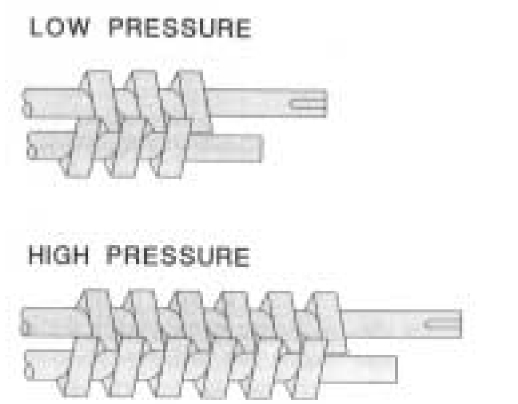
Hydraulic balance in multiple screw pipeline pumps

Multiple screw pumps for crude oil pipeline service are either of the two screw design or three screw design. At typical crude oil pipeline pressures of 500 to nearly 2,000 PSI (35 to nearly 140 BAR), pumping element loading due to hydrostatic pressure can be quite high. Without hydraulic balance to counteract this loading in one or two planes, bearing loads would be excessive, operating life short and the efficiency benefits of screw pumps lost to maintenance budgets.

In multiple screw pumps, each wrap of screw thread effectively forms a stage of pressure capability. High pressure pumps have 5 to 12 stages or wraps, whereas low pressure pumps may have only 2 or 3 wraps. The presure capability of a two screw pump, for example, is illustrated in Figure 1.

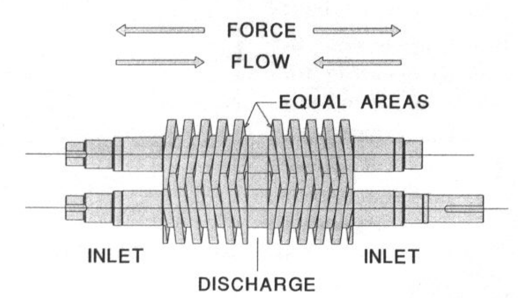
Figure 1 : Effect of staging on pressure capability



Two screw pumps

The two screw type pump normally uses timing gears outside of the pumped liquid to synchronize the mesh of the non-contacting screws. The most common arrangement is opposed helices (double suction) with the flow pattern being from the ends of the screw sets to the center of the pump, Figure 2. Within each wrap of thread, the stage pressure acts on all exposed areas. Due to blanking of some areas by the intermeshed threads, a radially unbalanced area exists within each stage. Stage pressure acting on these unbalanced areas cause radial forces on each screw set approximately

Figure 2 : Axial hydraulic balance of a two screw pump



90 degrees from the plane of the shafts.

In the axial direction, discharge pressure acts on the exposed center areas of each screw set. Because the diameters and lead angles of the opposed screw sets are equal, hydrostatic forces acting to the left are equal to those acting to the right. The net axial force due to discharge pressure is zero and the screw shafts are in tension.

Two screw pump hydraulic forces: Hydraulic radial forces on a two screw pump rotor due to differential pressure are illustrated in Figure 3. The forces are uniform along the length of the pumping threads. These hydraulic forces cause a deflection “d", for which running clearance must be provided in the surrounding pump body. Greater deflection requires larger clearances resulting in more slip flow or volumetric inefficiency, so “d" must be kept to a minimum. Excessive deflection will cause damage to the surrounding body and/or contribute to rotating bend fatigue, which will ultimately result in shaft breakage. The following is the general form of the deflection

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Figure 3 : Radial deflection of two screw pump rotor

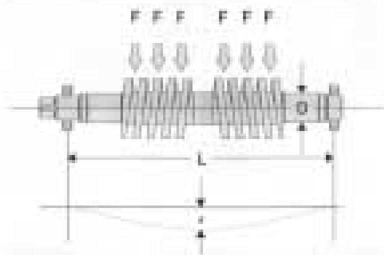
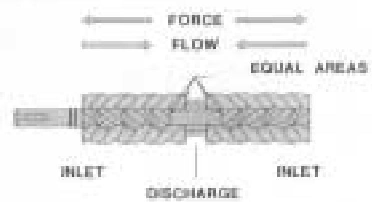


Figure 4 : Axial hydraulic balance of a three screw pump



equation:

$$d = \sum F \times L^3 \, I \, c / \times E \times I$$

where $\sum F$ is the summation of the hydraulic forces, L is the bearing span, c is a constant, E is the shaft material modulus of elasticity and I is the shaft moment of inertia. The shaft moment of inertia is a function of its diameter, D^4. This equation is simplified and, in practice, must account for the varying shaft and screw diameters as they change along the length of the rotor. If screw "shells" are not integral with the shaft, that is, not made from a single piece of material, then material differences as well as attachment schemes must be factored into the deflection calculation. In any event, it is easy to see that the bearing span, L, must be kept to a minimum to minimize deflection. Also of great importance, is the use of large diameter shafts and screw root sections to maintain minimum deflection.

Note that twin screw pumping screws will have either one or two thread starts. The single thread start sets displace more volume per revolution but form fewer stages within a given body length. Double thread start screw sets displace less volume than single starts but form more stages in the same body length and are thus suitable for higher pressures. Double thread start rotors are also inherently in dynamic balance due to their radial mass symmetry. Single thread start rotor sets must normally be dynamically balanced using dynamic balance machines and removing excess mass by drilling radial holes in the outside diameters of the screw threads.

Depending on the direction in which the threads are machined (left or right hand) and the direction of shaft rotation, the pump manufacturer can cause the deflection to be in either of the two radial directions, up or down for a horizontal pump. These radial deflection loads are absorbed through externally lubricated anti-friction bearings. Radial loads are proportional to differential pressure across the pump. Higher differential pressure produces higher radial loads or forces. Smaller lead angles of the screw set reduce these radial loads as well as reducing the flow rate. Larger lead angles increase the flow rate as well as the radial loading. Bearings are usually sized to provide 25,000 or more hours L1O bearing life at maximum allowable radial loading and maximum design operating speed. Because of this pumpage-independent bearing system, two screw pumps with external timing gears and bearings can handle high gas content as well as light oil flushes, water, etc.

Three screw pumps

Three screw pumps are produced in two basically different designs, single ended and double ended. The double end (double suction) design, Figure 4, is balanced in the axial direction in exactly the same manner as a two screw pump. The areas and lead angles are equal and opposite so that the axial hydrostatic forces on the shafts are opposed resulting in zero axial force due to discharge pressure. Virtually all three screw pipeline pumps use two thread starts on each of the rotors which provides inherent dynamic balance.

Single ended pumps use two 51-milar but different techniques to accomplish axial hydraulic balance. The center screw, called a power rotor, incorporates a balancing piston at the discharge end of the screw thread, Figure 5. This piston can be an integral part of the shaft material (one piece) as illustrated, a shrink fitted piece or a replaceable, hard coated piston. The area of the piston is made aboutl5% greater than the cross sectional area of power rotor thread. Thread inter-meshing exposes more than 360 degrees of the power rotor thread to discharge pressure.

The greater balance piston area compensates for the extra exposed thread area. Consequently, equal opposing forces produce zero net axial force due to discharge pressure and place the power rotor in tension. The balance piston rotates within a close clearance stationary bushing which may also be hardened or hard coated to resist erosive wear due to sand content in crude oil. The drive shaft side of the piston is normally internally or externally ported to the pump inlet chamber. Balance leakage flow across this running clearance flushes the pump mechanical seal, which remains at nominal pump inlet pressure.

Three screw pump idler rotors: The two outer screws, called idler rotors, also have their discharge ends exposed to discharge pressure. Through various arrangements, discharge pressure is introduced into a hydrostatic pocket area at the inlet end of the idler rotors, Figure 6. The effective area is just slightly less than the exposed discharge end area, resulting in approximately equal opposing axial forces on the idler rotors. The idler rotors are therefore in compre~~ion with a dcllbcrate imbalance to maintain their runnin~ position toward the inlet end of the pump. Should any force cause the idler rotor to move toward discharge, a resulting loss of pressure acting on the cup shoulder area or hydrostatic land area tends to restore the idler rotor to its design running position. The upper view in Figure 6 shows a stationary thrust block (cross hatched) and a stationary, radially self locating balance cup. Discharge pressure is brought into the cup via internal passages within the pump or rotor itself. The lower view shows a hydrostatic pocket machined into the end face of the idler rotor. It, too, is fed with discharge pressure. Figure 7 illustrates the pressure gradient across the inlet faces of the idler rotors. The gap shown is exaggerated and is actually very near zero. Diameter K_2 is approximately 50% of diameter K Discharge pressure introduced into the hydrostatic pocket acts on the full area of K~. This pressure then breaks down in a nonlinear manner as balance flow escapes into the inlet chamber across the land diameter, K_1-K_2. On average, half the discharge pressure acts on the land area. The developed force opposing that of discharge pressure, P, acting on the idler discharge area is:

$$P\pi (K_2)^2/4 + P/2(\pi/4(K_1)^2-(K_2)^2)$$

The force calculated above is slightly less then the force developed by discharge pressure acting on the exposed discharge area of the discharge end of the idler rotor. This exposed area is less than the full idler rotor diameter area due to the power rotor thread blocking some of the idler rotor thread from exposure.

For crude oil services, the hydrostatic end faces of the idler rotors are normally gas nitride hardened or manufactured from solid tungsten carbide and shrink fitted to the inlet end of the idler rotors. When the cup design is used, the cup inside diameter and shoulder area are normally gas nitride hardened. Both techniques are used to resist wear due to the normal contaminants found in crude oils.

Three screw pump hydraulic balance: In a radial direction, three screw pumps achieve power rotor hydraulic balance due to symmetry. Equal pressure acting in all directions within a stage or wrap results in no radial hydraulic forces since there are no unbalanced areas. The power rotor will frequently have a ball bearing to

Figure 5 : Three screw pump drive rotor balance

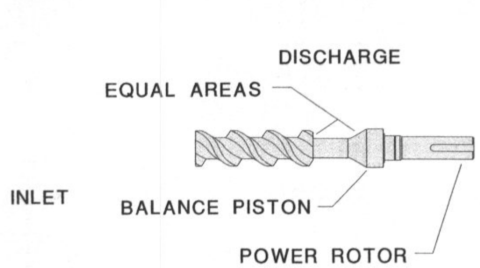


Figure 6 : Three screw idler rotor balance arrangements

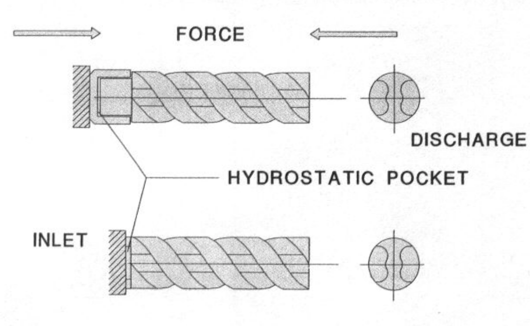


Figure 7 : Hydrostatic pressure gradient for idler rotor

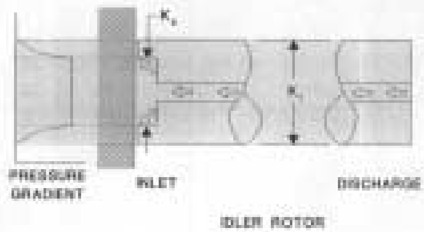
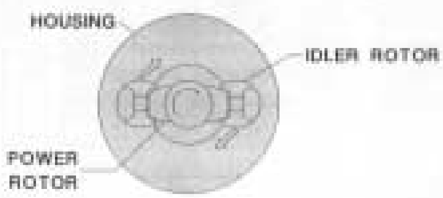


Figure 8 : Radial idler rotor loading



limit end float for proper mechanical seal operation but it is otherwise under negligible load. Idler rotor radial balance is accomplished through the generation of a hydro-dynamic liquid film, in the same fashion as a journal or sleeve bearing, Figure 8. The eccentricity of the rotating idler rotors sweeps liquid into a converging clearance resulting in a pressurized liquid film. The film pressure acts on the idler rotor outside diameters in a direction opposing the hydraulically generated radial load (see diagonally opposing arrows indicating direction of loading). Increasing viscosity causes more fluid to be dragged into the pressurized film causing the film thickness and thus pressure supporting capability to increase. The idler rotors are supported in their respective housing bores on liquid films and have no other bearing support system. Within limits, if differential pressure increases, the idler rotor moves radially towards the surrounding housing bores. The resulting increase in eccentricity increases the film pressure and maintains radial balance of the idler rotors.

Inlet pressures in three screw pumps:

In three screw pumps, inlet pressure above or below atmospheric will produce an axial hydraulic force on the drive shaft. In most pump applications, pump inlet pressures are below or slightly above atmospheric pressure so the forces generated by this low pressure acting on a small area are negligible. However, if the application requires the pump to operate at an elevated inlet pressure, from a booster pump for example, then inlet pressure acting on the inlet end of the power rotor is only partly balanced by this same pressure acting on the shaft side of the balance piston. In effect, the area of the power rotor at the shaft seal diameter is an unbalanced area. This area multiplied by the inlet pressure is the resulting axial load, towards the shaft end. When specifying pumps, be sure to clearly state the maximum expected inlet pressure so the pump manufacturer can accommodate this loading.

Several reliable methods are in use including use of the anti-friction bearing as an axial load carrying bearing, double extending the power rotor out the inlet end of the pump (adds a second shaft seal) or sizing the balance piston to counterbalance this axial force.

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