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Technical Aspects of Pumping Warranties Explained

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Abstract: The highly technical aspects of pumps and piping networks are often overlooked in pumping disputes. This may lead to lack of technically sound defense. This article attempts to bridge this gap. The discussion will be focused on steam driven centrifugal pumps installed on crude oil VLCC tankers but it can straightforwardly be extended to any size of centrifugal pump and tanker type. Pump characteristics will be firstly explained and the operation of a pump in a piping network will then be examined. The use of two or more pumps in the presence of a network will be also introduced before applying the technical concepts to a typical discharge operation of a tanker. Pumping warranties will be discussed in attempting to examine what conditions must prevail to allow a ship to perform within the charter party restrictions. It will be shown that the ship must be offered separate shore lines to discharge the cargo in 24 hours. For conditions where there is high back pressure, a ship can meet a common guarantee of 7 kg/cm^2 at the manifold provided the pumps are maintained in good condition, while meeting higher pressure warranties is subject to the technical limitations of the pumps. Fuel oil consumption concepts will also be discussed for time charter parties. A warranty should take into consideration that the fuel oil consumption for pumping depends strongly on the manifold pressure but is also affected by prolonged pumping times.

Key words: Charter parties, warranties, pumping.

1. Introduction

Pumping warranties are type of commercial warranties a vessel shall meet which involve highly technical issues, perhaps more than any other type of warranty. Yet, the disputes which usually arise are not technically focused. This is, in part, inevitable because the technical aspects of pumps and piping networks fall outside the tanker industry commercial, legal or operational daily practice. One could say that the pumps and the piping networks are treated as a “black box” which must deliver the work needed but nobody is fully aware how it works. Without a technical support, however, any argument put forward to defend one’s position will fail to withstand a rigorous examination.

In this article, an effort is given to present the technical aspects of pumping warranties in a simplified form without sacrificing the scientific accuracy. It is hoped that the curious reader will find this article a good starting point to explore further how pumps perform on a tanker and in a shore piping and shore tank installation, while the legally or commercially minded professional will get the tools needed to support his/her case.

A common pumping warranty stipulated in most voyage and time charter parties provides for a ship to discharge her entire cargo within 24 hrs (the “time” warranty criterion) or to maintain 100 *p.s.i.* at the manifold [1] (the “pressure” warranty criterion), provided that the shore facilities permit [1].

It is important to highlight that pumping warranties are often considered to describe and evaluate the operational performance and effectiveness of the pumps on a tanker. This is correct only partially. The performance and effectiveness of a pump is determined, as we shall see, to a high extent by the piping and shore tank network to which the pump is connected, and one cannot examine a pump as a stand alone equipment.

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[1] The International Standard unit for Pressure is N/m^2. However, in the practical world of tankers, pressure is expressed either in kg/cm^2 or *p.s.i.* (pounds per square inch). For our purposes, it suffices to say that pressure of 100 *p.s.i.* corresponds to 7.03 kg/cm^2 (or 6.9 10^5 N/m^2 or 6.8 *Atm*). Similarly, the International Standard unit for density is kg/m^3 while in tanker practice it is expressed in gr/cm^3.

Knowing the piping and receiving tank network is essential to describe correctly the performance of a pump.

Commonly asked questions are:

1) If shore facilities do permit, can a tanker vessel discharge her cargo in 24 hrs?
2) What do we mean by shore facilities? Is it only the tank height? What about the number or the diameter of the shore lines? What conditions determine that the shore facilities permit a 24 hrs discharge?
3) Is a "pressure" warranty of 7 kg/cm^2 or 100 *p.s.i.* feasible to be met? Is there an upper limit of the pressure a vessel can achieve?
4) In case the vessel does not meet the "time" or the "pressure" warranty, how much time are the charterers entitled to deduct from the laytime?
5) Why is a high back pressure usually considered a negative factor not allowing the ships to pump at their maximum capacity?

The discussion which follows will attempt to address all the above points. This article deals only with centrifugal pumps.

2. Centrifugal Pumps and Piping Networks

A pump is a device which enables the transfer of liquid from point A at height H_A to point B at height H_B, with usually, but not necessarily, $H_B > H_A$. This is achieved at the expenditure of energy. We can visualize this concept in familiar terms: let us imagine a mass m standing at height H_A (e.g., a person standing at the bottom of a cliff) which must be lifted at height H_B (the top of the cliff). There is a difference in the gravitational potential energy which is proportional to the mass and the height difference. Specifically, the difference in potential energy of a mass m between two points A and B is $mg\Delta H$, where $\Delta H = H_B\text{-}H_A$, g being the gravitational acceleration (= 9.8 m/sec^2). This energy must be provided by a source, in the case of a person climbing the cliff the energy is provided by the body in the form of chemical energy, in the case of the liquid it is provided by the energy (steam or electrical) driving the pump. For a frictionless cliff, or a frictionless piping from point A to B, the only energy needed to be overcome is the potential energy difference. If there is friction on the way from point A to point B, one has also to overcome the frictional losses.

A pump takes energy from its source (steam or electricity) and offers it as useful energy, ready to perform a work to lift the liquid to a new height or to be transported a certain length. A steam driven centrifugal pump transforms steam energy to mechanical energy of the shaft of the pump and then to kinetic energy of the liquid molecules. As for any device, the efficiency, n, of the pump is less than 1, meaning that the useful output energy offered to the system by the pump is less than the energy given to the pump by the source. The performance of a pump is usually described by its Total Head-Capacity curve [2-4]. This curve shows the net height a liquid, often water, is displaced as a function of the flow rate in m^3/hr, and it is usually given for the maximum *r.p.m.* (rounds per minute or RPM). This is shown in Fig. 1. The Total Head in the vertical axis is the difference between the head on the delivery side, H_D, of the pump minus the head on the suction side, H_S, plus some other terms which are much smaller and can be safely neglected. One would ask why we use the term "head" instead of "pressure". The reason is that the pressure at the bottom of a particular column of a liquid depends on the density of the liquid, so one wants to have a curve which can be easily applied to liquids of different densities. Knowing the head in m, one can derive the pressure for the specific medium, and vice versa[2].

One must not overlook the fact that a centrifugal pump, as the ones installed on tankers, is designed to work at its best, it is optimized as said, at relatively high

[2] The conversion between Head, H, in m and Pressure, P, in kg/cm^2 is given by the relation: $H = 9.8P/sg$ (sg (specific gravity) = the ratio of the density (mass of a unit volume) of a substance to the density of water).

head. This Head corresponds to a specific flow rate (Capacity). This pair of values of Total Head and Flow Rate defines the Best Efficiency Point (BEP). For the case of Fig. 1, the Best Efficiency Point corresponds to head of 140 *m* and to flow rate of 5,500 m^3/hr. The pump performs most well, with minimum wear and with maximum efficiency, when it is operated close to the BEP. There are concentric lines around the BEP along which the efficiency remains constant. These lines are called iso-efficiency curves. The further from the BEP (the center) the lower the efficiency is. Alternatively, the manufacturer may provide the efficiency *n* against Capacity at various RPM.

There is a Head/Capacity area around the BEP which we may call the 'comfort zone'. This means that a pump operates quite efficiently in this zone, but operating the pump outside this comfort zone may damage the pump. For example, if the pump of Fig. 1 is operated under condition of Head 110 *m* and Flow Rate 8,000 m^3/hr, it may pump the liquid faster but such an operation may eventually damage the pump.

As we have said, the performance of a pump is described by the Total Head-Capacity curve. The exact point where the pump will operate on this curve will depend on the energy demand of the system. The energy demand is determined by two factors: one is the height where the liquid must be transferred to, and the other is the friction which must be overcome during the transportation. As to the frictional losses, in general: a)

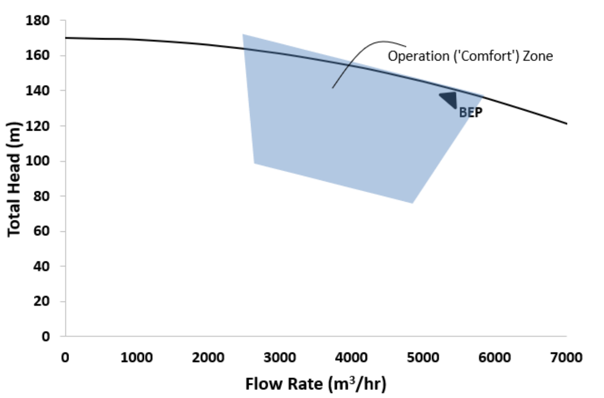

Fig. 1 Head-Capacity curve of a centrifugal pump showing the Best Efficiency Point at Head 140 m and Flow Rate 5,500 m³/hr. The curve corresponds to 1,200 RPM.

the wider the pipe which carries the liquid, the lesser the frictional losses are, b) bends, turns, fittings and valves on the way increase the frictional losses, c) the higher the transfer speed, the higher the losses are (frictional loses are proportional to the square of transfer speed or flow rate). The demand of the system is represented by the System curve. The operating point of the pumps will be at the intersection between the Total Head-Capacity curve and the System curve, as shown in Fig.2. So, the operating point is not determined by the pump alone but by the system {cargo tank-pump-piping-shore tank}. The System curve is usually the big unknown in any pumping and piping installation. The same holds for a tanker discharging liquid cargo to a shore tank. Despite this, there is a way to determine the System curve without much analysis. The System curve is of the form $H=a+bQ^2$, where a is the static head, b is a parameter characteristic of the frictional losses and Q is the flow rate. By running the pump, for example, at two different RPM for one hour and measuring the flow rate and the Total Head for each RPM, one can determine the parameters a and b and then can have the complete System curve.

The RPM of the pumps installed on tankers is adjustable, enabling the ship to meet varying conditions. By lowering the RPM, the Total Head of the pump is lowered, and the Total Head-Capacity curve is shifted downwards. The Total Head H_1 at RPM_1 and the Total Head H_2 at RPM_2 are related via the expression $H_1/H_2 = (RPM_1/RPM_2)^2$. Using this expression one can derive the Total Head-Capacity curve at different RPM by knowing the Total Head- Capacity curve at the maximum RPM. This expression is part of a family of relations called Affinity Laws, which express various parameters at different shaft speeds or RPM [2]. Another expression of Affinity Law is $Q_1/Q_2 = RPM_1/RPM_2$, where Q is the flow rate (volume per unit time, for example m^3/hr). It is worth mentioning that the BEP of tanker pumps lies on the highest RPM where a pump can operate. In other words, as

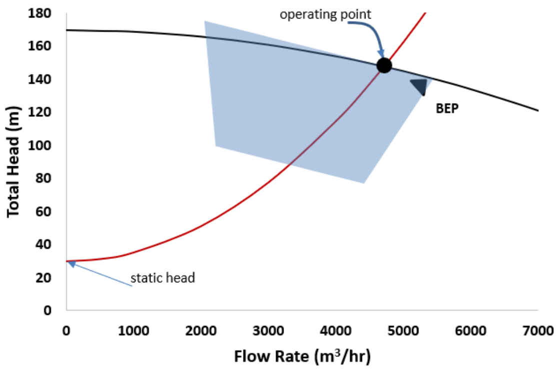

Fig. 2 Total Head-Capacity curve (black line) and System curve (red line). The System curve is shown with static head corresponding to shore tank elevation of 30 m. The point where the two curves intersect will be the operation point of the pump into the piping system it is connected. The operation point does not have to be the BEP but it must be within the "comfort" zone.

previously discussed, a pump works at its best at highly demanding conditions.

A tanker seldom does it utilize one pump to discharge the cargo. Usually, two or three pumps are used at the same time. There are two ways to connect two or more pumps to a pumping system. One is to connect them in series, that is one after the other. In tanker terminology, the second pump is often called buster pump and is connected ashore. This configuration helps to increase the final head, so the liquid can be transferred to shore tanks at higher elevation. The other way is to connect the pumps in parallel, that is their discharge sides are connected to the same shore pipe (the suction sides may be connected to the same or different tank or tanks). The parallel configuration needs to be examined in more detail to understand the pumping warranties because it is the configuration used in most cases on tankers and tanker terminals. To construct the Total Head-Capacity curve of the combined system of two or more pumps connected in parallel running at the same RPM, one works as follows: at any Total Head, the Flow Rate for two pumps running in parallel is twice the Flow Rate of one pump. In the same way, the Flow Rate for three pumps running in parallel is three times the Flow Rate of one pump at any given Total Head. This proportionality refers to the Total Head-Capacity curve of the system of pumps and not to the operating point. The final operating point of the system of the pumps connected to the network will be at the intersection of the combined Total Head with the System curve. Once we get the Flow Rate of the combined system, we can deduct the Flow Rate of each pump of the system (for two pumps in parallel the Flow Rate for each pump will be half the Flow Rate of the combined system, for three pumps in parallel the Flow Rate for each pump will be one third of the Flow Rate of the combined system, etc.). By connecting the combined system of pumps to a piping network, the aim is to increase the Flow Rate. The Total Head will also increase but this will be a by-product of the final configuration.

It is important to note that by connecting two identical pumps, operated at the same RPM to the same shoreline, will not double the flow rate. The flow rate will increase but the extent of the increase of the flow rate depends on the System curve. Fig. 3 shows the Total Head-Capacity curve of one, two and three pumps connected in parallel, and two System curves (one with low static curve and low frictional losses and another with high static head and high frictional losses). The operating point is always at the intersection of the Total Head-Capacity with the System curves. The red System curve will cause each pump to operate out of its comfort zone, so the example given here is for demonstration purposes only and should be avoided in practice. The graph shows that for a System curve with high static head and high frictional losses, the increase in the Flow Rate is marginal as we connect two pumps in parallel, and insignificant when a third pump is connected.

3. Pumping Warranties

We now have all we need to understand the technical aspects of pumping warranties. A tanker is commonly fitted with three identical centrifugal pumps. When the pumping of a homogeneous cargo starts, the cargo officer will utilize one or two or all pumps to transfer

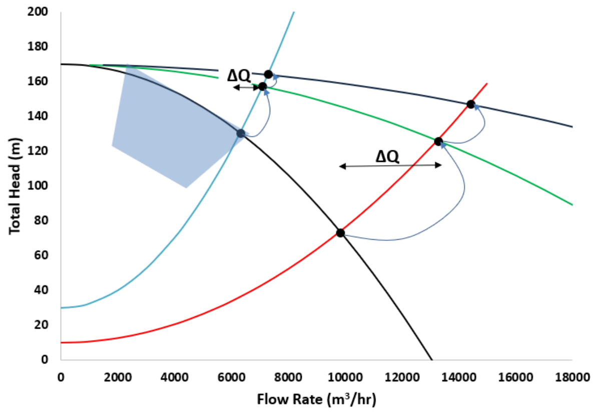

Fig. 3 Total Head-Capacity curve of one pump (black line), two pumps (green line) and three pumps (dark blue line) connected in parallel. Two System curves are also shown, one with low static head and low frictional losses (red line) and one with higher static head and higher frictional loses (light blue line). The change in the Flow Rate for each case is denoted by ΔQ.

the cargo from the cargo tanks to the manifold and then to shore.

For our calculations we will use technical details applicable for a Very Large Crude Carrier (VLCC). The cubic capacity of the ship is taken to be 330,000 m^3 and the vessel is taken to be equipped with 3 pumps, each one capable to deliver 5,500 m^3/hr at head 140 m at 1200 RPM. These values define the BEP of the pumps and together with the Total Head-Capacity curve provided by the manufacturers can describe the whole pump performance. As shown, by knowing the curve at one RPM value we can derive the corresponding curve at any other RPM value. We will restrict the discussion here to liquids of viscosity close to that of water. For liquids of viscosity different than water, the Head-Capacity curve is modified. In general, the higher the viscosity, the lower the Total Head will be for the same flow rate [5].

It is straightforward to see that using only one pump will take 60 hrs (330,000 ÷ 5,500), so the vessel will not meet the pumping warranty of 24 hrs. Here, we exclude any additional time for Crude Oil Washing and striping, both of which are not be examined here as we deal only with bulk discharge. The pumping warranty of 24 hrs can be met by using three pumps. Indeed, the discharge time in this case would be 20 hrs (330,000 ÷ (3 × 5,500)), but this will happen only if the three pumps work independently: each one is connected to one of the three segregated cargo tank groups, and, most importantly, is connected to separate manifold and shore line and the final transfer takes place to individual cargo tanks at shore. We have seen previously that by using two pumps in parallel the flow rate is not doubled, and by having three pumps in parallel we are far from having the flow rate of one pump multiplied by three. So, when the pumping warranty demands the vessel to discharge within 24 hrs *provided shore facilities permit*, it is meant that the shore should supply three shore tanks and three shore lines and that the vessel will be allowed to discharge with three pumps, each one pumping cargo from separate set of segregated cargo tanks to separate shore pipelines and shore tanks.

From the above it can be deducted that discharging at a terminal with low back pressure (manifold pressure below 7 kg/cm^2 or 100 *p.s.i.*), the vessel is obliged to follow the ‘time’ warranty criterion, and this will not be met easily with the pumps connected in parallel and restricted to operate within their comfort zone.

If, on the other hand, the vessel is required to keep 7 kg/cm^2 or above at the manifolds, and provided there is sufficient back pressure (sufficient static head and frictional losses) and the pumps are operated close to their maximum designed RPM, the vessel should not have any issue to meet this “pressure” warranty irrespective of the flow rate and irrespective of whether the pumps used are connected in parallel or they pump cargo to separate shore pipelines and shore tanks. A vessel may be asked by the terminal to maintain a higher manifold pressure, sometimes 10 kg/cm^2 or even 11 kg/cm^2. Although the vessel will perfectly meet the c/p warranty by maintaining 7 kg/cm^2, not meeting a higher manifold pressure may result to rejection of future calls. So, a question arises on whether there is an upper limit of manifold pressure a vessel can achieve. This is determined by the pump settings. The pumps are commonly constructed with a trip alarm which is set at a specific pressure at the delivery side of the pump. When this pressure reaches the set point, the pump is automatically shut down. For a pump of the size fitted on a VLCC, the set point is usually 14.5 kg/cm^2. There is an equation which governs the drop of pressure between two points of a piping system, in our case these points being the delivery side of the pump and the manifolds. The equation is known as the Bernoulli equation, which will not be explained here, but it can simply be stated that between the delivery side of the pump and the manifold, the pressure suffers two kinds of loss: One kind of pressure drop is due to height difference between the pump and the manifold, which is about 30 m and corresponds to a pressure drop of about 2.5 kg/cm^2 depending on the density of the cargo. Another is due to frictional losses along the piping system, which adds 1-1.5 kg/cm^2 to the losses. So, the max pressure a vessel can achieve at the manifold is about 10.5-11 kg/cm^2 before the pump is automatically stopped. In practice, a vessel is rarely operated to achieve manifold pressure above 10 kg/cm^2 and almost never close to 11 kg/cm^2. This limit is consistent with Total Head of 140 m at the BEP.

In case the vessel is asked to maintain a certain pumping rate but fails to do so, one can easily estimate the additional time taken by calculating firstly the theoretical time had the vessel been able to achieve the requested flow rate and comparing it to the actual time taken, on the condition that the shore has provided the facilities needed to achieve the requested flow rate (and that the pumps and boilers are maintained in good condition). We have seen earlier that this provision means that the piping and shore tank configuration allow the ship to pump the cargo to different shore pipelines and shore tanks. In case the vessel is not able to meet the requested pressure at the manifold, the calculation of the additional time taken is more complicated. The most widely known method is to use the so-called ASDEM formula which calculates the theoretical flow rate at the requested manifold pressure by knowing the actual flow rate as achieved at the actual pressure achieved at the manifold. According to ASDEM formula (ASDEM, 2000), the Flow Rate Q_1 and Q_2 at manifold pressure H_{M1} and H_{M2} respectively are related by $Q_1 = Q_2\sqrt{(H_{M1}/H_{M2})}$. The ASDEM formula is based on Affinity laws for centrifugal pumps which relate the Total Head to the Flow Rate (or RPM) of the pump. However, the Total Head of the pump is not the same as the head or the pressure at the manifold. The two are related through an algebraic relation but the one is not proportional to the other. This makes the ASDEM formula an approximate method for the calculation of additional time taken, and the formula will not be able to withstand a rigorous justification. If, on the other hand, the ASDEM formula is agreed in the charter party, as in BP VOY5, one should apply it safely. Simply stated, the ASDEM formula is accurate at zero static head, when the shore tanks are placed at the same height as the tanks on the ship (that is at sea level), but underestimates the theoretical flow rate as the static head increases [6].

A high back pressure is often blamed as being the reason not allowing the vessel to pump her cargo at full pumping rate. This argument is not correct. As we have seen, the centrifugal pumps installed on tanker ships are designed to work optimally at high RPM and high Total Head. So, a high head is not the reason for not pumping at high flow rate. Rather, the reason might be the shore tanks and shore lines configuration, which may not allow the full flow rate to be achieved. If there is a common shore line to which the pumps are connected, the pumps will run in parallel. The reason for low achieved flow rate lies in the system curve and the pump curve of the system of pumps connected in parallel: a sharp system curve (piping network with high frictional losses) with high static head (high elevation of shore tanks) permits a limited increase in the flow rate when two or more pumps are connected in parallel (Fig. 3), while a shallow system curve (piping network with low frictional losses) with low static head (low elevation of shore tanks) permits a considerable increase for pumps running in parallel (though not proportional to the number of pumps used). Had the pumps been able to pump cargo from separated segregations and be connected to individual shore lines and shore tanks, a tanker would perform very well at high back pressure.

4. Discharge Consumption Warranties of Time Charter Parties

Time charter parties contain additional warranties for the fuel oil consumption needed to discharge the cargo. Such additional warranties can be broadly distinguished into two categories: one that stipulates daily consumption, for example 80 tones (*mt*) of fuel per day, and one that stipulates a lumpsum amount, for example 140 *mt* for whole discharge (plus a daily amount for generators). Again, we restrict our discussion to bulk discharge only. A question usually arises as to which description is more representative of, and preferable to describe, the average performance of a tanker. Neither description does it take into account the fact that the energy demands may differ from one discharge terminal to another due to different static head or piping system and frictional losses, or the fact that there might be flow rate restrictions which prolong the discharging time. It is very important to understand these energy demands which come from two sources, as shown above. One is due to shore tank height: the higher the shore tank is above sea level, the more energy must be consumed to pump the cargo. The second energy demand is due to frictional losses: the longer the shore line or the more fittings, bends and valves are involved, the more energy must be demanded by the ship. Two factors which must be considered are that, as we have seen, the frictional losses increase with the flow rate and that as we move away from the BEP the efficiency of the pump drops. In summary, by knowing the Total Head of the pump, one can calculate the theoretical energy output needed to achieve this head, and using the efficiency of the pump one can calculate the theoretical energy input and convert this to steam and then to fuel oil consumption (Appendix). All these are perhaps too complicated to be considered when agreeing a pumping fuel oil consumption warranty, however the present discussion shows that different terminals (with different shore heights and piping networks) impose different demands to be met. A lumpsum fuel oil consumption warranty seems more appropriate to describe the actual energy needs, if the back pressure is known in advance. This will be the case of a charterer employing a ship for a dedicated terminal which requirements are known. If the back pressure is not known in advance, it is recommended to agree a lumpsum fuel oil consumption for the most demanding case. A lumpsum fuel oil warranty on the other hand will fail to describe correctly the actual consumption when the discharge is severely prolonged due to possible low flow rate requirements. In such a case, although the theoretical energy demand is the same as for shorter discharge operations, there will be energy losses on the ship as the operation of the pumps and the boilers cannot be optimized and some of produced steam and related work will be wasted. This exceptional condition is better described by a daily fuel oil consumption warranty or hybrid consumption warranty.

A proposed warranty could then be as follows:

> Vessel shall discharge her bulk cargo within 24 hrs or maintain 7 kg/cm^2 at the manifold provided shore facilities permit. Fuel Oil consumption, at 7 kg/cm^2 manifold pressure or less, shall be 130 mt for whole discharge plus 8 mt/d. For any additional kg/cm^2 the fuel oil consumption shall increase by 12% pro rata. In case the bulk discharge is prolonged in excess of 36 hrs due to shore reasons, the fuel oil consumption shall be increased by 1.5 mt/hr.

It must be noted that the above proposed pumping warranty clause must be treated as a model one. The user may alter the figures, tailor or modify the text according to what has been agreed by the contractual parties.

A good technical knowledge of the pump and boiler characteristics installed on the tanker as well as good record keeping of historical discharge operations, shore characteristics (elevation and distance of shore tanks), shore restrictions, pumping performance and consumption is essential to know what to expect, even on a rough scale, in a situation to be encountered. There are also software tools available which help the user to simulate the pumping performance to a very high degree of accuracy, and to calculate energy needs as well as to optimize the handling of the pumps on board for minimum energy consumption [7].

5. Summary

An attempt has been made to discuss the technical background of pump performance with an effort to make this knowledge accessible to a shipping professional not familiar with engineering concepts. It has been shown that the way a pump will finally perform depends on a large degree on the shore piping and the shore tank configuration. To discharge a full cargo in 24 hrs, as a typical pumping warranty stipulates, would require the shore facilities to allow separate shore lines and shore tanks for each one of the pumps and the tank groups on board. Meeting a pressure warranty will not impose serious challenges, provided that the requested manifold pressure is kept below the operational limit of the pump. Finally, the consumption allowance in a typical pumping warranty for a time charter party should take into consideration that the consumption depends on the geometrical and hydraulic characteristics of the shore network, all these being represented in the manifold pressure. Agreeing an amount for the fuel consumption without knowing the exact conditions to prevail in a discharge operation should be based on the most demanding case, or, at least, should include a description for energy consumption which steps up according to manifold pressure observed and possible prolonged time imposed by the shore.

Appendix

The offered power of a pump is P_o = flow rate × density × head × g. The consumed power of the pump is the offered power divided by the efficiency, $P_c = P_o/n$. For a ship with low rate of one pump 4,500 m^3/hr or 1.25 m^3/sec, Total Head 150 m, liquid density 960 kg/m^3, the offered power of one pump is P_o = 1.25 m^3/sec × 960 kg/m^3 × 150 m × 9.8 m/sec^2 = 1,764 kW. For n = 0.75, the consumed power is 1,764/0.75 = 2,352 *kW*. From the pump manual we can get, for example, that a pump running at full capacity of 2,500 kW output requires 22,550 kg/hr of steam, so 2,352 kW will require 21,215 kg/hr (= (2,352 × 22,550) ÷ 2,500) of steam. From the boiler's manual we can get that 40,000 kg/hr of steam consume 3,013 kg/hr fuel oil. Therefore for 2,352 kW of power requirement from the pump, or 21,215 kg/hr of steam, the boiler will burn 1,600 kg/hr (= (21,215×3,013) ÷ 40,000) of fuel oil. A full cargo of 330,000 m^3 at a rate of 4,500 m^3/hr will require 73 hrs for the bulk discharge. The ship will burn 117 *mt* (= 73 ×1.6) of fuel oil for the bulk discharge of the cargo.

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