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Variable Speed Tri – Rotor Screw Compression Technology

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ABSTRACT

It is the intent of this paper to compare two compression approaches using helical screw technology in large flooded evaporator water cooled chillers. The paper will compare the basic compression losses of a twin screw compressor to that of a tri-rotor helical screw set with equivalent displacement. Losses will be evaluated at both full load and part load operation. It will be shown that for large water cooled chillers the tri rotor concept offers improvement in both full and part load performance.

1. INTRODUCTION

The objective of this paper is to identify the technical differences between twin and tri – rotor screw compression technology with a focus on performance loss. Twin screw technology is well known in the HVAC industry but application of helical screws to a tri - rotor concept is new. Tri – rotor technology has successfully been applied in the hydraulic industry for pumps. This paper will first discuss the operation of the tri – rotor and then cover the basic loss mechanisms for each technology.

A detailed design effort was undertaken to design twin and tri – rotor screw compressors with equivalent displacement and tip velocity. The details of these designs can not be presented due to the proprietary nature of the information but the generalized results are within the scope of this paper.

It should be noted that the tri – rotor compressor is a variable speed machine operating with reduced oil in circulation. It is the first commercial application of helical screws in an oil reduced compressor for the HVAC industry. The twin screw is a conventional oil flooded slide valve compressor. It will be shown that the reduction in total deflection of the tri – rotor permits oil reduced operation with two orders of magnitude less oil than an oil flooded twin screw. This significantly reduces drag losses providing substantial improvement in part load efficiency.

2. BACKGROUND

A tri – rotor screw compressor is shown in figure 1. This particular design is a variable speed compressor that operates from 50 to 500 tons. Suction and discharge ports are identified. Unloading of the compressor is achieved by simply reducing the speed of the compressor.

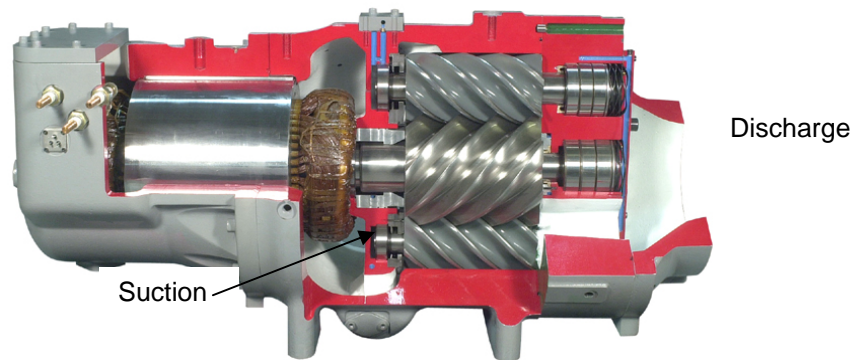


Figure 1: 500 Ton Tri – Rotor Screw compressor

The compression process for the tri – rotor is accomplished with a set of rotors shown in figure 2. This particular set has a lobe combination of 8 flutes on the male and 6 flutes on both females. An equivalent size twin screw set is shown in figure 3. For this paper we will be comparing the tri - rotor to a 5/6 twin set shown below.

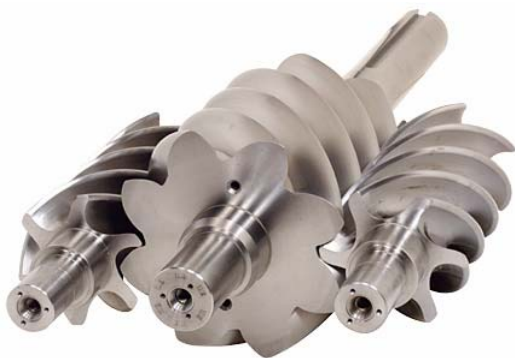


Figure 2: 8/6 tri – rotor set



Figure 3: 5/6 twin rotor set

Compression in the tri – rotor set is achieved with two simultaneous compression chambers on each side of the male rotor as shown in figure 2.

This paper will focus on the four major losses in screw compression of most interest to the designers; leakage losses, flow losses through the discharge port, drag losses of the rotors and part load operation losses.

The process for comparing the above losses will be to use the fundamental equations describing each loss and develop a ratio of tri – rotor to twin. It is not the intent of this paper to do an in-depth technical review of the losses. Once comparative ratios are established this paper will use a generalized twin screw loss weighting process to determine the overall performance ratio of the two technologies.

3. LEAKAGE LOSSES

Leakage losses for comparison purposes will be limited to the blow hole, seal line and tip clearance. Both twin and tri – rotor machines will be given the same rotor tip clearance and the required additional clearance for deflection.

Deflection calculations of the twin and tri - rotors is the first step to calculating the leakage losses. For purposes of comparison we will calculate the deflection based on a simply supported cylinder with an equivalent loading (W) shown in figure 4 and a diameter equivalent to the root diameter of the rotor.

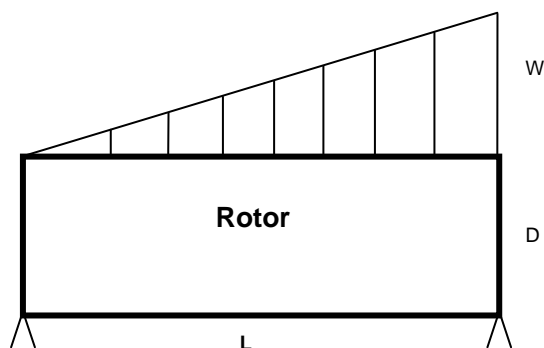


Figure 4: Relative deflections are calculated on the basis of a simply supported beam.

The deflection is calculated using equation (1) and the ratio of deflection is shown by equation (2).

$$Y = CWL^3 / EI \quad (1)$$

$$Y_{tr} / Y_{tw} = L_{tr}^4 D_{tr}^4 / D_{tr}^4 L_{tw}^4 \quad (2)$$

Where:

- a) Y = max deflection
- b) L = rotor length
- c) D = root diameter
- d) Def = total deflection of male and female rotors
- e) W = unit loading

Assumptions for the rotor geometry are as follows:

- f) Equivalent displacement
- g) Equivalent rotor tip speed
- h) 8/6 lobe combination – tri
- i) 5/6 lobe combination for twin

The above equations can be shown to produce the following deflection ratio for the female rotors.

$$Y_{tr} = .81Y_{tw} \quad (3)$$

It is also interesting to note that the tri - rotor male has no deflection due to the balanced gas loading. Therefore, the calculations for leakage of the tri rotor are primarily dependent on the female deflection while the twin total deflection is dependent on both the male and female rotors. Accurate deflection of the female rotor is shown in figure 5. Calculations are based on an accurate finite element model. This deflection will be the base for calculating the other rotor deflections.

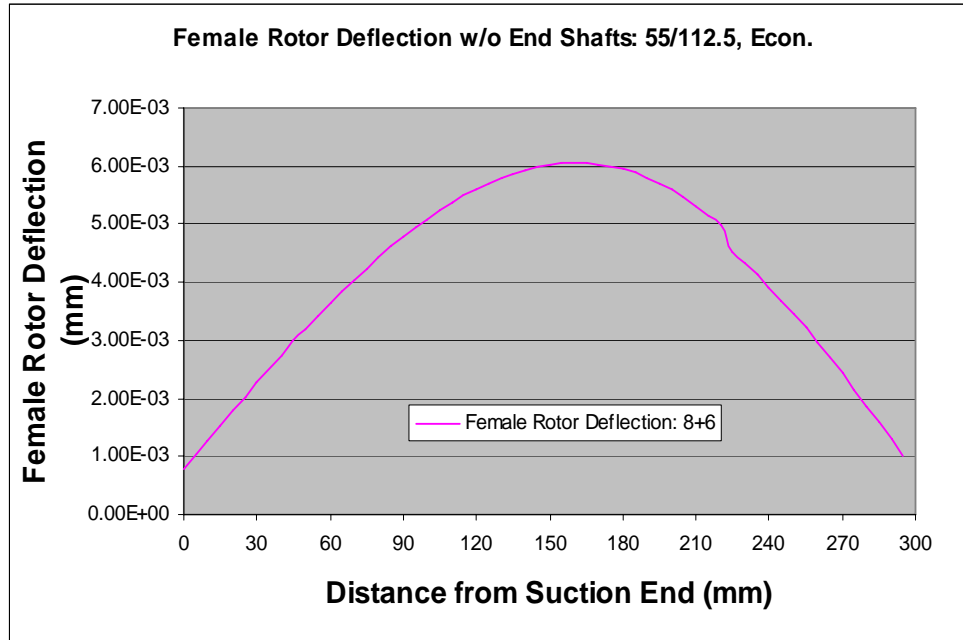


Figure 5: Finite element results of female tri – rotor deflection

Now that deflections have been established we can calculate the leakage ratio based on the three leakage paths mentioned in the 1st paragraph of this section. The basic equation for leakage ratio is shown in equation (4) below.

$$\frac{L_{tw}}{L_{tr}} = \frac{B_{tw} + S_{tw}(Y_f + Y_m)C + T_{f_{tw}}(Y_f + Y_{std}) + T_{m_{tw}}(Y_m + Y_{std})}{B_{tr} + S_{tr}Y_fC + T_{f_{tr}}(Y_f + Y_{std})} \quad (4)$$

Where:

- 1) B = blow hole area
- 2) S = seal line
- 3) Y_f = female rotor deflection
- 4) Y_m = male rotor deflection
- 5) Y_{std} = standard operating clearance
- 6) T = tip sealing line
- 7) L = leakage
- 8) C = weighting factor to incorporate leakage path differences

For the above calculations we made the following assumptions:

- 1) Deflection profile for tri - rotor and twin are equivalent
- 2) No liquid seal at the blow hole and the seal line
- 3) At the rotor tip we have 60% gas, 40% liquid. The twin uses oil at the tip for sealing and the tri - rotor uses liquid refrigerant

Based on the above assumptions the leakage ratio is shown in equation (5). Notice that even though the tri-rotor has no male deflection, the increase in sealing line length with two female rotors results in more leakage area for the tri-rotor.

$$L_{tw} = .80L_{tr} \quad (5)$$

It should be noted that even though the leakage area is greater for the tri – rotor, the max gap between the rotors as defined in equation (6) for the tri – rotor is 62% of the twin gap, see equation (7).

$$Def_{tr} = (Y_{trf} / (Y_{twf} + Y_{twm})) Def_{tw} \quad (6)$$

$$Def_{tr} = .62 Def_{tw} \quad (7)$$

It is this reduction of the total gap shown in equation (7) that allows the designers to maintain efficiency with an oil reduced compressor.

4. DRAG LOSSES

The tri-rotor compressor is an oil reduced compressor operating in a flooded chiller with saturated vapor feeding directly to the suction inlet. The chiller generally runs with less than 15 degrees F superheat. Therefore, all sealing at the rotor tips is with liquid refrigerant while sealing of the oil flooded twin screw is with oil.

For a Newtonian fluid the drag shear stress can be shown to be:

$$\tau = \mu du / dy \quad (8)$$

Where:

- 1) τ = shear stress
- 2) du / dy = velocity gradient
- 3) μ = absolute viscosity

If equation (8) is integrated over the rotor gap and then converted to power, the drag losses are given by equation (9).

$$D_l = \mu V^2 A / Y_t \quad (9)$$

Where:

- 1) D_l = drag losses
- 2) V = tip velocity
- 3) Y_t = avg. tip clearance between rotor and housing
- 4) A = shear area

For the drag loss ratio of tri – rotor to twin we will make the following assumptions:

- 1) The primary drag loss is @ the rotor tip, therefore the end face and seal line drag losses are not incorporated into the ratio.
- 2) The tri-rotor compressor is an oil reduced machine with 166 PPM oil flow
- 3) The twin is an oil flooded machine with 83,000 PPM oil flow

Therefore the drag loss ratio is given by equation (10).

$$D_{ltr} / D_{ltw} = (\mu_{tr} Y_{tr} V_{tr}^2 A_{tr}) / (\mu_{tw} Y_{tr} V_{tw}^2 A_{tw}) \quad (10)$$

With a resulting ratio shown in equation (11).

$$D_{ltr} = .032 D_{ltw} \quad (11)$$

Based on the above equation it becomes obvious that reduced oil compression of the tri – rotor compressor provides a significant reduction in drag losses.

5. DISCHARGE PORT LOSSES

For the intentions of this paper we will calculate the discharge port losses using Bernoulli's equation to calculate the dynamic head and then assume associated kinetic energy is lost. Therefore, the port flow losses are proportional to the velocity cubed and the port power loss can be given by equation (12).

$$P_{tr} / P_{tw} = v_{tr}^3 A_{tr} / v_{tw}^3 A_{tw} \quad (12)$$

- 1) P = Power loss
- 2) v = average gas velocity
- 3) A = port area

The port loss ratio is shown in equation (13)

$$P_{tr} = 1.31 P_{tw} \quad (13)$$

It is interesting to note that the port power loss of the twin screw is less than that of the tri - rotor screw even though there are two discharge ports. The basic reason for this is that the tri - rotor screws are 30% shorter than the twin rotors thus the wrap angle of the tri - rotor needs to be reduced to 165 degrees versus 300 degrees for the twin. With our design constraints, a low wrap angle reduces the gas exit time thus increasing velocity.

6. FULL LOAD PERFORMANCE LOSS

The full load total power loss comparison for the twin and tri-rotor compressors will be based on a generalized weighting factor shown below in table 1. Losses are typical for twin screw compressors.

Twin screw typical loss values

1) Leakage losses (L)	=	8.9%
2) Drag losses (D)	=	4.6%
3) Discharge Port Losses (P)	=	2.7%
Total Loss (T_l)	=	16.2%

Figure 1: Twin Screw loss summary

Using equation 14 the total loss (T_l) for the Tri – rotor screw compressor are shown in equation (15).

$$T_{l_{tr}} = (L_{tr} / L_{tw})8.9 + (D_{tr} / D_{tw})4.6 + (P_{tr} / P_{tw})2.7 \quad (14)$$

$$T_{l_{tr}} = .91T_{l_{tw}} \quad (15)$$

Therefore, we see that the total loss of the tri - rotor screw compressor is less than that of the twin, primarily driven by the reduction in drag losses.

7. PART LOAD PERFORMANCE LOSS

This section will evaluate the impact of variable speed unloading versus slide valve unloading independent of the tri – rotor versus twin rotor screw compression issue. We will use the ideal gas laws to compare the compression work required. See equation (16).

$$W = (P_2V_2 - P_1V_1)/(1 - n) \quad (16)$$

- 1)
- 2) W = work of compression
- 3) P = Pressure
- 4) V = flute volume
- 5) n = gas constant

If we apply equation 15 to a standard P-V diagram using refrigerant properties of R134a and properly account for over compression due to V_i mis-match we can generate the graph shown in figure 6.

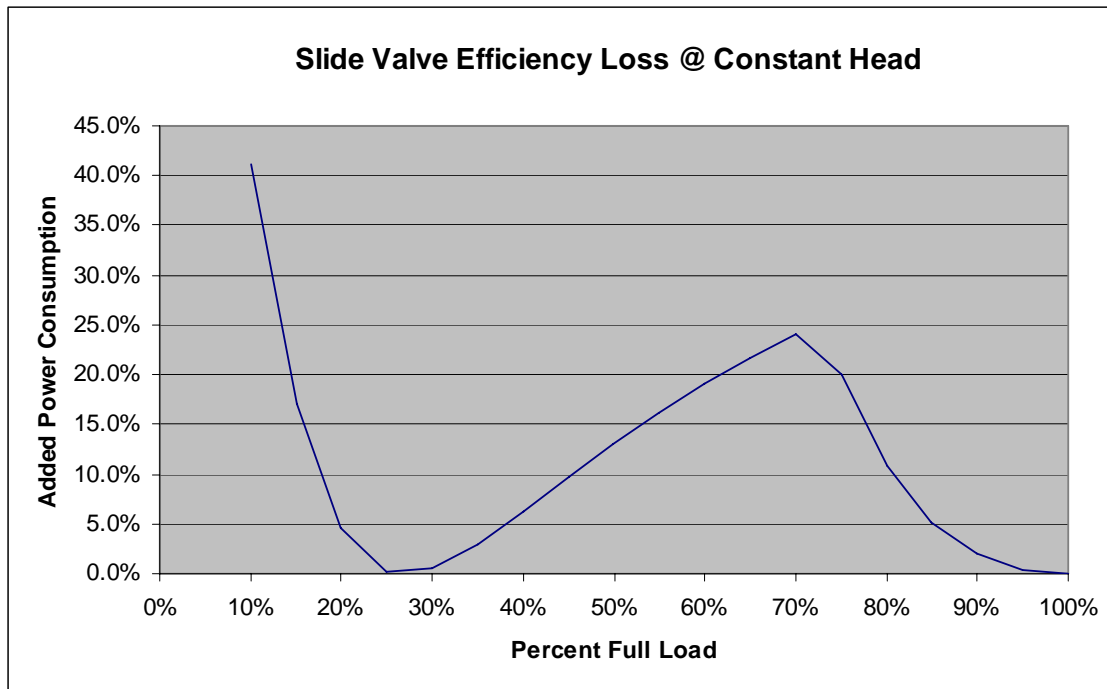


Figure 6: Slide Valve Power Loss

If we apply the Added Power Loss from figure 6 to the standard weighting factors for the ARI part load equation we find that the part load efficiency of a variable speed machine will be 15% higher than that of a slide valve machine. Additional improvement in part load performance will also result from the reduction in drag losses as a variable speed tri – rotor is unloaded. As was pointed out in equation (9), the drag losses are reduced by the square of the velocity.

8. CONCLUSIONS

It was the intent of this paper to analyze the tri – rotor and twin rotor screw compression concepts. We have shown that the primary driver providing the tri – rotor with improved efficiency for both full and part load conditions is the reduced oil operation. The reduction in oil into the compression process reduces the drag losses. We have shown that the tri – rotor has a significant reduction in the total deflection gap between the rotors due to the balanced male compression loading thus allowing oil reduced operation. The tri – rotor further takes advantage of the low drag losses by utilizing variable speed operation to unload the compressor.