ANALYSIS OF STEAM TURBINES FOR FEED WATER PUMPS ON LNG SHIPS

Martina Živković, B.Sc.
Antuna Barca 9D, Zadar, Croatia
martina.latinovic@gmail.com

Josip Orović, D.Sc.
Igor Poljak, M.Sc.
University of Zadar, Maritime Department
Mihovila Pavlinovića 1, Zadar, Croatia
jorovic@unizd.hr, ipoljak1@unizd.hr

ABSTRACT

Steam propulsion plants for LNG ships have main feed water pumps that are driven with steam turbines. Those steam driven feed water pumps are one of the most important components in the plant. Their function is to dose the feed water for main steam boilers timely and exactly in order to maintain normal functioning of the steam propulsion plant. In this paper the mostly used steam driven feed water pumps are presented. Their construction details, steam temperatures, pressures, velocities, efficiencies and losses over the steam turbine stages and wide range of steam turbine operating loads are analyzed.

Keywords: Steam turbine, feed water pump, working parameters, efficiency

1 INTRODUCTION

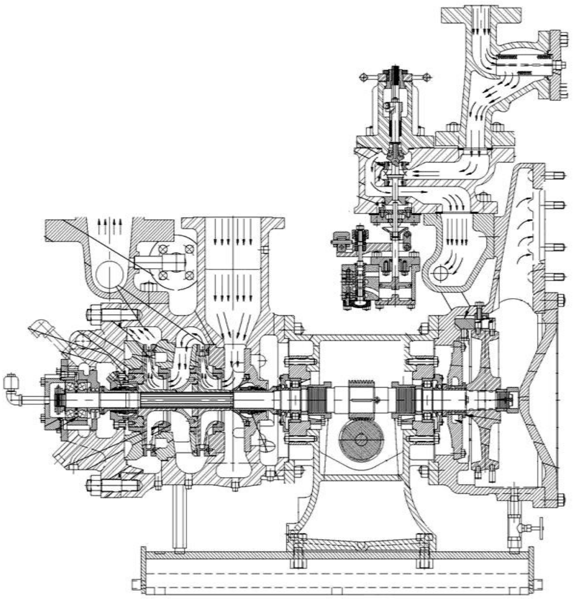
This paper presents two types, Coffin and Shinko, feed water pumps that are driven with steam turbines and used to deliver the feed water to main steam boilers of LNG ships. Coffin main feed water pump is constructed as a two staged centrifugal pump [1] and Shinko main feed water pump is of three stages centrifugal type pump [2]. Both feed water pumps are driven by the short and light weight Curtis wheel steam turbine [1, 2].

Feed water pumps take water suction from the highest point in the engine room, from the deaerator which acts as a header tank, and deliver the feed water with high pressure, over 60 bar, into the main steam boilers. Before it enters the main steam boilers the feed water temperature is usually raised passing through the high pressure feed water heater and economizer in order to reduce the fuel oil consumption in the boilers and to increase the overall efficiency of the steam plant.

The main steam boilers level control and thus the capacity control of main feed water pump is often a three term control. This control measures water level, steam and water flow and controls the output to the boiler feed water control valves. The steam flow to the steam turbine and the running speed of feed water pump is controlled by the differential discharge pressure control. This control compares the feed water discharge pressure with boiler steam drum pressure and adjusts the feed water pump speed by regulating the steam flow across the steam turbine in order to maintain the constant difference across the feed water regulator [3].

Coffin feed water pump, shown in Figure 1, consists of four sections: turbine assembly, bearing housing assembly, governor gear and oil pump assembly and pump assembly. At the one end, the driving steam enters the single stage, impulse, two bucket row, velocity compounded, axial flow turbine and expands through the turbine wheel and then exhausts in an axial direction directly above the turbine end cover. A single shaft, used for driving the two stage centrifugal pump at the other end, passes through the turbine housing, bearing housing, and pump housing. This shaft is supported by two roller bearings located in the bearing casing and the thrust bearing located on the outboard end of the pump housing. The bearing casing additionally serves as an oil reservoir and for providing the power to the horizontal governor drive shaft through a worm and a worm gear. Feed water pump control is accomplished by the pump discharge pressure via constant pressure regulator, through centrifugal speed governor when rated speed has been exceeded and through excess exhaust back pressure trip and low lube oil pressure trip protection [1].

Outer design of one Shinko pump is shown in Figure 2. It consists of feed water pump assembly and steam turbine assembly interconnected by the gear coupling. Feed water pump is also multi stage centrifugal pump (in this case three stages) where the first stage impeller is of the double suction type and the rest are of the single suction type. It has horizontally split casing in order to have easier maintenance. Rotor is supported by plain forced lubricated bearings and tilting pad type thrust bearing. Feed water pump is directly connected to the driving steam turbine but in this case through a forced lubricated gear coupling. Steam turbine is of the horizontal single stage speed compound impulse type. The turbine casing is also horizontally split into two parts. Bearing housings are at the both ends of the turbine shaft. The governor end bearing housing is provided with radial bearing and thrust bearing. The bearing housing near the feed water pump has radial bearing only. For the control of the steam turbine operation the Woodward governor is used. Its speed setting mechanism and the pressure controller unit are interconnected to effect constant discharge pressure control [2].



Source: http://coffinpump.com/wp-content/uploads/2013/07/Turbine-Driven-Boiler-Feed-Pump-Page-2.jpg [4]

Figure 1: Construction details of Coffin feed water pump

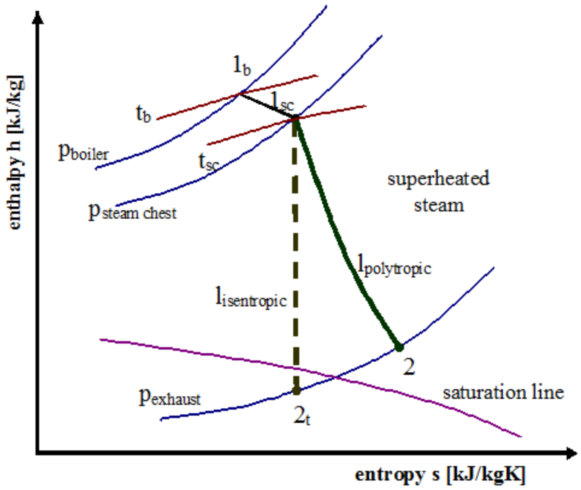
Source: http://sftspb.com/content/21-shinko [5]

Figure 2: Shinko feed water pump

2 STEAM TURBINE FOR FEED WATER PUMPS

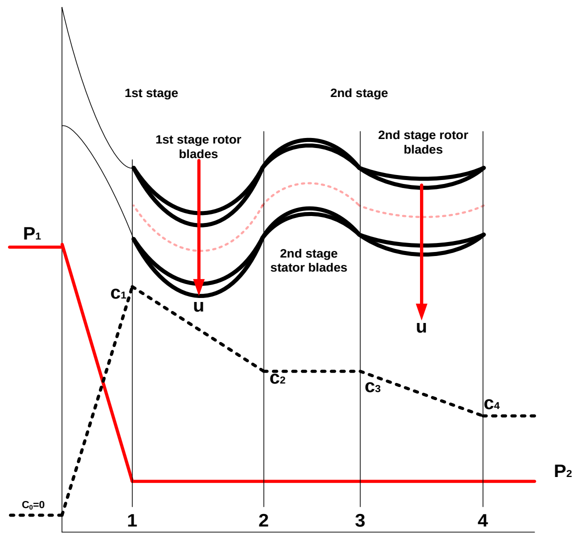
Before steam enters in the steam turbine, the pressure is reduced from boiler pressure to the steam chest pressure. This is due to steam passing over the turbine governing valve. There, the steam pressure and temperature is reduced but the steam specific enthalpy remains constant. After passing the governing valve, steam enters the steam chest from where it flows to the first stage nozzles of the steam turbine and expands to exhaust pressure (Figure 3).

Curtis wheel steam turbine, mostly used for feed water pumps drive, comprises of first row of stationary or stator nozzles, two rows of moving or rotor (bucket) blades and one stator row of guide blades situated between two rotor stages. In the first row of stator nozzles the steam expands from inlet pressure to the exhaust pressure and a kinetic energy of the steam is increased. This expansion should be adiabatic as it gives the greatest possible heat drop but due to friction losses it is polytropic (Figure 3). After the first stator nozzles the steam enters the first row of moving or rotor blades where the kinetic energy and thus the absolute velocity of the steam is reduced. This reduction of kinetic energy and velocity produces a force which acts on the turbine rotor blades and turns the rotor of the steam turbine. From there, steam flows through second stage of stator blades or guide blades where steam is proceeded to the second row of moving or rotor blades. There the kinetic energy and thus the absolute velocity of the steam is again reduced in order to produce a rotational force. Figure 4 shows the behavior of the steam pressure and absolute velocity over the stages in one Curtis wheel steam turbine.



Source: Authors

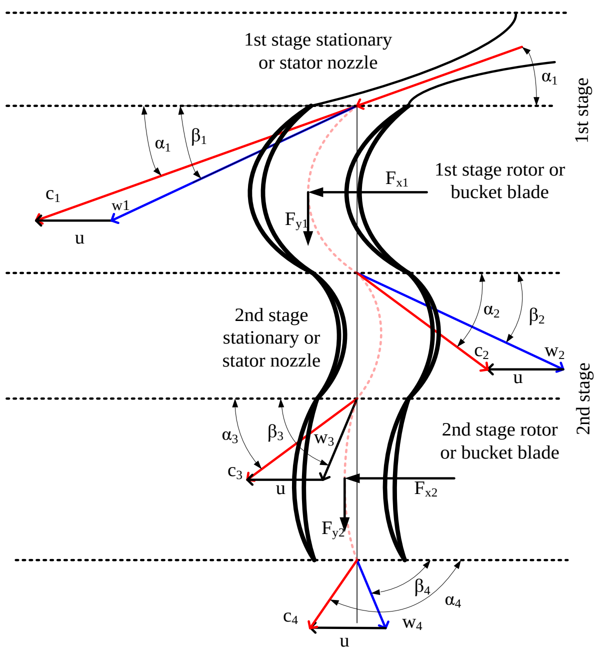
Figure 3: Main feed water steam turbine isentropic and polytropic expansion



Source: Authors

Figure 4: Steam pressure and absolute velocity diagram in Curtis wheel steam turbine

Because of the steam turbine rotor rotation the absolute steam velocity has to be divided into relative steam velocity *w* and rotational velocity *u*. Figure 5 shows diagram of absolute steam velocities *c*, relative steam velocities *w*, rotor rotational velocity *u*, axial forces F_x, and thrust forces F_y over the stages in one Curtis wheel steam turbine.



Source: Authors

Figure 5: Absolute and relative steam velocity diagram at Curtis wheel steam turbine

Each value can be calculated using following equations [6]:

$$c_1 = \varphi \cdot \sqrt{2(h_1 - h_2)} \ \left[\frac{m}{s}\right] \quad (1)$$

$$u = D \cdot \pi \cdot n \quad \left[\frac{m}{s}\right] \quad (2)$$

$$w_1 = \sqrt{u^2 + {c_1}^2 - 2 \cdot u \cdot c_1 \cdot \cos\alpha_1} \ \left[\frac{m}{s}\right] \quad (3)$$

$$w_2 = \psi \cdot w_1 \quad \left[\frac{m}{s}\right] \quad (4)$$

$$c_2 = \sqrt{u^2 + {w_2}^2 - 2 \cdot u \cdot w_2 \cdot \cos\beta_2} \ \left[\frac{m}{s}\right] \quad (5)$$

$$c_3 = \varphi \cdot c_2 \quad \left[\frac{m}{s}\right] \quad (6)$$

$$w_3 = \sqrt{u^2 + {c_3}^2 - 2 \cdot u \cdot c_3 \cdot \cos\alpha_3} \ \left[\frac{m}{s}\right] \quad (7)$$

$$w_4 = \psi \cdot w_3 \quad \left[\frac{m}{s}\right] \quad (8)$$

$$c_4 = \sqrt{u^2 + {w_4}^2 - 2 \cdot u \cdot w_4 \cdot \cos\beta_4} \ \left[\frac{m}{s}\right] \quad (9)$$

$$F_x = F_{x1} + F_{x2} \quad [N] \quad (10)$$

$$F_{x1} = m \cdot (w_1 \cdot \cos\beta_1 + w_2 \cdot \cos\beta_2) \ [N] \quad (11)$$

$$F_{x2} = m \cdot (w_3 \cdot \cos\beta_3 + w_4 \cdot \cos\beta_4) \ [N] \quad (12)$$

$$F_y = F_{y1} + F_{y2} \ [N] \quad (13)$$

$$F_{y1} = m \cdot (w_2 \cdot \sin\beta_2 - w_1 \cdot \sin\beta_1) \ [N] \quad (14)$$

$$F_{y2} = m \cdot (w_4 \cdot \sin\beta_4 - w_3 \cdot \sin\beta_3) \ [N] \quad (15)$$

$$P = F_x \cdot u \quad [W] \quad (16)$$

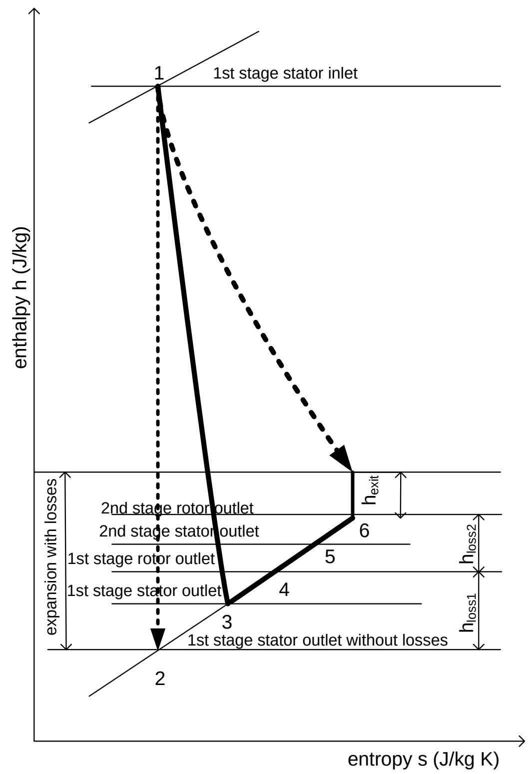
$$\eta_T = \frac{P}{(h_1 - h_2) \cdot m} \quad (17)$$

Where φ is friction loss in stator nozzle, h_1 and h_2 are steam specific enthalpies, D is rotor diameter, n is number of turbine revolutions, ψ is friction loss in rotor blades, m is steam mass flow in kg/h, P is steam turbine power and η_T is steam turbine efficiency.

Angles α and β can be calculated using laws of cosine or sines. In an impulse steam turbine, which Curtis wheel is, the $\beta_1=\beta_2$ and $\beta_3=\beta_4$ because of the same inlet and outlet geometry of the rotor blades [6].

Manufacturing defects, carry-over deposits, corrosion and erosion in stator and rotor blades increase the friction and reduce steam velocity. Some of the kinetic energy converts back into heat energy, re-heating the steam and representing losses in the turbine [7].

Those steam velocity losses are presented as a velocity losses in stator nozzles φ and are from 0.90-0.98 and velocity losses in rotor blades ψ (from 0.70-0.90) [6]. Also, Curtis wheel has high exhaust velocity (which is like a available heat drop) and this represents a considerable loss of energy that is not used in other stages [7].



Source: Authors

Figure 6: Losses in the stator and rotor blades of Curtis wheel steam turbine

Because of the all mentioned losses, steam turbine efficiency decreases. Figure 6 shows how steam velocity reduction and high exhaust velocity influence the reduction of heat drop and thus on steam turbine efficiency.

This velocity losses and consequently heat drop losses can be calculated using following equations [6]:

$$h_{loss1} = \frac{c_{1t}^2}{2} - \frac{c_1^2}{2} + \frac{c_2^2}{2} - \frac{c_3^2}{2} \quad \left[\frac{kJ}{kg}\right] \tag{18}$$

$$c_{1t} = \sqrt{2(h_1 - h_2)} \quad \left[\frac{m}{s}\right] \tag{19}$$

$$h_{loss2} = \frac{w_1^2}{2} - \frac{w_2^2}{2} + \frac{w_3^2}{2} - \frac{w_4^2}{2} \quad \left[\frac{kJ}{kg}\right] \tag{20}$$

$$h_{exh} = \frac{c_4^2}{2} \quad \left[\frac{kJ}{kg}\right]. \tag{21}$$

Other losses in steam turbine like windage, gland and blade tip leakage can be neglected because of its small influence on overall efficiency and in this paper it will not be taken into consideration.

3 STEAM TURBINE ANALYSIS

In this paper the steam turbine for feed water pump is analyzed using the data from the performance test on one LNG ship. During the test, steam flow to the turbine was changed by opening and closing the governing valve. This change affects the turbine revolution, steam chest pressure and exhaust temperature. Available steam turbine data from the performance test is presented in Table 1. Steam specific enthalpies and entropies necessary for the calculation were taken from steam tables [8].

Table 1: Data from the performance test of steam turbine for Shinko feed water pump

Steam inlet pressure bar	Steam inlet temperature °C	Steam chest pressure bar	Exhaust pressure bar °C	Exhaust temperature °C	Steam flow kg/h	Revolution min^{-1}
59.8	372	29	2.8	200	2591	5420
59.8	376	35	2.8	193	2994	5491
59.8	379	43	2.8	189	3510	5583
59.8	382	46	2.8	185	4031	5762
59.8	384	46	2.8	185	4668	5981
59.8	384	49	2.8	185	4893	6115
59.8	383	46	2.8	197	4620	5977

Source: Shinko Ind. Ltd. (2006): Final drawing for Main Feed Pump & Turbine, internal ship documentation. Hiroshima, Japan.

Steam turbine parameters for calculating all other data presented in Table 2 are as follows:

- Rotor diameter D=800 mm,
- Steam inlet angle α_1=18°,
- Stator losses φ=0.94,
- Rotor losses ψ= 0.7-0.79.

Rotor diameter and steam inlet angle used in this analysis are values taken just for the calculation and in reality depend on steam turbine production model or type. Stator and rotor losses, as already mentioned, depend on various factors. In this example value for stator losses was assumed to be 0.94 in order to simplify the calculation. Values for rotor losses were modified from 0.7-0.79 depending on the initial steam and temperature situation in order to obtain certain steam turbine power. In reality, steam turbine power depends on various combinations of stator and rotor losses, rotor diameter and steam inlet angle. Major influence on this analysis have steam inlet and exhaust pressures and temperatures. Rotor diameter and steam inlet angle are fixed for any analyzed turbine and losses depend on the actual status of the steam turbine and will not change end parameters a lot.

Table 2: Calculated data for steam turbine used for driving the Shinko feed water pump

t_1 °C	372	376	379	382	384	384	383
p_{sc} bar	29	35	43	46	46	49	46
t_2 °C	200	193	189	185	185	185	197
m kg/h	2591	2994	3510	4031	4668	4893	4620
min^{-1}	5420	5491	5583	5762	5981	6115	5977
φ	0.94	0.94	0.94	0.94	0.94	0.94	0.94
ψ	0.79	0.77	0.73	0.74	0.75	0.73	0.70
c_1	955	985	1014	1024	1026	1033	1025
u	227	230	234	241	250	256	250
w_1	743	770	795	799	792	794	791
α_1 °	18	18	18	18	18	18	18
β_1 °	23.4	23.3	23.2	23.4	23.6	23.7	23.6
w_2	587	593	579	588	593	579	554
c_2	389	392	375	379	377	360	340
α_2 °	36.8	36.7	37.4	38.0	39.0	40.4	41.0
β_2 °	23.4	23.3	23.2	23.4	23.6	23.7	23.6
c_3	366	369	353	356	355	338	319
w_3	229	230	219	223	225	219	209
α_3 °	36.8	36.7	37.4	38.0	39.0	40.4	41.0
β_3 °	73.2	73.4	77.7	79.8	83.6	89.6	92.2
w_4	181	177	160	164	168	160	146
c_4	246	247	254	267	286	301	295
α_4 °	135	137	142	143	144	148	150
β_4 °	73.2	73.4	77.7	79.8	83.6	89.6	92.2
F_x	964	1138	1310	1502	1703	1713	1564
F_y	78	100	140	158	176	198	202
h_{l1}	69	73	76	78	78	78	76
h_{l2}	114	131	160	158	149	159	171
h_{exh}	30	30	32	36	41	45	43
η_T %	58.8	57.2	53.9	54.4	55.2	53.3	51.2
P kW	219	262	306	362	426	439	391

Source: Authors

As it can be seen from the analysis, steam turbine efficiency depends on many parameters and varies from 51.2 to 58.8%. Enthalpy of the inlet and outlet steam, steam losses in stator and rotor blades, value of exhaust velocity and steam mass flow are one of the crucial parameters that affect the steam turbine efficiency. Steam velocities across the turbine depend on the operational and manufacturing parameters and that affects φ and ψ parameters and stator and rotor enthalpy losses. Steam turbine power varies from 219 up to 439 kW. It depends on steam mass flow, revolution and converted kinetic energy in rotor blades into the axial force for turning the rotor. The higher the value of these parameters is, the higher will be the value of steam turbine power.

Analysis performed in this paper shows an example of how steam turbine can be modeled and presented in a way that end users can easily understand all processes in the turbine and through available data/parameters in the engine room calculate all other specific parameters of the steam turbine.

Every engineer should follow instruction manual recommendation for operation and maintenance of the steam turbine in order to timely recognize any fault or failure of the feed water pump. Additional information through parameter analysis can help in better understanding the processes and recognizing any problem during normal operation of the steam turbine.

4 CONCLUSION

This paper presents types of feed water pumps and driving steam turbine used for delivering the feed water into the main boilers of the LNG ships. Coffin and Shinko feed water pumps are mostly used types. They are centrifugal pumps with two or three stages. Both feed water pumps are driven with Curtis wheel steam turbine which comprises of first row of stationary or stator nozzles, two rows of moving or rotor blades and one stator row of guide blades situated between two rotor stages.

The expansion in the steam turbine stator nozzles is polytropic where increased steam kinetic energy in the stator is converted into force which acts on the turbine rotor blades and turns the rotor of the steam turbine. Steam velocities and losses in the stator and rotor blades directly influence the steam turbine efficiency. Analysis performed using the performance test data provides the end user all necessary data and enables him a better understanding of the processes and to recognize the problem if any exists.

Steam turbine efficiency in analyzed example depends on many parameters and varies from 51.2 to 58.8%. Some of the crucial parameters that affect the steam turbine efficiency are: enthalpy of the inlet and outlet steam, steam losses in stator and rotor blades, value of exhaust velocity and steam mass flow. Steam velocities across the turbine depend on manufacturing defects, carry-over deposits, corrosion and erosion in stator and rotor blades. Other losses that can affect the steam turbine efficiency are windage, gland and blade tip leakage. Steam turbine power in analyzed example varies from 219 up to 439 kW. It depends on steam mass flow, revolution and converted kinetic energy in rotor blades into the axial force for turning the rotor.

REFERENCES

[1] Coffin turbo pump, Inc. (2006). Operation Maintenance Manual for Coffin turbo pump, internal ship documentation. Englewood, USA

[2] Shinko Ind. Ltd. (2006): Final drawing for Main Feed Pump & Turbine, internal ship documentation. Hiroshima, Japan.

[3] LNGC Maersk Qatar (2006). Machinery operation Manual. South Korea: Samsung Heavy Industries

[4] http://coffinpump.com/wp-content/uploads/2013/07/Turbine-Driven-Boiler-Feed-Pump-Page-2.jpg

[5] http://sftspb.com/content/21-shinko

[6] Tireli, E., Martinović, D. (2001) Brodske toplinske turbine. Rijeka: Visoka pomorska škola u Rijeci.

[7] Roy, G. J. (1975). Steam Turbines and Gearing. London: Stanford Maritime Limited.

[8] Ražnjević, K. (1989). Termodinamičke tablice. Zagreb: Narodna tehnika Hrvatske. Sarajevo "Svjetlost".