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Rotary Compressor With The Stationary Crankshaft

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ABSTRACT

Novel rotary compressor provides an unitary assembly, in which a rotor of the driver -external rotor electric motor, is radially integrated with a pump and arranged on a stationary crankshaft which is fixedly connected to the hermetic housing and supports as a rigidly fixed to it motor stator so spinning rotor block and an eccentrically fit revolving piston with integrally formed or rigidly fixed vane. The rotor block and eccentrically fit revolving piston are assembled without clearance in between and have line rolling contact where tangential (linear) velocities of both are unidirectional and equal in magnitude. Such coupling of two cylindrical bodies having rolling contact is used to transfer motion from the rotor block (driver) to the revolving piston (follower). Such arrangement is not only supplement basic moment transferred to the revolving piston by rigidly fixed vane (coupler) but also reduce and eliminate frictional and leakage losses. Novel rotary compressor further comprises the following: a suction gas delivery system which eliminates external accumulator, provides cooling of the motor and delivered refrigerant into suction chamber under higher pressure (supercharged) due to action of the build-in impeller; a discharge system utilizing a tubular thin-walled discharge valve and circular expansion cavity equipped with nozzles through which the discharge gas ejects tangentially from side of the revolving piston in direction opposite to the direction of rotation to produce reaction force and momentum which will also supplement moment transferred to the revolving piston by rigidly fixed vane; a lubricating oil system employing positive displacement oil pump, oil channels bored in the crankshaft and plurality of oil accumulated circular pockets which prevents formation of "gas lock" condition and accelerates delivery of oil to bearings and mating surfaces, said oil which is thrown outward by centrifugal force will not only lubricate, but will also form linear or annular shaped liquid seal at the edges of abutting parts and, consequently, reduce leakage losses.

1. INTRODUCTION

A motor and a pump of existing rotary compressors typically are installed axially with a predetermined distance there between. This distance combined with height of the motor and height of the pump defines as an axial length of the compressor so a transmission loss of the rotating power generating by the motor and distributed by long, revolving crankshaft to the pump. Furthermore, the cantilevered position of a rotor on the shaft contributes to deflection of the crankshaft and make bearings load relatively high. In addition, the parts of the contemporary rotary compressors are supported as by crankshaft (rotor, roller, etc.), so by the housing (stator, cylinder block, suction accumulator, etc.). Such dual supporting structure complicates assembly of a compressor due to the necessity of precision axial and radial positioning of the pump parts, housing, rotor and stator.

Developed novel rotary compressor described in this paper provides a unitary assembly in which a rotor of the driver -external rotor electric motor, is integrated with concentrically situated pump and rotatably arranged on the stationary crankshaft, (Dreiman , 2013). A hermetic housing is rigidly connected to the stationary crankshaft which supports as the fixed to it motor stator, so the spinning rotor block and a revolving piston. The single structure supporting the motor stator, the rotor block, the revolving piston and the housing simplifies compressor assembly, and allows precision, reliable setting of the air gap, concentricity and eccentricity due to the common single datum - axial line of the crankshaft.

The revolving piston is disposed eccentrically outside of the rotor block with the direct (no operating clearance) contact between the inner peripheral surface of the revolving piston and external peripheral surface of the rotor lock.

The direct contact line of the revolving piston and the rotor block lie in the plane passing through the lines of centers of rotation which are fixed. In such kinematic pair (an external and internal cylinder) a motion of the rotor block (driver) can be transmitted to the revolving piston (follower), and both cylindrical parts rotate in the same direction due to a force developed at the contact line. In such direct-contact mechanism rolling contact exists only if there is no sliding which can be eliminating if normal to the line of contact radial force P_{12N} will create coupling friction F_f between the cylinders (see Figure 1). The force P applied to the cylinder 3 is expressed as $P = M_3 / R3$ where M_3 is momentum applied to the cylinder 3 with radius R_3**.** For reliable transmission of the torque without sliding, F_f has to be in some degree larger than P**,** so $F_f = \beta P$, where $\beta = 1.25 – 1.50$. With small error the normal radial force P_{12N} can be expressed by using power value of the driver

$$P_{12N} = F_{f\,/}\ f = \beta\,(P / f) = \beta\,(102\ N_2 / f\,v) = \beta\,(102\ N_2\ 60_{*}\ 100 / f\,\pi\,D_3\,n_3), \qquad (1)$$

where f – coefficient of friction, D_3 and n_3 **–** diameter and RPM of cylinder 3, N_2 power of driver, kW

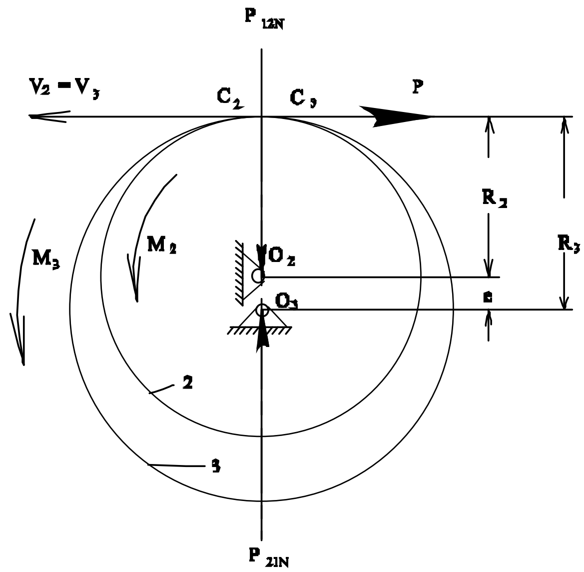

Figure 1: Schematic view of the rotor block (cylinder 2) and revolving piston (cylinder 3) coupling

The center distance

$$e = D_3 / 2 – D_2 / 2 = D_2\,(i - 1)/ 2 = D_3\,(i - 1) / 2, \quad \text{so} \quad D_3 = 2e / (i - 1), \qquad (2)$$

where $i = \acute{\omega}_2 / \acute{\omega}_3 = D_3 / D_2$ is a transmission number. Substituting this in Eq. (1) gives

$$P_{12N} = 97400\,(\beta / f)\,(N_2 / n_3)\,(i - 1) / (i\,e) \qquad (3)$$

In Figure 1 cylinders 2 and 3 are in direct contact. Points $\mathbf{C_2}$ and $\mathbf{C_3}$ are the coincident points at contact point which became the instant center where the tangential velocities $\mathbf{V_2}$ and $\mathbf{V_3}$ of, consequently, the rotor block(cylinder 2) and the revolving piston (cylinder 3) are unidirectional and equal. It means that there is no sliding friction and frictional losses at the contact line of both parts are minimal due to the relatively low value of the rolling friction. In addition, the angular momentum developed through such contact will be supplemental to the momentum transferred from the rotor block to the revolving piston through the coupler –rigidly fixed vane.

2. DESCRIPTION OF THE COMPRESSOR DESIGN

Referring to Figure 2, there is shown a vertically oriented novel rotary compressor comprising generally housing (1) defined by cylindrical main body portion (2) coaxial with the stationary crankshaft (3). The upper cap (4) and the lower cap (not shown) are defined the hermetically sealed high side (5) part of the housing. The hermetically sealed input suction cavity (6) is an integral part of the housing. Located in the main body portion (2) of the housing is mounting on stationary crankshaft compressor pump (7) which is formed by radial integration of an external rotor motor and pump parts. The pump (7) construction comprises the revolving piston (8), the rotor block (9), a disk shaped thrust block (10), the electric motor compartment (11) housing the stator (12), and an oil pump assembly (not shown).

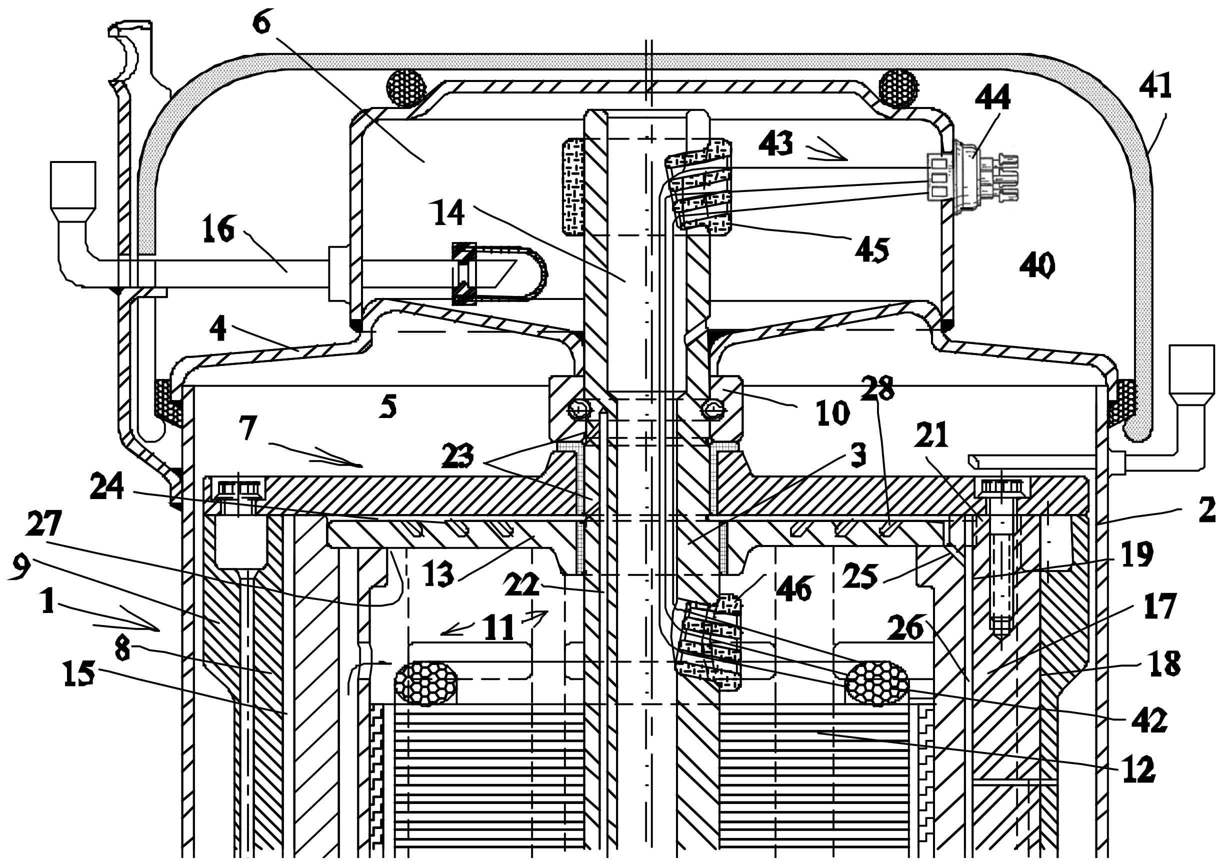

Figure 2: A side sectional view of a revolving piston compressor with the stationary crankshaft, (Dreiman, ,2013)

2.1 Suction System

The suction system of the novel rotary compressor consists of an input suction cavity (6) which is disposed inside of the compressor housing and is an integral part of it, an electric motor compartment (11) equipped with impeller (not shown) which has been rigidly fixed to the revolving rotor block head (13) below or above the stator (8) and a variable volume suction chamber (52), (Dreiman, 2013). The electric motor compartment (11) is in fluid communication as with the input suction cavity (6) through a channel (14) inside of the crankshaft, so with the suction chamber through plurality of the vertical channels (15) formed in the wall of rotor block (9), and the suction port (50),as shown in Figure 3. A suction inlet pipe

(16) directs a vapor-liquid mixture of refrigerant and lubricating oil through a screen (to filter an impurities) into the inner volume of the input suction cavity (6), where gas flows to the top and the liquid collects above the upper end cap (4) separating high and low side of the housing. The heat generated by high side discharge gas will be transferred trough the upper end cap (4) to the liquid collected at the bottom of the input suction cavity and will significantly accelerate a vaporization process. The refrigerant drawn from the suction input cavity (6) will be delivered to the part of the electric motor compartment spaced below the stator where centrifugal force triggered by rotation of the rotor block (9) will forcibly guide oil to the mitering (bleeding) hole formed in the side wall of the rotor block (9). The vapor portion of the refrigerant, pressure of which has been increased by the impeller, will be supercharged through the suction port to the suction chamber. Such arrangement of the suction system prevents direct delivery of gas-liquid-oil mixture to the suction chamber, eliminates external accumulator, increases the liquid refrigerant storage capacity and provides cooling of the compressor motor during and after its duty cycle.

2.2 Revolving Piston - Vane Assembly

The vane (17) of the novel rotary compressor separates suction chamber and compression chamber, as shown in Figure 3, and has an axial edge (18) rigidly fixed without operating clearance to the inner periphery of the revolving piston (8) with opposite axial edge (19) fitted in between sliding segmental bushings (20) mounted in the rotor block, (Dreiman, 2004). The vane, however, not only serves to separate the working cavity between rotor block and revolving piston into suction chamber and compression chamber, but it also forms a mechanical connection (coupler), between the driver (rotor block) and follower (revolving piston) so, that the external rotor motor revolves simultaneously the rotor block (9) and the revolving piston (8). Such coupling arrangement also prevents slippage in between cylindrical surfaces at the line of contact.

The vane radial edges (21) are axially fixed between the revolving piston heads, as shown in Figure2, without operating clearances. It eliminates frictional losses and also blocks high-low side leaks. Integration of the vane with the revolving piston excludes frictional and leakage losses observed in contemporary compressors where a vane and roller slide against facing surfaces of the stationary cylinder block and stationary heads. There is also no" grinding" interference between vane and roller (roller and spring are eliminated) and, in addition, nose of the vane (axial edge 19) does not required any special treatment, such as precision machining, TiN coating, special vane tip seal, etc., recommended and used in prior art compressors.

The length of the exposed part of the vane separating compression and suction chambers changes with variation of the turning angle θ during compression cycle and will be, (Dreiman and Bunch 2005a),

$$L(\theta) = e\,(1 - \cos\theta + \lambda^2/2\,\sin^2\theta)\,, \qquad (4)$$

The compression chamber volume trapped within rotor block - revolving piston at an arbitrary turning angle θ and corrected for the volume of the exposed part of the vane will be

$$V_v(\theta) = V(\theta) - t\,H\,L(\theta) = V(\theta) - e\,t\,H(1 - \cos\theta + \lambda^2/2\ \sin 2\theta),$$

where (5)

$$V(\theta) = \lambda H R_2^2 / 2\,[\,(1+\lambda)\,\theta - 2\sin\theta - \lambda^2/2\,\sin 2\theta\,],$$

Here V (θ) – volume of the working cavity, $\lambda = e / R_3$, t and H, thickness and height of the vane, R_2 and R_3 – radius of a rotor block and a revolving piston, e- eccentricity

2.4 Lubrication

Novel rotary compressor is provided with a lubricating oil flow path through which oil accumulated in an oil sump is directed to the compressor components. Located in the lubrication flow path is positive displacement, reciprocating piston type oil pump, (Dreiman N.I., and Bunch R.L., 2005b) which is mounted on the stationary crankshaft below the compressor pump. The oil flow tests of the pump with variety of inlet port diameters and positions, and related data for sliding vane rotary, reciprocating and scroll compressors are shown in Figure 4.As you can see, the pumping ability of the prospective oil pump is superior in comparison with rotary, reciprocating and scroll compressor pumps which, as usual, are mounted in the rotating crankshaft. The oil pump discharge channel is in fluid communication with axially extending channel (22) through which oil delivered to plurality of oil passages (23) to lubricate the compressor bearings (see Figure 2). The lubricant is also supplied to the annular chamber(24) and through a passage (25) thus to a portion of axial slot (26) in which the vane (17) is disposed.

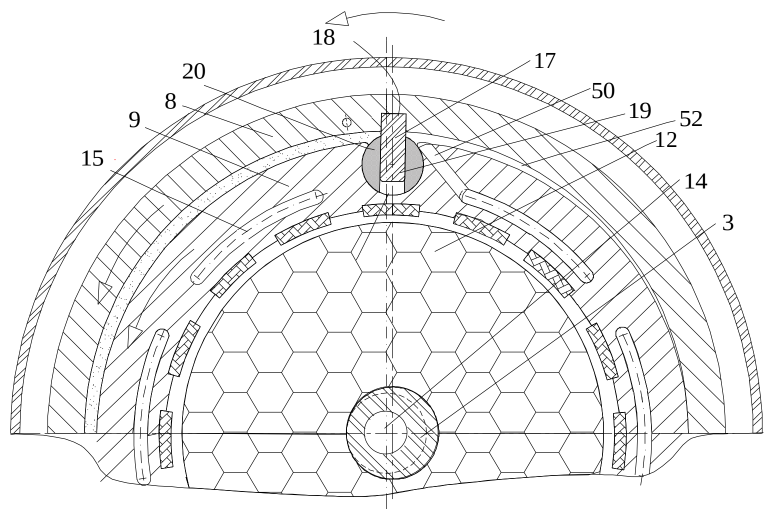

Figure 3: A horizontal cross-sectional view of the revolving piston rotary compressor, (Dreiman, 2013)

During operation of the compressor the lubricant delivered into the chambers (24) and axial slot (26) is thrown outward by centrifugal force to form annular seal of liquid at the periphery of chamber and along the axial edges (19) of the vane. Any leakage that may take place past the sides of the vane or past the end (26) of the rotor block will be leakage of the lubricant inward into the crescent-shaped space, as this lubricant is under the delivery pressure (discharge + pumping) in addition to the pressure due to centrifugal force. By this means therefore leakage of the fluid to be compressed past these relatively moving members will be prevented and at the same time a bath of lubricant will be provided in between mating parts.

The lubricating oil tends to drain away from bearings and mating surfaces upon shutdown of the compressor. Upon startup of the compressor, there may be some delay before oil can be resupplied to the bearings. In order to prevent oil delivery delay, compressor is provided with pockets (28) in which oil accumulates at the compressor stop and will be thrown by centrifugal force toward periphery at start, (Dreiman 2013).

2.4 Discharge System

Referring now to Figure 5, the discharge system of new compressor comprises a discharge valve assembly, a circumferential gas expansion cavity (31) equipped with plurality of nozzles (32) through which discharge gas ejected in form of high velocity jets in direction opposite to the direction of rotation, (Dreiman, 2013).

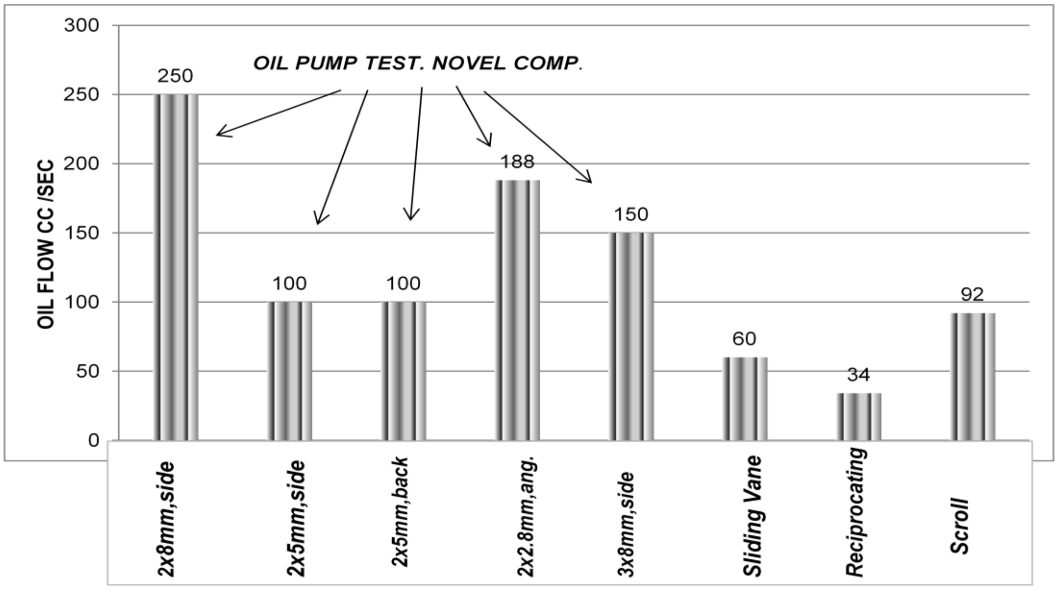

Figure 4: The oil flow test results of the pump with variety of inlet port diameters and positions

A reed valve used in a contemporary compressor is clamped at one end (cantilever type) with another end moving unsupported toward valve stop at the opening of the valve. Such design promotes localization of the stress at the reed valve part spaced close to the clamped end and increases probability of failure.

The design used in novel rotary compressor overcomes the aforementioned problem by providing a discharge valve assembly with an increased vapor flow, minimal gas re-expansion volume and reduced residual stress. The discharge valve comprises a cylindrically shaped valve member (33) made from thin- walled spring steel tubing which has been clamped sidewise to the wall (34) of an elliptically shaped discharge chamber (35) formed at the radial end face of the revolving piston. An advantage of the discharge valve system of the novel compressor is that the tubular thin walled valve member has its port seating surface immediately exposed to fluid pressure generated within the compression chamber on opening. The curved shape of the exposed valve member surface has larger area than any exposed to the discharge surface of the same port diameter prior art flat discharge valve member. It accelerates valve opening thereby increasing the performance of the compressor while decreasing possible throttling effects.

High velocity discharge gas jets ejected from the plurality of circumferentially positioned nozzles (32) creates a reaction forces acting in the direction of rotation and angular momentum due to the trust developed by such forces will, definitely, positively affect rotation of the revolving piston and reduce a load of the motor. An approximate value of the reaction force N_n exerted on the revolving piston by a discharge gas ejected from the single nozzle can be calculated by using the following general equation:

$$N_n = \rho_d A_n V_d^{\,2} + A_n (p_d - p_h), \tag{6}$$

where ρ_d, v_d, and p_d are, correspondingly, density, velocity of flow and pressure of the discharge gas ejected from the nozzle which has cross-sectional area A_n, p_h - pressure of the media surrounding the revolving piston inside of the housing. At this juncture, it should be noted that reaction force causing rotation of the revolving piston assembly is a function of pressure of gas supplied to the expansion chamber, the size and capacity of the nozzle coupled to the expansion chamber, the larger the supply pressure and/or larger the nozzle size, then greater the reaction force created, and, hence then higher the rotational speed of revolving piston assembly.

Another advantage of the discharge valve system is that use of cylindrical valve retainer (36), spaced within the tubular valve, to clamp valve sidewise to the wall of the elliptical chamber provides that no special valve alignment is necessary at the time of compressor assembly. During assembly, after the tubular valve is slid inside the elliptical discharge chamber (35) and the valve retainer is in place, the tubular valve clamped by the retainer to the wall of the elliptical discharge chamber will be automatically align with the valve port.

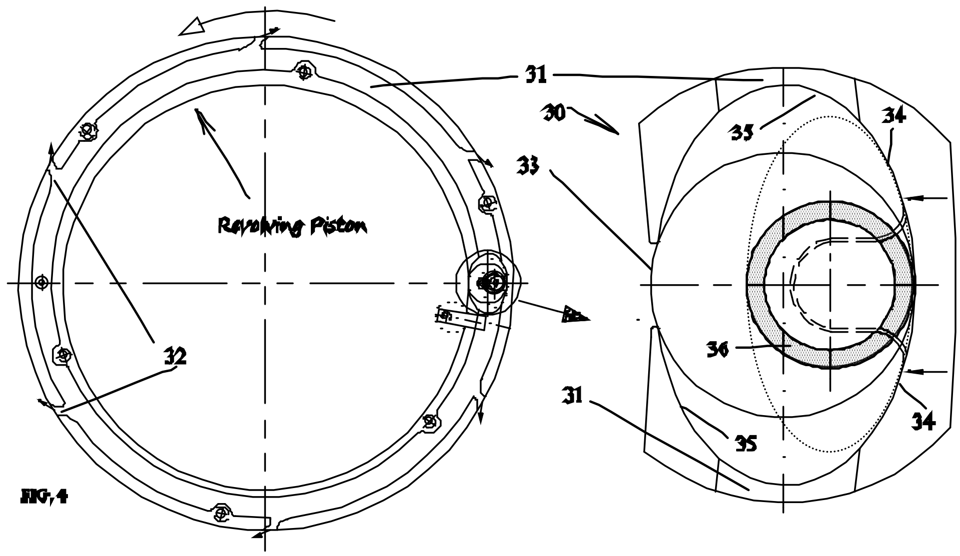

Figure 5: Top view of the revolving piston with the discharge nozzles and enlarged view of the valve, (Dreiman , 2013)

Still another advantage of the discharge valve system of the present invention is that the tubular valve is supported during compression by parts of the elliptical discharge chamber wall (34) adjacent symmetrically on both sides of the clamping line of the tubular valve and valve back support area will increase as the load triggered by the discharge pressure rise. It prevents concentration of the stress in the valve member and improves reliability.

2.5 Miscellaneous Modifications

In conventional rotary compressors an external electrical circuit basically includes an electrical terminal carrying the circuit though the housing, start and run capacitors, a solid state relay, a thermally operated overload protector, etc. An electrical terminal box, secured, as usual, externally to the top cap or to the outer circumference of the compressor high side housing, is used to accommodate some or all of the items specified above. Furthermore, another inverter storage box is provided on an outer circumference of the housing for inverter controlled compressors. In addition, the power supply and control wires located inside of a prior art compressors housing are in proximity to the mowing parts and are subjected to intensive discharge pressure pulsations and an elevated temperature of a discharge gas-oil mixture passing at high velocity (70 to 135 ft/sec) through the motor stator –rotor gap to an discharge outlet.

The design of the novel rotary compressor made it possible to locate specified above items of the external electrical circuit in the storage space (40), as shown in Figure 2, positioned in the limits of the compressor housing and adjusted to the cool wall of the suction cavity (6), (Dreiman, 2013).

The hollow part of the crankshaft (4), denoted as the suction channel (14), is used also as a conduit for power supply wires and wires controlling operation of the electric motor, so for delivery of suction gas. The power to the stator windings (42) is supplied by wires 43 running from inner part of hermetic terminals (44) through an upper insulator (45), part of the suction channel (14), and an insulator (46) located above the stator.

An advantage of such modifications is that, after elimination of plurality of external electrical boxes, the compressor is compact (smaller package space), has better configuration, lower manufacturing cost and is more reliable. In addition, the power supply circuit elements are protected from effect of high ambient temperatures, moisture, are safer, and elimination of bulky boxes open access to the housing surface areas for painting, thereby avoiding potential oxidation and rust. Furthermore, the close proximity of the cylindrical storage space (40) to the cool wall of the suction cavity (6) makes it convenient for compressors utilizing inverters to arrange cooling of power semiconductor modules.

3. LEAKAGE AND FRICTIONAL LOSSES

Analytical and experimental studies of sliding vane rotary compressors indicate that leakages and frictional losses occur at the radial and axial operating clearances between roller and facing surfaces of stationary cylinder block and stationary cylinder heads, and between the latest and a sliding vane, (Costa, C., at al., 1990), (Wu J., 2000), (Yang Jun, 2004), see Table 1.

Table 1: Relative Compressor Losses

LINE OR PLANE OF CONTACT	FRICTION LOSSES, %.		LEAKAGE FLOW FRACTION LOSSES *)	
	Conventional Rotary, [Costa, 1990]	Novel Rotary (expected)	Conventional Rotary, [Wu, J., 2000]	Novel Rotary (expected)
Roller O.D- Vane tip	24.3	*Eliminated*	0.0025	*Eliminated*
Roller O.D-Cylinder I.D.	1.0	0.5	0.0309	0.0
Roller I.D-Eccentric	18.7	*18.7*	---	---
Roller Radial Ends-Cylinder Heads	2.7	*2.5*	0.0294	0.0294
Vane Radial Ends - Cylinder Heads	N/A	*Eliminated*	0.0005	*Eliminated*
Vane Sides-Slot Sides	28.1	*Eliminated*	0.0310	*Eliminated*
Crankshaft- Cyl. Head Bearings	25.2	26.5	---	---
Vane Sides- Bushing (Swing comp.)	-	25.5	---	0.03
Bushing –Pocket Wall (Swing comp.)	-	5.0	---	---
Sub-Total	-	-	0.0943	0.0913
Total Loss, %	100	78.7	100	63.0

**) Leakage flow fraction losses relative to delivered suction volume.*

The dominant source of a frictional loss is rubbing of the vane nose against the end wall of the roller. The design of swing type rotary compressors excluded such interference by integrating roller and vane. But radial surfaces of the vane and end wall of the roller still mowing against facing surfaces of the stationary cylinder heads. Sinji Tanaka (2004) studied friction and lubrication characteristics between a vane and a vane guiding bushing used in a swing type compressor and compared test results with that of sliding vane rotary compressor. The results indicate that the p_c V (pressure-velocity) value of swing compressor is about one tenth of that of sliding vane compressor. Teh Y.L., and Ooi K.T., (2006), performed frictional analysis of a revolving vane compressor, (OOI K.T 2012). Their theoretical study has shown a higher mechanical efficiency of the revolving vane design as compared to that of sliding vane rotary compressor.

The values of frictional and leakage losses obtained by analytical and experimental study of sliding vane compressor and indicates expected relative values of the losses for a revolving piston rotary compressor are summarized in Table 1. The losses of the new compressor have been reduced or eliminated at the following locations:

- **Vane tip-roller O.D**. Roller and associated losses are eliminated. The tip of the vane is free from interference.
- **Vane edges - facing surfaces of the piston heads**. The vane is rigidly fixed without operating clearances to the heads. There are no associated leakage and frictional losses
- **Vane sides- slot sides in the cylinder block**. Leakage to the suction side through the clearances of the leading semi –cylindrical parts of the guide bushing is minimal due to the compression of guide bushings by the vane flat surface against the bearing seat as a result of a driving force pressure exerted by the rotor block. Frictional losses are also relatively small , (Sinji Tanaka, 2004).
- **Roller O.D.-Cylinder I.D**. Absence of the axial operating clearance between the revolving piston and the rotor block will completely eliminate related leakage losses, and minimize frictional losses.
- **The frictional losses between the radial end surfaces of the rotor block and facing surfaces of the revolving piston heads** will be also minimal due to the low relative rubbing speed between synchronously revolving in one direction components.

Preliminary general evaluation shows (see Table 1) that expected mechanical losses of the developed compressor can be up to 20% lower than the losses of a conventional compressor and expected reduction of the leakage losses is around 35%.

4. CONCLUSIONS.

A novel revolving piston rotary compressor with the stationary crankshaft is characterized by the following;

- The vane is rigidly secured without any operating clearances as to the inner periphery of the revolving piston so to the facing surfaces of the piston heads. The design also excludes rubbing contact between a vane and a piston. Such arrangement completely eliminates leakage and frictional losses associated with a sliding vane of contemporary compressors.
- There is no axial operating clearance between the rotor block and the eccentrically fit revolving piston. Such coupling of two cylindrical bodies will not only transfer moment from the rotor block (driver) to the revolving piston(follower) but also reduces frictional losses and eliminates leakage losses.
- A suction side delivery system has eliminated external accumulator and provides cooling of the motor. The suction gas is distributed (supercharged) in the suction chamber under higher pressure due to the action of an impeller.
- A discharge system utilizing a tubular thin-walled discharge valve and a circular expansion cavity equipped with nozzles through which the discharge gas ejects tangentially from side of the revolving piston to produce reaction force and momentum which will also supplement moment transferred to the revolving piston through the rigidly fixed vane.
- The single structure (the stationary crankshaft) supporting the motor stator, the rotor block, the revolving piston and the housing simplifies compressor assembly and allows precision, reliable setting of the air gap, concentricity and eccentricity due to the common single datum -axial line of the crankshaft.

NOMENCLATURE

A	cross-sectional area	(m^2)
D	diameter	(m)
F_f	coupling force	(N)
e	eccentricity	(m)
f	friction coefficient	(-)
H	height of vane	(m)
i	transmission ratio	(-)
L	exposed length of vane	(m)
M	moment	(Nm)
N	reaction force	(N)
n	speed of rotation	(RPM)
P	force	(N)
P_{12N}	force normal to contact line	(N)
p	fluid pressure	(Pa)
R	radius	(m)
t	thickness of vane	(m)
V	volume of working cavity	(m^3)
V_v	corrected volume of working cavity	(m^3)
v	velocity	(m/s)
β	slippage prevention coefficient	(-)
θ	turning angle	(rad)
ρ	fluid density	(kg/m^3)
$\acute{\omega}$	angular velocity	(rad/s)

Subscripts

2	rotor block (driver)
3	revolving piston (follower)
d	discharge
h	housing
n	nozzle
v	vane

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