Feb. 11, 1964 J. WALEFFE ETAL 3,120,920

POCKET COMBINATION FOR EXTENSION FOR SPEED AND LOAD RANGE OF AWM SUPERCHARGER

Filed Aug. 22, 1961 2 Sheets-Sheet 1

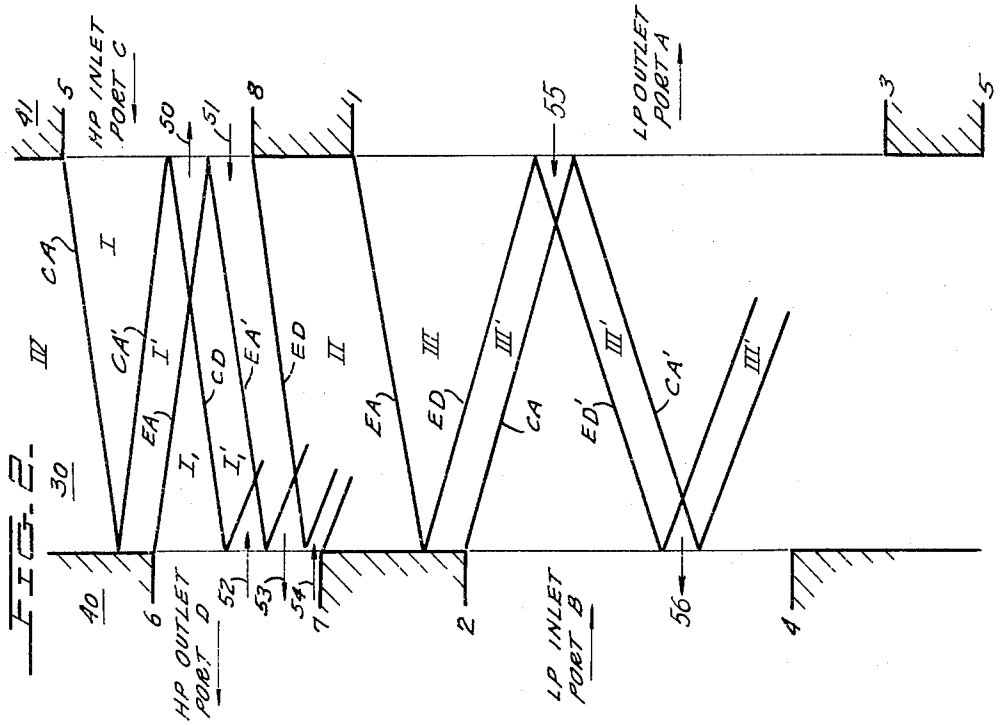



FIG. 2.

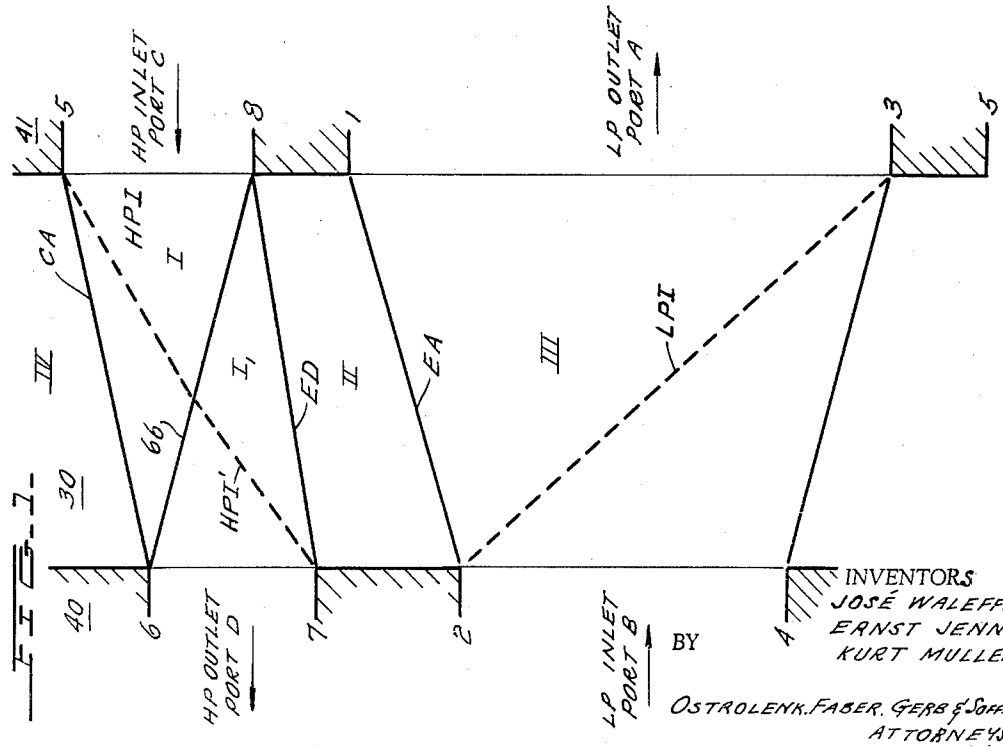



FIG. 1.

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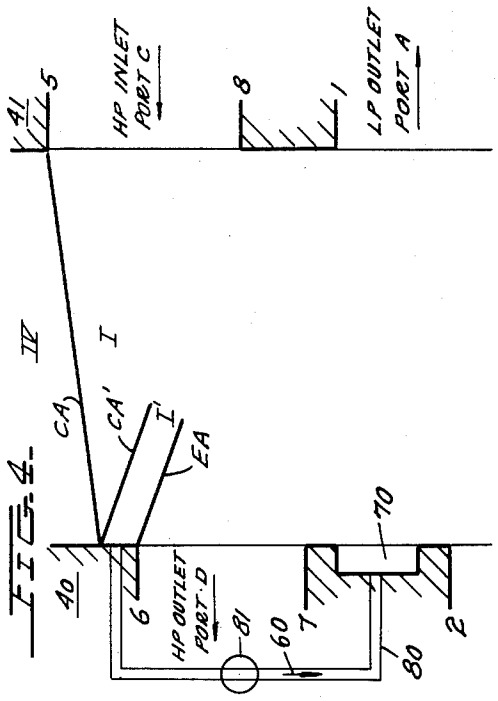



FIG. 4.

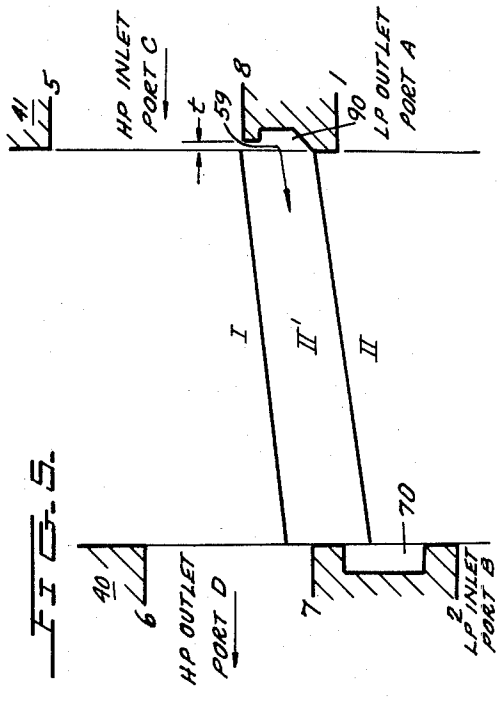



FIG. 5.

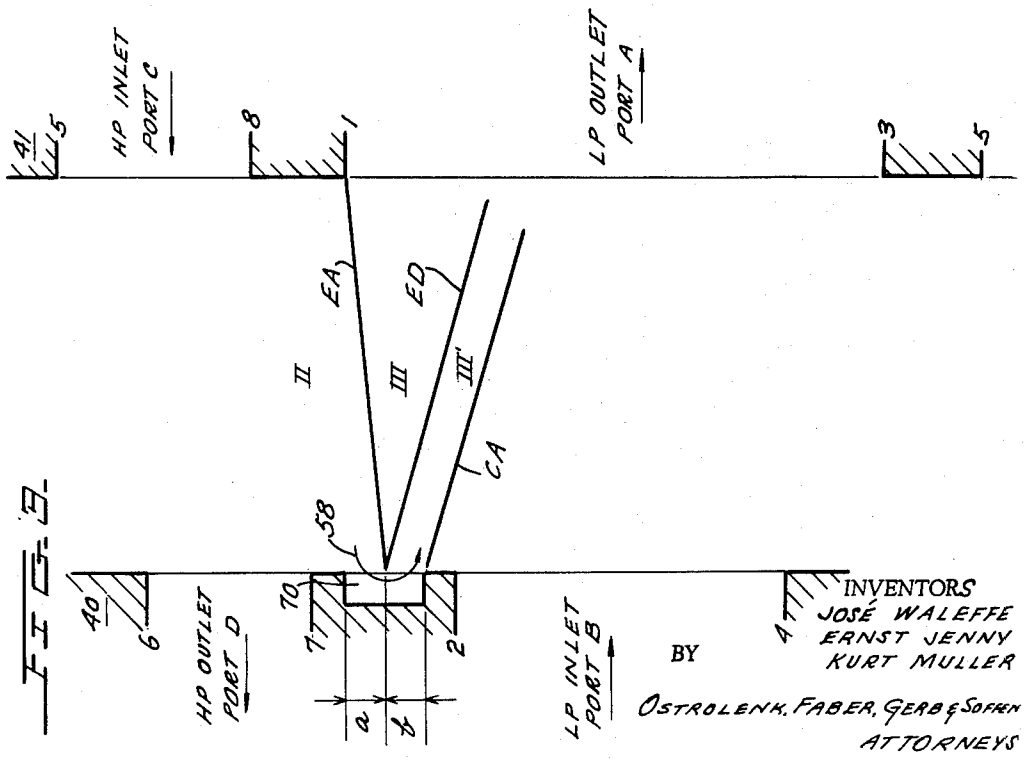



FIG. 3.

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United States Patent Office 3,120,920 Patented Feb. 11, 1964



3,120,920

# POCKET COMBINATION FOR EXTENSION FOR SPEED AND LOAD RANGE OF AWM SUPERCHARGER

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Filed Aug. 22, 1961, Ser. No. 133,104
Claims priority, application Switzerland Aug. 30, 1960
5 Claims. (Cl. 230—69)

Our invention relates to pressure exchangers and, more particularly, is directed to a novel construction for pressure exchangers which are directly driven by the unit being supercharged by the pressure exchanger.

Pressure exchangers are well known in the art and the general arrangement to which our invention is directed is illustrated in several prior art patents, such as: U.S. Patent 2,853,987, issued September 30, 1958, to M. Berchtold et al., entitled "Diesel Engine Supercharging by the Aero-Dynamic Wave Machine"; U.S. Patent 2,867,981, issued January 13, 1959, to M. Berchtold, entitled "Aero-Dynamic Wave Machine Functioning as a Compressor and Turbine"; U.S. Patent 2,957,304, issued October 25, 1960, to M. Berchtold, entitled "Aero-Dynamic Wave Machine Used as a Supercharger for Reciprocating Engines"; U.S. Patent 2,959,344, issued November 8, 1960, to E. Niederman, entitled "Reverse Cycle Aero-Dynamic Wave Machine"; and U.S. Patent 2,970,745, issued February 7, 1961, to M. Berchtold, entitled "Wave Machine." All of the aforementioned patents are assigned to the I-T-E Circuit Breaker Company.

In the construction of pressure exchangers, also known as aero-dynamic wave machines, it is not always possible to have the rotor of the machine rotating at the r.p.m. necessary for optimum results. That is, in the event it is desirable to use the pressure exchanger as a supercharger, and to have the pressure exchanger directly or belt-driven from the reciprocating engine being supercharged, then there will be a variation in the speed of the rotor.

Unfortunately, the mis-timing of the waves within the machine are detrimental to low pressure scavenging as the rotor speed is rdeuced. Furthermore, there will be an undesirable back-flow from the pick-up or high pressure outlet port as a result of the mis-timing of the waves created by the slowing down of the rotor.

It is a primary object of our invention to provide a novel construction for a pressure exchanger whereby a variable speed rotor will have minimum effect on the mass flow out of both the high pressure outlet port and the low pressure outlet port.

Another object of our invention is to provide a construction in which a pressure exchanger is provided with means located between the high pressure zone and the low pressure zone in the direction of rotation, which means will permit an elevation of pressure to thereby increase the mass blow in both the high pressure and low pressure outlet ports.

Still another object of our invention is to provide a construction whereby a pocket is located in the first or cold stator plate between the high pressure outlet port and the low pressure inlet port in the direction of rotation.

Another object of our invention is to provide a construction whereby a pocket is located in the first or cold stator plate between the high pressure outlet port and the low pressure inlet port in the direction of rotation and a second pocket located between the high pressure inlet port and low pressure outlet port in the direction of rotation.

Another object of our invention is to provide a construction whereby a pocket is located in the first or cold



stator plate between the high pressure outlet port and the low pressure inlet port in the direction of rotation whereby additional fluid and high pressure is introduced into the first pocket.

These and other objects of our invention will be apparent from the following description when taken in connection with the drawings in which:

FIGURE 1 is a cycle diagram showing the various conditions existing within the rotor of the pressure exchanger under optimum operating conditions, namely, when the rotor is rotated at design r.p.m.

FIGURE 2 is a cycle diagram of pressure exchanger similar to FIGURE 1 but illustrates the mis-timing of the waves within the rotor when the rotor is driven at a speed less than design speed.

FIGURE 3 is a partial cycle diagram illustrating a first pocket of our invention and the manner in which the pressure in the zone between the high and low pressure zones in the direction of rotation can be increased.

FIGURE 4 shows a cycle diagram of a pressure exchanger with the novel pocket arrangement of FIGURE 3 but shows a modification whereby additional pressure can be introduced into the rotor through the additional pocket.

FIGURE 5 is a cycle diagram similar to FIGURE 3 but illustrates the manner in which a novel second pocket can be utilized in a pressure exchanger to increase the efficiency thereof.

In many applications, where a pressure exchanger is used as a supercharger for a reciprocating engine, the reciprocating engine will operate over a large speed range, as for example, from 500 to 2,500 r.p.m. In order to avoid the necessity of providing a separate prime mover for the pressure exchanger and also for the purposes of mechanical simplicity, it is desirable to drive the pressure exchanger directly from the reciprocating engine by way of a fixed ratio drive means from the crankshaft of the reciprocating engine. Thus, in such situations, the pressure exchanger will operate over the same general speed range as the reciprocating engine. The use of a pressure exchanger supercharger and reciprocating engine is illustrated and described in the aforementioned U.S. Patents 2,853,987 and 2,957,304.

In FIGURE 1 I have shown the wave diagram for a pressure exchanger in which the rotor is rotated with respect to the stator plates at design speed and, therefore, operating under optimum conditions. A more detailed description of the optimum operating conditions of a pressure exchanger is illustrated in the aforementioned U.S. Patents 2,867,981, 2,957,304, 2,959,344 and 2,970,745.

For the proper operation of a pressure exchanger as a supercharger for a reciprocating engine, it is desirable that no contaminated hot gases enter the pick-up or high pressure outlet port D. That is, the high pressure interface HPI should not reach into the high pressure outlet port D and, furthermore, the low pressure interface LPI should have left the rotor 30 before or at least at the closing of the low pressure outlet port A. That is, the high pressure interface created at the leading edge 5 of the high pressure inlet port C should terminate at the trailing edge 7 of the high pressure outlet port D and the low pressure interface created at the leading edge 2 of the low pressure inlet port B should terminate at the trailing edge 3 of the low pressure outlet port A.

In general, the pressure in the field $I_1$ will be somewhat higher than in the field I. That is, the wave created at the leading edge 6 of the high pressure outlet port D will be a compression deceleration wave and identified by the numeral 66. Thus, if the wave 66 is a compression wave, the flow speed in the rotor will be reduced as shown by the change in the high pressure interface between HPI and HPI'. That is, as the high

pressure interface crosses the wave 66, the flow speed in the rotor will be reduced. The compression wave 66 will result in an increase of the pressure so that the field $I_1$ will be greater than the field I which will also reduce the flow speed still further and, hence, will result in a reduction of the mass flow out of the high pressure outlet port D. On the other hand, if the wave 66 is an expansion acceleration wave, the flow speed will be increased and, hence, the mass flow through the high pressure outlet port D will be increased.

The magnitude of the mass flow moving through the high pressure outlet port D is a function of the demand of the reciprocating engine being supercharged by the pressure exchanger. Hence, the pressure in the field $I_1$ depends on the flow demands of the reciprocating engine. The higher the demands of the reciprocating engine, the lower will be the pressure of the field $I_1$ and the lower the demand of the reciprocating engine, the higher will be the pressure of the field $I_1$.

The portion of the rotor located between the high pressure inlet port C and the high pressure outlet port D indicated by the field I and $I_1$, is generally referred to as the high pressure zone of the rotor 30. The low pressure zone III is in that portion of the rotor located between the low pressure inlet port B and the low pressure outlet port A. The higher the flow speed in the field III, the earlier the interface LPI will reach the trailing edge 3 of the low pressure outlet port A.

It is essential to provide complete scavenging of the machine through the low pressure outlet port A. That is, the low pressure interface must reach the right-hand end of the rotor 30 before the rotor 30 is closed by the trailing edge 3 or, ideally, exactly when the channels within the rotor 30 are closed by the trailing edge 3. If this sequence does not exist between the pressure exchanger, the contaminated gases exist in the field III will be trapped in the rotor and may subsequently be discharged through the high pressure outlet port D. Although this effect is substantially minimized by the reverse cycle illustrated in FIGURE 1, it is nevertheless a condition to be avoided when possible.

In order to avoid incomplete low pressure scavenging, a predetermined minimum amount of flow speed in the field III is required. The flow speed in the low pressure zone III, depends on the pressure existing in the field II in the sense that the higher the pressure in the field II, the higher will be the flow speed in the field III. However, the pressure in the field II depends on the pressure in the field $I_1$. Thus, the higher the pressure in the field $I_1$, the higher will be the pressure in field III, and, therefore, the greater will be the flow speed to achieve complete scavenging through the exhaust or low pressure outlet port A.

It, therefore, becomes clear that the level of pressure in the field $I_1$ determines the degree of low pressure scavenging through the low pressure outlet port A since the higher the pressure in the field $I_1$, the earlier the low pressure interface LPI will reach the low pressure outlet port A. To put this another way, the higher the flow speed demands of the reciprocating engine being supplied with compressed air through the high pressure outlet port D, the lower will be the pressure in the field $I_1$, thereby subsequently resulting in a reduction of the flow speed in the field III and possible incomplete low pressure scavenging.

In the illustration of FIGURE 1, there is shown basically the ideal cycle in which there is complete high pressure scavenging between the leading edge 6 and the trailing edge 7 of the high pressure outlet port D and complete low pressure scavenging between the leading edge 1 and the trailing edge 3 of the low pressure outlet port A.

In FIGURE 2, we have shown a wave or a cycle diagram for a pressure exchanger in which the rotor 30 is rotated at a speed below the design speed of FIGURE 1. In this case, there are two effects which simultaneously tend to jeopardize the low pressure scavenging to such an extent that contaminated hot gases may remain trapped in the rotor and subsequently leave through the high pressure outlet port D. This effect is extremely undesirable since these contaminated hot gases would then be fed by the pressure exchanger directly to the reciprocating engine being supercharged thereby and could result in the stalling of the reciprocating engine.

As seen in FIGURE 2, a compression acceleration wave CA is created when the rotor 30 is opened by the leading edge 5 of the high pressure inlet port C. This wave CA arrives too early at the hot stator plate 40, that is, before the high pressure outlet port D has been opened by its leading edge 6. The early arrival of the wave CA can be seen by comparison of FIGURES 1 and 2. Since the wave CA impinges upon the stator plate 40, it will be reflected in the closed channel ends as wave CA'. The pressure behind the wave CA' in the field I' is considerably higher than the pressure in the field I. In fact, the pressure in the field I' is higher than the pressure in the high pressure inlet port C and, hence, when the wave CA' arrives at the high pressure inlet port C, there will be an out-flow of gas into the port C as shown by the arrow 50. The wave CA' will be reflected at the port C and travel to the left through the rotor as wave CD.

When the high pressure outlet port D is opened by the leading edge 6, an expansion acceleration wave EA will be created in the rotor so that the pressure in the field $I_1$ will be reduced from the pressure existing in the field I' to a magnitude approximating that pressure in the high pressure outlet port D. The pressure in the field $I_1$ could be higher or lower than the pressure in the field I, the difference being relatively small. The pressure in the field I', on the other hand, is significantly higher of the order of twice as high or more than the pressure in the field I. Therefore, the effect of mis-timing of the wave CA by virtue of too low a rotor speed is the existence of a high pressure band identified by the field I' travelling upstream in the rotor.

Upon the arrival of the wave EA at the high pressure inlet port C, the wave will be reflected as wave EA' thereby re-establishing in-flow conditions from the high pressure inlet port C into the rotor as indicated by the arrow 51. The field $I_1'$, which is bounded by the waves CD and EA', will upon reaching the high pressure outlet port D, cause an undesirable flow reversal. That is, there will be an in-flow from the high pressure outlet port D into the rotor as indicated by the arrow 52. This undesirable in-flow into the channels of the rotor corresponds to the undesirable out-flow into the high pressure inlet port C indicated by the arrow 50. However, the arrival of the wave EA' re-establishes the out-flow conditions into the high pressure outlet port D as indicated by the arrow 53.

Upon the closing of the high pressure inlet port C by the trailing edge 8 an expansion deceleration wave ED is created. Behind the wave ED, in the field II, the pressure will be substantially below the pressure existing in both the high pressure inlet port C and the high pressure outlet port D. Since the rotor is operating below design speed, the wave ED will arrive too early at the stator plate 40 as can be seen by a comparison of FIGURES 2 and 1. Hence, an additional in-flow will result at the high pressure outlet port D as illustrated by the arrow 54.

Thus, in summary, the mass flow at both the high pressure inlet port C and the high pressure outlet port D are reduced as a result of the flow reversal fields existing in the corresponding ports. It must be noted that there are two flow reversal fields 52 and 54 acting at the high pressure outlet port D and only one flow reversal field 50 acting at the high pressure inlet port C. Therefore, the

tendency is for the flow at the high pressure outlet port D to be reduced more than the flow in the high pressure inlet port C. However, when the pressure exchanger is used as a supercharger for a reciprocating engine, both the input flow at the high pressure inlet port C and the outgoing flow at the high pressure outlet port D must be the same. However, this can only occur if the flow of speeds at the high pressure outlet port D is increased; that is, if the pressure in the field $I_1$ is increased. However, as the speed of the rotor is reduced, the pressure in the field $I_1$ is decreased which is detrimental to low pressure scavenging.

When the low pressure outlet port A is opened by the leading edge 1, as shown in FIGURE 2, an expansion acceleration wave EA is generated and will arrive at the left end of the rotor (stator 40) before the low pressure inlet port B is opened by the leading edge 2. This is best seen by a comparison of FIGURES 1 and 2. However, the wave EA drops the pressure in the field III to a value close to ambient pressure and the flow speed is still directed toward the low pressure outlet port A. However, upon the reflection of the wave EA as wave ED on the closed end of the rotor, a pressure will be generated which is considerably below the pressure existing in the low pressure outlet port A illustrated by the field III'.

When the low pressure inlet port B is opened by the leading edge 2, a compression acceleration wave CA will be generated which will essentially re-establish the conditions in field III. When the low pressure field III' bounded by the waves ED and CA arrives at the low pressure outlet port A, a flow reversal will occur thereby resulting in a back flow into the rotor indicated by the arrow 55. As the band III' travels back and forth through the rotor, it generates a back flow each time it arrives at an open port. This is illustrated by the arrow 56 in the low pressure inlet port B. This undesirable flow reversal hinders the scavenging of the hot gases out through the low pressure outlet port A and, therefore, is extremely detrimental to low pressure scavenging.

Accordingly, there are two effects which are detrimental to low pressure scavenging as the rotor speed is reduced, namely, a reduction of the pressure in the high pressure zone $I_1$ due to more flow reversals in the high pressure outlet port D than in the high pressure inlet port C, and also back flow in the low pressure ports A and B due to the mis-timing of the wave EA.

As the rotor speed is reduced, the proper operation of the pressure exchanger completely breaks down and, in effect, there is no supercharging for the reciprocating engine. At this point, it is necessary to have the pressure exchanger by-passed by means of a by-pass valve and a butterfly valve which is governed by the pressure differences between the ports C and D. A detailed description of this construction is set forth in the aforementioned U.S. Patent 2,853,985. However, the by-pass arrangement has many drawbacks in addition to the mechanical complications and additional expenses that are involved. At very low speeds, the butterfly valve will always be closed. If full fuel flow is applied to the reciprocating engine under these conditions, the reciprocating engine will not receive enough air and may smoke severely. It is, therefore, extremely desirable to have a construction wherein the pressure exchanger can run at very low speeds, as characterized by a possible negative difference between the ports D and C, and will maintain proper low pressure scavenging. In this case, it will be possible to supercharge the reciprocating engine at low speeds and avoid the aforementioned smoke difficulties.

Our present invention provides an arrangement whereby the low speed difficulties are overcome without resorting to the aforementioned by-pass construction.

Our invention includes the two constructions, the first of which may solve the problem up to a point for some installations and may be a complete solution for other installations. The second feature is used in conjunction with the first feature and together these features provide a complete solution to the problem.

The first feature of our invention is provided to mitigate the harmful effects of the mis-timing of the wave EA shown in FIGURE 2 and consists of the recess or pocket 70 placed in the first or cold stator plate 40 in the area between the trailing edge 7 of port D and the leading edge 2 of port B as seen in FIGURE 3.

As previously mentioned, the harmful effects of mis-timing results from the low pressure existing in the field III' which causes flow reversal. It is also recalled that the pressure in the field II is higher than the pressure in the field III and III'. A recess or pocket 70 arranged in the manner shown in the FIGURE 3 allows certain quantities of fluid to pass from the field II into the field III'. As a consequence, the pressure in the field III' will be higher than without the pocket 70. In fact, the pressure in the field III' may possibly even be higher than the pressure in the field III. That is, the flow reversals can be substantially reduced or even completely eliminated and low pressure scavenging, thereby materially improved, or in fact, complete low pressure scavenging can be achieved. Thus, the recess or pocket 70 in effect provides for the fluid flow from the field II to the field III' as indicated by the arrow 58 to raise the pressure in the field III' and thereby reduce or eliminate back flow of conditions in the low pressure outlet port A.

It is noted that the average pressure in the pocket 70 must be high enough in order for the pressure in the field III' not to be below a dangerous value. In fact, it can be shown experimentally and analytically that if the pressure in the pocket 70 falls below a certain minimum, low pressure scavenging ceases to be complete.

However, the pressure in the pocket 70 depends on two conditions; the first is that the pressure in the field II must be high enough. However, as previously noted, when the rotor speed is decreased, the pressure in the field II falls. Secondly, the pressure in the pocket 70 depends on the timing of the wave EA, the slower the rotor turns, the earlier the wave EA arrives at the cold stator 40 and the shorter the distance "*a*" and the longer the distance "*b*." A short distance for the length "*a*" means that there will be a small volume flow into the pocket 70 from the field II. A long distance "*b*" means a large volume flow from the pocket 70 into the field III'. Hence, the shorter the length "*a*" and the longer the length "*b*," the lower is the pressure in the pocket 70. To re-phrase this, the slower the rotor turns, the lower will be the pressure in the pocket 70. Thus, both of these conditions result in a decrease of the pressure in the pocket 70 with a decrease in the rotor speed and there exists a rotor speed in which the pocket 70 by itself is inadequate as a low pressure scavenging aid. However, this situation can be improved if the pressure in the pocket 70 is artifically increased by supplying fluid from an outside duct 80 as illustrated in FIGURE 4. The duct 80 supplies high pressure fluid in the direction of the arrow 60 to the pocket 70 and the supply of this high pressure fluid can be from any outside source. That is, the pressure in the high pressure fluid for the duct 80 can be supplied from the exhaust of the reciprocating engine being supercharged by the pressure exchanger or it is possible to feed the duct 80 with air or gas tapped off somewhere in the pressure exchanger itself where high pressure fluid is available.

One example is shown in FIGURE 4 wherein the field I' is used as a supply of high pressure fluid. As previously mentioned, at low rotor speeds the wave CA created at the leading edge 5 is completely reflected as wave CA' at the closed channel end thereby resulting in a very high pressure field I'. It would be possible to divert some of this high pressure air through the duct 80 into the pocket 70 to maintain the required minimum pocket pressure as illustrated in FIGURE 4. In this case, it would be necessary to provide a check valve 81 in the duct 80 in order to avoid back flow of fluid from the pocket 70 into the field

I' at the operating points where the pressure of the pocket 70 is high.

Still another possibility is to provide some hot gas from the exhaust manifold of the reciprocating engine into the pocket 70. This has the advantage that sufficient pressure is always available in the exhaust manifold of the reciprocating engine. However, it is not always desirable to introduce contaminated exhaust gases at this point of the cycle and, furthermore, it is possible that the duct 80 may be clogged by carbon particles existing in the exhaust gases.

However, the use of the pressure from the exhaust manifold of the reciprocating engine can be made practical with an arrangement illustrated in FIGURE 5 wherein a second pocket 90 is located in the cold stator plate 41. The pocket 90 is located between the trailing edge 8 of high pressure inlet port C and leading edge 1 of port A. A portion of the pocket 90 extends up to the leading edge 8 thereby forming a gap having a width *t* between the cold stator plate 41 and the rotor 30. Thus, at the trailing edge 8, the high pressure inlet port C remains partially open and fluid is allowed to flow through the gap *t* and then into the rotor as indicated by the arrow 59. This results in an elevator pressure band in the rotor as shown by the area II'. The pressure in the area II' is in between the pressure in the field I and II. The pressure in field II' is governed by the geometry of the pocket 90 and can be chosen such that the pressure in the first pocket 70 will always be high enough to guarantee complete low pressure scavenging. The geometry of the pocket shown in FIGURE 5 is one possibility that it has been tried and has proved to be successful. However, it could, for instance, be possible to slightly enlarge the high pressure inlet port C or to employ another geometric configuration for the pocket 90 in the gap.

Thus, the novel arrangement of providing the second pocket 90 insures that there will be sufficient pressure in the field II' to permit the flow of fluid through the first pocket 70 to the field III' as indicated by the arrow 58 of FIGURE 3.

In the foregoing, we have described our invention only in connection with preferred embodiments thereof. Many variations and modifications of the principles of our invention within the scope of the descritpion herein are obvious. Accordingly, we prefer to be bound not by the specific disclosure herein but only by the appending claims.

We claim:

1. A pressure exchanger comprising a rotor for receiving and discharging fluids, said fluids being supplied to said rotor from a high pressure inlet port and a low pressure inlet port; fluid being extracted from said rotor from a high pressure outlet port and a low pressure outlet port; said high pressure outlet port and said low pressure inlet port being located in a first stator plate on one side of said rotor; said high pressure inlet port and said low pressure outlet port being located in a second stator plate on the other side of said rotor; said rotor being rotatable with respect to said first and second stator plates; said high pressure inlet and outlet ports defining a high pressure zone of said rotor; siad low pressure inlet and outlet ports defining a low pressure zone of said rotor; means located between said high pressure and low pressure zones in the direction of rotation of said rotor in increase the pressure in said rotor between said zones, said means including a first means in said first stator plate and a second means in said second stator plate.

2. A pressure exchanger comprising a rotor for receiving and discharging fluids, said fluids being supplied to said rotor from a high pressure inlet port and a low pressure inlet port; fluid being extracted from said rotor from a high pressure outlet port and a low pressure outlet port; said high pressure outlet port and said low pressure inlet port being located in a first stator plate on one side of said rotor; said high pressure inlet port and said low pressure outlet port being located in a second stator plate on the other side of said rotor; said rotor being rotatable with respect to said first and second stator plates; said high pressure inlet and outlet ports defining a high pressure zone of said rotor; said low pressure inlet and outlet ports defining a low pressure zone of said rotor; means located between said high pressure and low pressure zone in the direction of rotation of said rotor to increase the pressure in said rotor between said zones; said means being comprised of a first means located in said first stator plate and a second means located in said second stator plate; said second means comprising a gap in said second stator plate extending from said high pressure inlet port to an area in said rotor between said high pressure and low pressure zones in the direction of rotation of said rotor.

3. A pressure exchanger comprising a rotor for receiving and discharging fluids, said fluids being supplied to said rotor from a high pressure inlet port and a low pressure inlet port; fluid being extracted from said rotor from a high pressure outlet port and a low pressure outlet port; said high pressure outlet port and said low pressure inlet port being located in a first stator plate on one side of said rotor; said high pressure inlet port and said low pressure outlet port being located in a second stator plate on the other side of said rotor; said rotor being rotatable with respect to said first and second stator plates; said high pressure inlet and outlet ports defining a high pressure zone of said rotor; said low pressure inlet and outlet ports defining a low pressure zone of said rotor; means located between said high pressure and low pressure zone in the direction of rotation of said rotor to increase the pressure in said rotor between said zones; said means being comprised of a first means located in said first stator plate and a second means located in said second stator plate; said second means comprising a gap in said second stator plate extending from said high pressure inlet port to an area in said rotor between said high pressure and low pressure zones in the direction of rotation of said rotor, said first and second means also including a pocket located in said first and second stator plates respectively.

4. A pressure exchanger comprised of a first and second stator plate and a rotor; said first and second stator plate being located on opposite sides of said rotor; said first stator plate having a high pressure outlet port and a lower pressure inlet port; said second stator plate having a high pressure inlet port and a low pressure outlet port; said each of said ports having a leading edge and a trailing edge; said rotor being rotatable past said ports; said ports being circumferentially spaced in the following sequence; leading edge of said high pressure inlet port, leading edge of said high pressure outlet port, leading edge of said low pressure outlet port, leading edge of said low pressure inlet port, trailing edge of said low pressure inlet port, trailing edge of said low pressure outlet port, a first means located in said first stator plate between said trailing edge of said high pressure outlet port and said leading edge of said low pressure inlet port to aid scavenging of fluid through said low pressure outlet port, a second means located in said second stator plate between said high pressure inlet port and said leading edge of said low pressure outlet port; said second means comprising a passageway from said high pressure inlet port to an area in said rotor opposite said first means.

5. A pressure exchanger comprising a rotor for receiving and discharging fluids, said fluids being supplied to said rotor from a high pressure inlet port and a low pressure inlet port; fluid being extracted from said rotor from a high pressure outlet port and a low pressure outlet port; said high pressure outlet port and said low pressure inlet port being located in a first stator plate on one side of said rotor; said high pressure inlet port and said low pressure outlet port being located in a second stator plate on the other side of said rotor; each of said ports having a leading edge and a trailing edge; said rotor being rotatable with respect to said ports; said ports being

circumferentially spaced with respect to said rotor in the following sequence: said leading edge of said high pressure inlet port; said leading edge of said high pressure outlet port; said trailing edge of said high pressure inlet port; said trailing edge of said high pressure outlet port, said leading edge of said low pressure outlet port, said leading edge of said low pressure inlet port; said trailing edge of said low pressure inlet port; said trailing edge of said low pressure outlet port; said high pressure inlet and outlet ports defining a high pressure zone of said rotor; said low pressure inlet and outlet ports defining a low pressure zone of said rotor; means located between said high pressure and low pressure zone to increase the



pressure in said rotor between said zones; said means being a first pocket located in said first stator plate between said high pressure outlet port and said low pressure inlet port and a second pocket located in said second stator plate; said first and second pockets providing a passage for high pressure fluid to pass into an area adjacent said low pressure zone.

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