 3, 1928.

1,664,488

C. A. SCHELLENS

CENTRIFUGAL PUMP

Filed July 1, 1924

3 Sheets-Sheet 3

Fig. 3.

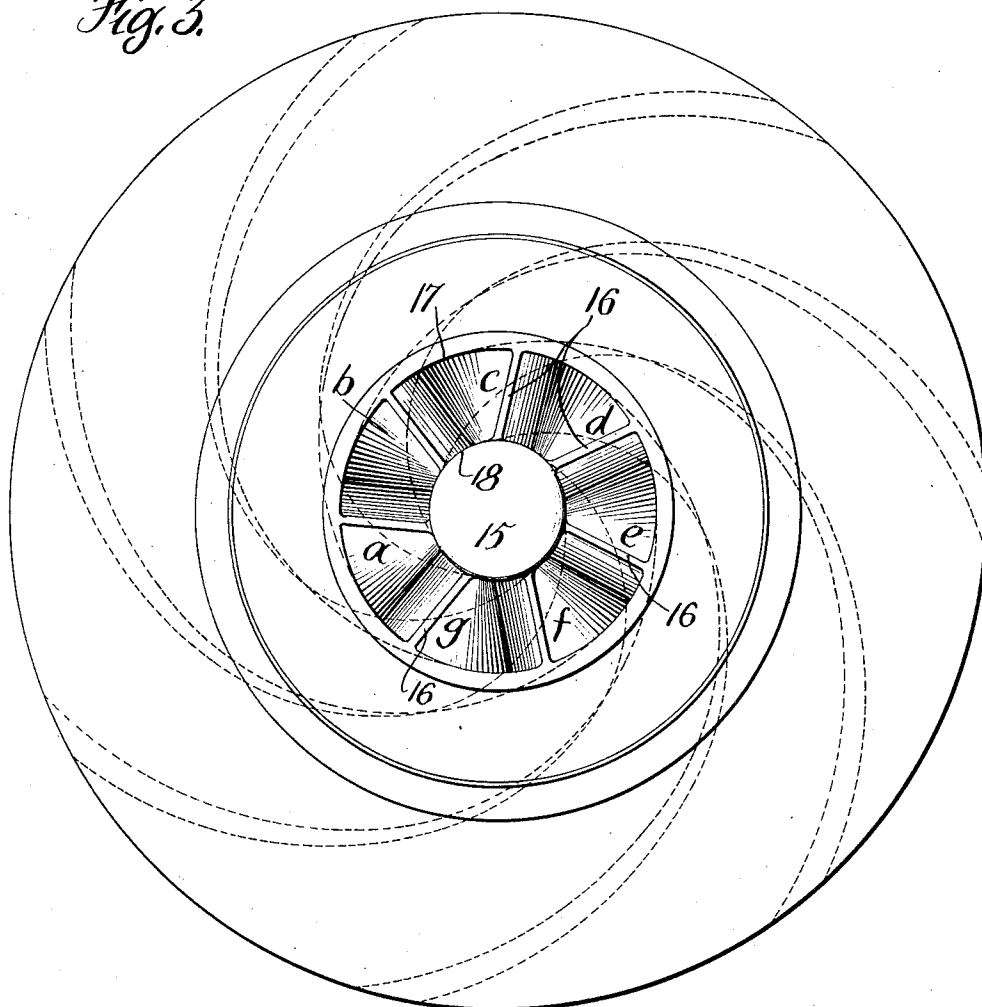
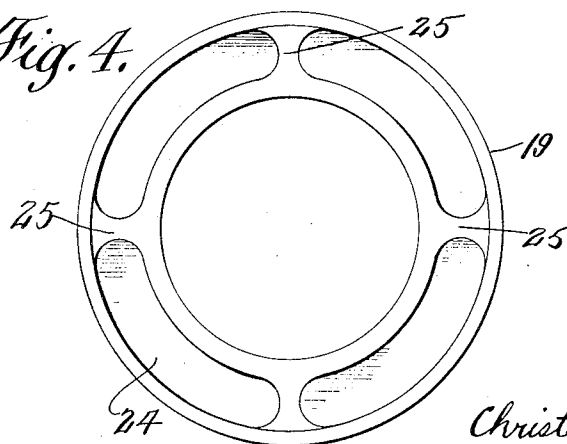


Fig. 4.



INVENTOR
Christopher A. Schellens
BY
Symmett & Lechner
ATTORNEYS

UNITED STATES PATENT OFFICE.

CHRISTOPHER A. SCHELLENS, OF MARBLEHEAD, MASSACHUSETTS, ASSIGNOR, BY
MESNE ASSIGNMENTS, TO C-S ENGINEERING COMPANY, A CORPORATION OF DELA-
WARE.

CENTRIFUGAL PUMP.

Application filed July 1, 1924. Serial No. 723,469.

The invention relates to centrifugal pumps, and is particularly useful as a boiler feed water pump for which use the pump is primarily designed and in connection with which it will be described.

Centrifugal pumps in their usual form are not well adapted to the small delivery volumes and high delivery pressures which are required in feeding locomotive boilers, or other boilers of relatively small capacity. In such boilers the pressure is comparatively high whereas the volume to be delivered is comparatively small.

For this purpose, I propose to provide a high speed centrifugal pump which will deliver a small volume at high pressure.

One difficulty incident to high speed pumps is the occurrence of cavitation at the runner inlet, for the water, due to the high speed of the entrance edges of the blades, is not able to follow the contour of the passages where it enters the runner and voids are formed in the stream. This causes a loss in pressure, rapid wear of the blades and a disagreeable noise, making operation impractical under such conditions.

It is, therefore, one of the objects of the invention to overcome this cavitation in order to make such a pump suitable for such use, and this is accomplished by reducing the entrance velocity to a minimum and by other features hereinafter to be set forth.

Another difficulty is that in centrifugal pumps of this character it is necessary to maintain a sensible clearance between the pump runner and the casing in which it runs. The water, in passing through the runner develops considerable pressure, and leaks back and, where the delivery volume is small, this leakage, which is a dead loss, forms a large proportion of the delivery volume, and is, therefore, prohibitive.

Another of the objects of the invention is to overcome this difficulty.

More generally, my invention contemplates a simple, compact and efficient centrifugal pump, and one, which, among other things, is suitable for use as a locomotive boiler feed pump and is capable of being operated by a small single pressure stage turbine wheel of high speed, especially since a simple, compact and efficient driving turbine is necessitated in a locomotive.

How the foregoing, together with such

other objects and advantages as may hereinafter appear, or are incident to my invention, are obtained is illustrated in preferred form in the accompanying drawings, of which

Fig. 1 is a section through the pump and the turbine drive therefor;

Fig. 2 is an enlarged view showing the runner partly in elevation and partly in section;

Fig. 3 is a plan view of the intake side of the runner;

Fig. 4 is a face view of a packing forming a part of my invention and illustrates a detail thereof;

Figs. 5 and 6 are respectively a plan view from the under side and side view of the core utilized in forming the passages in the runner, Fig 6 being shown upside down from the operating position.

Referring now to Fig. 1, the reference letter A denotes the pump of my invention and the letter B, a simple, compact, single pressure stage, high speed turbine motor for driving the pump. The pump is a separate invention and this application is concerned with it alone. It will suffice to point out that the drive shaft 7 is threaded into the face of the runner casting opposite the intake side, the runner having a hub 8 for this purpose; and that the shaft is suitably supported by bearing means 9.

The pump in general comprises the casing having a water inlet 11 to the pump bowl 12, around which there is the discharge chamber 13 having an outlet 14 to the boiler; and the runner mechanism C.

The runner has an axial intake and a radial discharge from its periphery. It has a plurality of water passages *a, b, c, d, e, f* and *g* terminating at the entrance or intake side in a plane perpendicular to the axis of rotation. The intake ends of these passages are arranged on a small radius around the central boss or crown 15, the walls 16 between the passages at this point constituting cutting blades. The intake ends of the passages are thus segmental in section. The lower radial surfaces of the blades are the driving surfaces at the entrance, the circumferential surfaces 17 and 18 being the side walls of the passages at this point.

However, the radial surfaces of the passages at the entrance spiral, gradually twist

or change to lie in a plane at right angles to the axis of rotation and thus become the side walls of the passages toward and at the exit ends thereof. The circumferential surfaces which, at the entrance, are the side walls of the passages gradually become the driving surfaces as the exit is approached. This will be clear from inspection of Figs. 5 and 6 showing the form of core used. The passages gradually merge from a substantially segmental section at the intake ends thereof, into an approximately square or rather slightly oblong rectangular section and then gradually into a markedly oblong rectangular section at the exit. Hence, while the exit ends of the passages succeed one another at the periphery in the same plane, and while the intake mouths are also side by side at the center, the passages overlies one another at intermediate points, as shown in Fig. 2.

This arrangement makes it possible to locate the intake ends quite close to the center and hence the velocity of the blades is comparatively low which, together with the smooth transition of driving from the radial to the circumferential surfaces before described, causes the water to enter at relatively low velocity without the formation of eddies. Although the water is discharged at high velocity and high pressure, the low entrance velocity and the smoothness of the driving surfaces prevent cavitation, making it possible to operate the pump at the high speeds (preferably in the neighborhood of 8000 R. P. M.'s, far higher than common practice with centrifugal pumps) which will deliver the small volume at the requisite pressure for the uses for which the pump is primarily intended.

With reference to efficiency and the matter of leakage, hereinbefore mentioned, I provide an automatic, pressure operated packing for the runner which provides the necessary clearance for the runner but holds such clearance at a minimum so that the loss through leakage is kept within allowable limits. Two of the packings are employed and, as they are preferably identical, only the lower one as shown in Fig. 1 will be described. In its preferred form this packing comprises a ring 19 having inner flanges 20 and 21 extending from opposite faces of the ring. The lower flange 20 slidably fits in the sleeve 22 threaded into the bowl of the pump, and the ring can thus shift axially. Rotation of the ring is prevented by the pin 23.

The body of the ring lies in the pressure space of the bowl, and since the external diameter of the flange 21 is less than the external diameter of the bowl, the area of the upper face of the body of the ring is greater than the area of the lower face. The upper face of the ring has an interrupt-

ed groove 24 leaving several ribs 25 having preferably the same depth as the flange 21 which tend to prevent cocking of the ring. The outer flange 26 of the ring has less depth than the flange 21.

The operation is as follows. Assuming that the pump is at rest and the flange 21 is in contact with the runner, on starting up, the ring will be subject to unbalanced pressure which will move the ring away from the runner axially. As the ring moves a gap is opened between the flange 21 and the runner and the ratio of clearance as between the flange 21 and the runner to that between the flange 26 and the runner will increase, in consequence of which the pressure between the ring and the runner decreases. (The same result could be obtained by drilling holes in the flange 26 instead of making it of less depth.) Soon a point is reached where the force tending to push the ring away from the runner is balanced by the pressure on the back of the ring and the ring "stays put" at a definite clearance from the runner. By proper proportioning, this clearance can be accurately determined in advance and I prefer to so proportion the parts as to maintain a clearance of from .001 to .002 of an inch. With this clearance, the leakage is not sufficient to impair the efficiency of the pump in an amount which would render the pump unsuitable for the use intended, while at the same time rapid wear is avoided and the pump will remain in satisfactory operating condition for long periods. While I prefer to proportion flanges 20 and 21 so as to obtain a slight clearance between the ring 19 and the runner C as described above, I may so proportion these flanges as to permit the ring to touch the impeller with a light pressure between the surfaces in contact and I desire to claim this construction broadly.

Having the shaft extend only partly into the runner from the face opposite the intake side thereof, as well as the overlying arrangement of the passages, makes it possible to arrange the inlet ends of the passages on a small radius.

The walls between passages at the entrance, are slightly chamfered.

No specific claims are made herein to the governor mechanism, the arrangement whereby I am enabled to locate the impeller and the bearing close together; or the overhung impeller with a small throat to minimize leakage, as claims directed thereto have been made part of the subject matter of applications Serial Nos. 157,527, 160,539 and 166,041.

I claim:

1. In a centrifugal pump, a runner having a plurality of axial inlet and radial discharge passages of quadrilateral cross section, two walls of each passage being

formed through part of their length as helical portions and through the remainder as circular portions of spiral circumferential surfaces spaced radially from each other; the other two walls being formed from surfaces which at the inlet end are in planes spaced circumferentially from each other and are substantially radial, and which are warped therefrom helically into planes perpendicular to the axis and spaced axially from each other.

2. A centrifugal pump runner having a plurality of enclosed circumferentially spaced water passages of substantially quadrilateral cross section at the inlet and exit, each of said passages being so formed that two non-driving walls at the runner exit are continuations of two driving walls at the runner entrance, and two driving walls at the runner exit are continuations of two non-driving walls at the runner entrance.

3. A centrifugal pump runner having a plurality of enclosed circumferentially spaced axial inlet and radial discharge water passages of substantially quadrilateral cross section at the inlet and exit, each of said passages being so formed that its two non-driving walls at the runner exit are continuations of the two driving walls at the runner entrance, and its two driving walls at the runner exit are continuations of the two non-driving walls at the runner entrance.

4. An axial entrance centrifugal pump runner having a plurality of enclosed circumferentially spaced water passages of substantially quadrilateral section, two of the walls of said passages being surfaces which are substantially helicoidal and the other two walls of said passages being surfaces which are substantially spiral throughout the length of said passages, the helicoidal surfaces being driving surfaces at the runner entrance and non-driving flow guiding surfaces at the runner exit and the spiral surfaces being non-driving flow guiding surfaces at the runner entrance and driving surfaces at the runner exit.

5. An axial entrance centrifugal pump runner having a plurality of enclosed circumferentially spaced water passages of substantially quadrilateral cross section at their inlet and exit which are so formed that the two walls which at the runner entrance separate two adjacent passages gradually extend into two surfaces of revolution which constitute two boundary walls common to

all of said passages at the runner exit, and the walls which at the runner exit separate two adjacent passages gradually extend into two surfaces of revolution which form boundary walls common to all of said passages at the runner entrance.

6. In a centrifugal pump, a runner having a plurality of enclosed circumferentially spaced axial inlet passages of substantially quadrilateral cross section, two walls of each passage, in the main, being spaced radially and circumferentially and the other two walls, in the main, being spaced axially and circumferentially.

7. A centrifugal pump runner having a plurality of enclosed circumferentially spaced water passages of substantially quadrilateral cross section throughout their length, said passages being warped so that two walls thereof beginning at the runner entrance and for a portion of their length are driving walls while for the balance of their length up to the runner exit they are non-driving flow guiding walls, and that the other two walls beginning at the runner entrance and for a portion of their length are non-driving flow guiding walls while for the balance of their length up to the runner exit they are driving walls, the transition being gradual.

8. In an axial entrance, centrifugal pump runner, an enclosed water passage a wall of which is substantially radial at the runner entrance and transapical at the runner exit and a contiguous wall of which is circumferential at the runner entrance and axial at the runner exit, the change in the walls from radial and circumferential respectively to transapical and axial respectively being gradual.

9. A centrifugal pump runner having a plurality of enclosed water passages of substantially quadrilateral cross section throughout their lengths, said passages being defined by two walls which are passage separating and driving walls at the runner entrance and flow guiding walls at the runner exit, and by two walls which are flow guiding walls at the runner entrance and passage separating and driving walls at the runner exit, the change in the walls from passage separating and flow guiding walls respectively to flow guiding and passage separating walls respectively being gradual.

In testimony whereof I have hereunto signed my name.

CHRISTOPHER A. SCHELLENS.