

# HYDROCARBON ENGINEERING

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**Sulzer Pumps –  
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# Multistage pumps for NGL

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## COVER STORY

**B**lack Cat Construction, a leading EPC contractor in the state of Qatar, needed a raw natural gas liquid (NGL) injection pump for a reputed end user in the country. The design conditions were:

- 166 m<sup>3</sup>/h (730 gpm) flow rate.
- 2100 m (6890 ft) head.
- 30 m (98 ft) NPSHa.
- 0.45 - 0.53 specific gravity.
- 0.098 - 0.14 cp viscosity.
- 2.7 - 2.79 kJ/kg/°C (0.64 - 0.67 BTU/lb/°F) specific heat.
- 11 - 52 °C (52 - 126 °F) suction temperature.
- 50 bara (725 psia) vapour pressure at 50 °C.
- 55 barg (798 psig) suction pressure with 95 barg (1377 psig) possible.

Concerns in pumping these high vapour pressure fluids include their low lubricity and the possible effects of heat caused by seal face friction and/or pump inefficiencies.

The combination of high delivery head at relatively low flow and 50 Hz operating speed indicated a multistage pump would be required. Since the head produced by a centrifugal pump varies with the square of the speed, a pump running at 42% higher speed could theoretically require half the number of stages. However, for a simpler design with a higher availability, the client requested that the pump be direct drive with no speed increasing gearbox. They also wanted to avoid the use of a force feed lubrication system and its associated higher initial cost, complexity and maintenance requirements.

The project used ISO 13709 (API 610) as the primary specification, requiring a radially split casing for this light hydrocarbon service: designation BB5. Furthermore, paragraph 5.6.9 and Appendix I of the specification require that the pump have rotodynamic stability with two times the normal wear ring and bushing clearances. This added several complications to the pump selection. A number of standard designs were considered.

### Inline rotor design

The most economical horizontal barrel pump is an inline rotor design, i.e. all the impellers face toward the suction end of the pump, creating a very large axial thrust. To coun-

teract the axial thrust generated by the impellers, a balancing drum is installed on the shaft after the last stage impeller. Since these drum length and diameter are selected based upon a set of operating conditions, operation at lower flows produce higher residual axial thrust. With so many stages in this application, the residual axial thrust would require the use of a large pivot shoe (tilting pad) thrust bearing and associated force feed lubrication system.

The primary rotor support, dampening and stiffening devices in multistage horizontal pumps with an inline rotor design are the bearings, the balancing drum and the impeller labyrinths. The low fluid density of NGL, together with low viscosity, are important factors for the dynamic behaviour. If the same pump was specified on water with a viscosity of 1.0 cp (10 times that of the raw NGL), each bushing or wear ring would provide far more support. With such extremely low viscosity, there is less support at each stage in this application. Furthermore, considering compliance with Appendix I of ISO 13709 (API 610), when the wear ring and balancing drum clearances are doubled, the dampening and stiffening effect of those components is substantially reduced. Therefore, for an inline rotor design with so many stages and worn clearances, there is a valid concern regarding rotor stability.

Shaft deflection is a function of  $L^3/d^4$ . If an inline rotor design is the only option, an extremely large shaft must be used to avoid excessive deflection mid span of the rotor. Unfortunately, increasing shaft diameter causes an

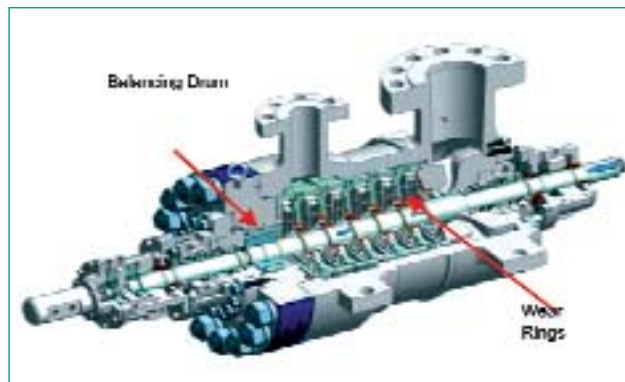


Figure 1. GSG inline diffuser barrel type pump.

increase in the first stage, as well as in the following impeller hub diameters. To provide adequate eye area with a larger hub diameter, the peripheral eye diameter has to be larger. This increased inlet diameter of first stage results in reduced suction performance and increased recirculation at part load. With an inline rotor design, the only alternative is to reduce the number of stages and increase the rotational speed of the pump that requires higher speed, a speed increasing gearbox, pivot shoe (tilting pad) thrust bearings and a force feed lubrication system. The combination of all these components results in a high initial cost and adds a substantial maintenance cost over the equipment's life.

### Back to back rotor design

If half the required stages are arranged in one direction and the other half in the other direction, the axial thrust is almost balanced over the entire operating curve. The back to back design allows the use of a 7300 series ball type thrust bearing without a force feed lubrication system. A long bushing is installed mid span and another bushing is added at the end opposite the suction end of the rotor to reduce half the differential pressure down to suction pressure. This design dramatically improves rotor dampening and stiffness even with wear, as the stiffening effects of the long bushings are so substantial. Such a design allows the use of smaller eye impellers and a greater number of stages. Virtually all the thousands of multistage product pipeline pumps in North America utilise this back to back rotor construction for these reasons.

### Wear part materials

There are hundreds, possibly thousands, of light hydrocarbon pipeline pumps in operation today. Many have been running for more than 30 years. Due to the low specific gravity, some of these pumps have to produce over 50% more head than similar pumps supplying motor fuels into the same pipeline operating pressure. Increased discharge head at the same speed means more stages. More stages equal more impellers and extra weight on the rotor. In longer pumps, the rotating wear parts could actually touch the stationary wear parts when the pump is idle. Even if they do not touch when the pump is idle, providing wear ring materials that are gall resistant is extremely important in non-lubricating services such as this. Upsets, load changes or discharge valve throttling, etc. can all excite the rotor to create deflections beyond the radial wear ring clearances. Wear parts that have touched without galling have served their purpose and allowed the pump to keep operating successfully. Utilising straight S-5 or S-6 materials of construction on this service could result in acceptable factory water performance tests, but disappointing results on light hydrocarbon in the field. In non-erosive light hydrocarbon services, there are a variety of material combinations that achieve this level of tolerance and should be used based upon a manufacturer's experience and wear testing.

### Vapour pressure vs. temperature

Many hydrocarbons have a specific heat coefficient approximately half that of water. For a given amount of heat input, the temperature of a hydrocarbon will increase almost twice as fast as water. Volatile, light hydrocarbons have a relatively steep vapour pressure (VP) vs. temperature curve. For example, water has little VP at 20 °C, but VP of propane at 20 °C is 8.3 bar (126 psia). Increase the temperature to 50 °C, and the VP of water is 0.12 bar (1.8 psia) while the VP of propane has doubled to 17 bar (247 psia). If the pumped fluid were ethane, at 50 °C it would be in the supercritical region and require different equations of state. In

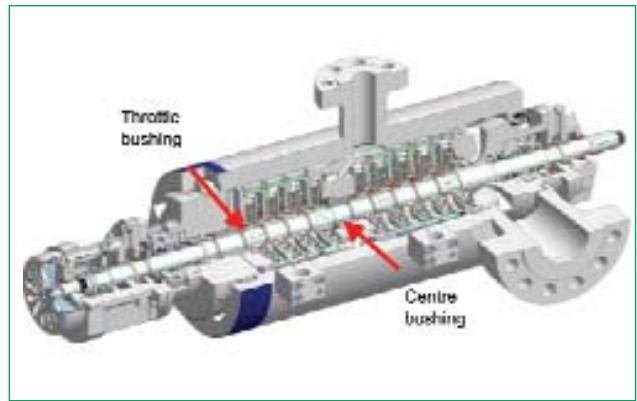


Figure 2. GSG opposed impeller diffuser barrel type pump.

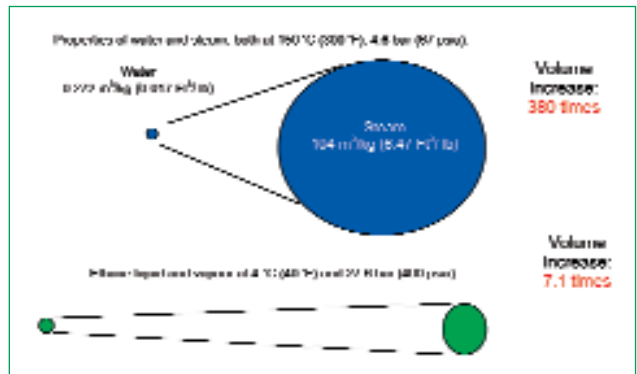


Figure 3. Representation of specific volume increase for water vs. ethane.

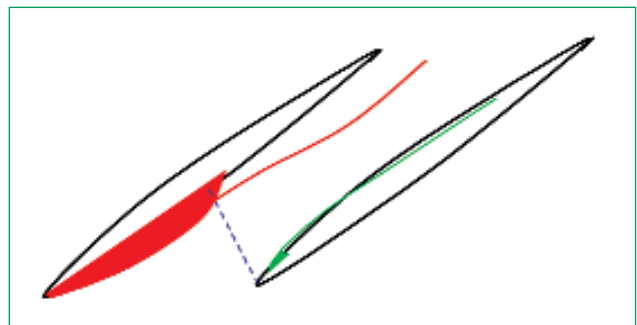


Figure 4. Example of non uniform occlusion of vane space by vapour bubbles.

this application the raw NGL is a mixture of light hydrocarbons with a resultant high vapour pressure.

If the pressure surrounding the fluid is at or near the vapour pressure, vapour bubbles form. High vapour pressure hydrocarbons have a relatively small change in specific volume going from liquid state to vapour state as they are at a much higher percentage of their respective critical pressure. In other words, bubbles that form in light, volatile hydrocarbons at 10 bar (145 psi) are much smaller than a bubble formed in water would be at the same pressure. Smaller bubbles take up much less space between impeller vanes. Thus, the average density of the fluid/bubble mix between the vanes is little changed with a small, light hydrocarbon bubble. However, the average density of the fluid/bubble mix would be lowered dramatically for a bubble formed in water due to the large size of the much lighter vapour in water.

As the vapour bubbles are not uniform between all the vanes of an impeller, an imbalance and an uneven pressure are created in the volute or diffuser, which causes vibrations. The small density change with light hydrocar-

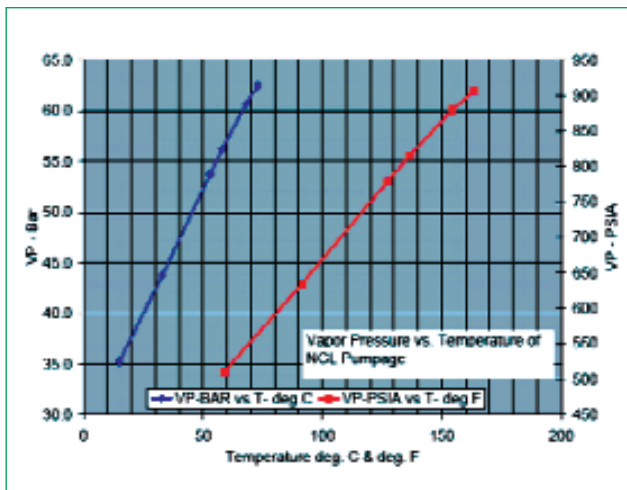


Figure 5. Vapour pressure vs. temperature for NGL pumpage.

bons helps explain why pumps can run at higher flow rates with less NPSHa on light hydrocarbons. It also explains why the vibration signature may not change much with lower suction pressures, to a point. However tolerant volatile fluids may be, flashing across bushings or balance devices within the pump should be avoided as it can cause oscillations and rotor vibration to the point of wear surface contact.

### Temperature rise

The temperature rise of a stage of a centrifugal pump is a function of the head generated by that stage, the specific heat of the fluid and the efficiency of that stage of the pump:

$$TR_C = \frac{H \times (1 - E)}{102 \times C_p \times E} \quad TR_F = \frac{H \times (1 - E)}{778 \times Sp.Ht. \times E}$$

Where:

TR = Temperature rise in degree Celsius or Fahrenheit.

H = Total head in metres or feet.

E = Efficiency.

Cp = Specific heat in kJ/kg/°C = 2.1 for many hydrocarbons, 4.2 for ambient water.

Sp.Ht. = Specific heat in BTU/lb/°F = 0.5 for many hydrocarbons, 1.0 for water at ambient temperature.

For pumps with an inline rotor design, the total head can be used to calculate temperature rise. Thus, in this application, with a pump efficiency of 65%:

$$TR_C = \frac{2100 \times (1 - 0.65)}{102 \times 2.7 \times 0.65} \quad TR_F = \frac{6890 \times (1 - 0.65)}{778 \times 0.65 \times 0.65}$$

$$TR_C = 4.0 \text{ } ^\circ\text{C} \quad TR_F = 7.3 \text{ } ^\circ\text{F}$$

At rated conditions discharge temperature would be approximately 11 °C + 4.0 °C = 15.0 °C (59 °F). Vapour pressure at 15 °C (59 °F) is approximately 35.2 bar (511 psia) from the VP chart (Figure 5).

In this example, the temperature rise values are relatively small. When the pump is operated back on its curve at approximately 40 m<sup>3</sup>/h and 2570 m head with 30% efficiency, the temperature rise increases to 21.4 °C (38.5 °F). The vapour pressure at 11 + 21.4 = 32.2 °C (90.4 °F) is approximately 43.5 bar (630 psia) from the same VP chart. In the summer, startup conditions with suction temperature of 52 °C (126 °F) plus temperature rise of 21.4 °C mean the discharge temperature is 73.4 °C (164 °F), which has a corresponding VP of 62.6 bar (908 psia).

The balancing drum of an inline rotor design pump sees a discharge pressure and temperature as the balancing drum is located adjacent to the last stage impeller. The

opposite side of the balancing drum is normally subjected to suction pressure. Due to the temperature increase of the fluid through the pump and its much higher vapour pressure, bubbles begin to form partway across the balance drum as the pressure is reduced. As the fluid approaches suction pressure on the far side of the balancing drum, larger bubbles form non-uniformly and can result in vibration. Using orifice plates in the balance leakoff restricts the leakoff flow and increases back pressure on the low pressure side of the balance drum. However, restricted balance leakoff flow also affects the residual thrust load and must be considered in sizing the thrust bearing. If the balance line on an inline design rotor pump is piped back into the suction nozzle of the pump, flashing can occur and bubbles can be entrained into the first stage, further exciting rotor vibration. Therefore, for inline rotor design pumps, it is prudent to pipe balance leakoff flow to the suction vessel or well upstream of the pump if on light hydrocarbon pipeline service.

For back to back rotor design pumps, only approximately half the number of stages and thus half the energy has been imparted to the fluid before it is bled back to suction pressure through the throttle bushing on the end of the rotor opposite the suction. For an equal number of stages stacked in opposite directions, simply use half of the above temperature rise values. At 40 m<sup>3</sup>/h and 2570 m head with 30% efficiency, the temperature rise to the throttle bushing is approximately 10.7 °C (19.5 °F). The vapour pressure at 11 °C + 10.7 °C = 22.7 °C (73 °F) is approximately 38.8 bar (563 psia) from the VP chart above. For summer startup conditions of 52 °C (126 °F), plus temperature rise of 10.7 °C, discharge temperature is 62.7 °C (145 °F), which has a corresponding VP of 58.4 bar (846 psia). The minimum pressure around the centre bushing is at least half the total pump differential pressure plus suction pressure, so flashing across the centre bushing is virtually impossible while any flow is moving through the pump. Even with a back to back rotor on a high vapour pressure fluid, it may still be prudent to pipe the leakoff from the throttle bushing back to the suction vessel or well upstream of the pump if no suction vessel is available in a pipeline application.

The issue of temperature rise at the mechanical seal faces has been the subject of many articles and papers. Clearly, for contacting face seals, light hydrocarbon service requires much higher seal flush rates, special face contours/materials, etc. Gas seals are sometimes specified. Discussing these requirements with the seal supplier can substantially improve the life of the mechanical seals.

### Conclusion

When the service conditions require a BB5 horizontal barrel pump, an inline rotor design construction is usually the least expensive and thus desirable when the pump is small enough for antifriction thrust bearings be utilised. However, when larger pumps, many stages or high vapour pressure conditions are contemplated, the back to back rotor design may prove to be not only much less expensive, but also substantially more reliable. Full understanding of the vapour pressure vs. temperature relationship and specific heat characteristics of the pumped fluid are important on high vapour pressure fluids. On multistage pumps in particular, non-galling wear part materials are likewise critical to help withstand operating upsets on extremely low viscosity fluids. Communication between the client, EPC and pump supplier can result in economical, reliable applications, reducing both installed cost and lifecycle cost.