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Millennium Transfer Span Operating Machinery, a Standard Approach

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HEAVY MOVABLE STRUCTURES, INC.

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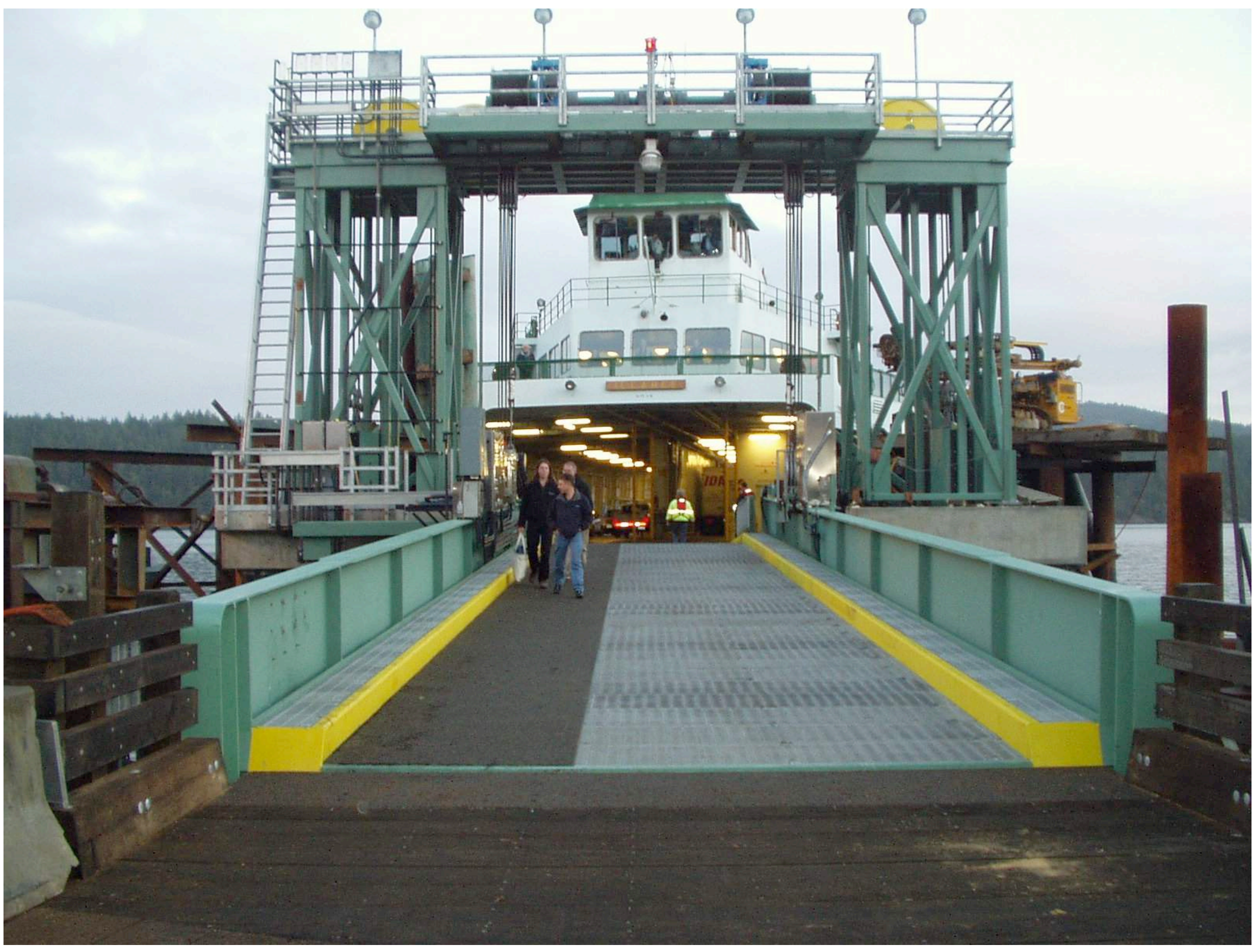
Background

Washington State Ferries

Since its creation on June 1, 1951, Washington State Ferries (WSF) has become the largest ferry system in the United States and the third largest in the world. It operates 29 vessels and 20 ferry terminals throughout Puget Sound, from Pt. Defiance in the south to Sidney, B.C., Canada, in the north. Over 26 million passengers a year ride on Washington State Ferries – more people than travel on Amtrak in a year. By comparison, BC Ferries carries 21 million riders, Staten Island Ferries carries 20 million, and Alaska Marine Highways carries 400,000.



The Ferry arrives to the New Terminal

Description of existing slips

All the movable ramps or "Transfer Span" (TS) systems used to load vessels at each slip are similar and typically consist of a two-lane, movable span 90' in length with an adjustable 15' hydraulically operated apron cantilevered off the end. When loading or unloading vessels, the Transfer Span is simply supported by bridge seat bearings at the end of the trestle at the on-shore end, and at the off-shore side by counterweights and a pair of live load hanger bars, one at each side, whose pins are hydraulically powered, mounted to a load beam located at 80' away from the bridge seat. The counterweight sheaves are supported by structural towers and a headframe. During loading, the apron is simply supported between the end of the Transfer Span and the vessel car deck. After removal of the live load pins, the ramp is operated by a single wire rope hoist system. The wire rope goes from the winch drum located at one side of the transfer span through a sheave block on one side, to another sheave block on the other side and attached to the bridge girder. The hoist system has no redundancy.

This system has been utilized by WSF since the system's creation and has proven to be generally reliable. The counterweighed system has a theoretical 10% nose heavy imbalance, which accounts for a low accelerating force and a large mass providing a low acceleration rate.

This basic system has been incrementally improved over the years by gradually incorporating newer materials and equipment but no fundamental redesign of the system had been undertaken. Recent failures of the counterweight and hoist system led WSF to revisit its basic design. This design exercise led to the development of a totally new standard transfer span, hoist and counterweight system which significantly improves its safety and reliability due primarily to the redundancy of every component.

In one incident, one of the counterweight cables failed causing the hoist to take the additional load. The hoist cable then failed as well and, from impact, the second counterweight failed too, sending the TS and its human operator into the water. The first cable failure was attributed to partially worn and badly corroded inner wires. It had not been inspected regularly because there was no easy means of supporting the counterweight.

Design Requirements and Specifications

Over many years of service WSF has acquired a lot of experience in the way their systems work, their strengths and weaknesses and how to manage them. Some of the original weaknesses have been improved and new experience gained. Some other weaknesses couldn't be improved unless a major redesign was performed and thus affecting other components. The experience gained and the set of pending redesigns necessary to upgrade the system were gathered in the specification for the design of a new transfer span, called the Millennium Transfer Span operating machinery. This new design will be standardized throughout all the WSF terminals.

Prior to issuing the specifications, it was believed that the TS provided limited torsional stiffness, not enough to make the redundancy possible. The required additional stiffness would be provided by cross-reeved counterweights. During the design process, one of the already-existing Transfer spans, similar to the new design, was tested for torsional stiffness by keeping it supported at the bridge seat and lifting the off-shore end from one side only, allowing it to twist under its own weight. The lift force was measured at 150 kips while the vertical displacement of the lifted end was measured at the time the supported end started to lose contact. The deflection was approximately 3 ft. Thus it was confirmed that the TS didn't have enough torsional stiffness to guarantee the safe static dead loads plus the possible live load plus the dynamic overload of losing support on one side. This test reinforced the intuitive feeling that the counterweights should be cross-reeved.

Every machine or every heavy structure has its own character, personality, performance and noises. A completely new transfer span design could have been conceived but the experience gained so far would be rendered useless, and the design would need to have started from scratch. This is why WSF preferred to stay with the old transfer span concept and improve it by making use of the latest technology, available new materials, and components.

At an estimated cost of two million dollars for the new TS, its operating machinery should have a fully redundant heavy duty cable hoist capable of handling the unbalanced dead load only during normal operating conditions, and a fully redundant cross reeved and guided counterweight system leaving an unbalance of 30,000 lb. The live load should be taken by two live-load steel bars with hydraulically operated pins. The hydraulic fluid should be Aviation fluid. It should be possible to rest the counterweights on support beams to free them from the load and allow proper cable inspection.

The Normal Operating Conditions (NOC) are defined as lifting and lowering the TS, operator, snow load and wind load with a useful operating vertical stroke of 20 ft at the tip during an operating time of 180 seconds. In practice, standardization considerations forced the design to provide a lifting range of about 23 feet.

Besides the NOC other load cases had to be considered. Regularly an HS25 low-slung truck loaded with wood chips boards the ferry. Very rarely, during extreme low tide or extreme high tide the truck gets stuck at either end of the TS. The way to free it is to lift the TS, remove the live load pins and lift or lower the TS as needed to free the truck. The truck might be on one side of the two-lane ramp. Reduced speed is acceptable under these conditions.

A single failure consisting in loss of any one cable, loss of one counterweight, a jammed counterweight, jammed TS on one side, loss of any one piece of machinery, including a gear reducer, shaft or sheave block, should not result in harm to persons, vehicles or structures. Additionally, all machinery and structures should be strong enough to withstand the full motors' maximum torque with no damage.

The boat can provide up to 15 kW to operate the system in case of a power outage.

In addition to the operating conditions, the proposed criteria on critical mechanical elements include:

- Factor of safety of 16 for the hoist wire rope, based on normal operating load
- Factor of safety of 6 for the counterweight wire rope
- 2 wire ropes per counterweight
- No live load during raising or lowering – only the operator (note: personnel lifting)
- Nine parts of rope per drum, with one full wrap left on drum at furthest travel
- Motor brake required
- The cable is allowed to stack in multiple layers on a smooth drum
- Means to prevent slack rope
- Limit switches to shut off hoist and set brake
- CW sheave size 30 times rope diameter, hoist sheave size of 18 times rope diameter, hoist drum size 19 times rope diameter
- Live Load Pins engagement verification
- Compliance with Americans with Disabilities Act (ADA)

Also, the practicality of using safety brakes of the type of a tapered sleeve clamping on a rod instead of a disk brake should be investigated.

Design

The hoisting machinery is located on the headframe and the arrangement consists of two electric motors, one of them with an integral disk brake driving a parallel shaft gear reducer. The reducer drives two floating shafts, one for each wire rope drum, directly coupled to their shafts. Each drum has a built-in disk for static parking and emergency brakes. The cable from each drum goes to a sheave block, attached at each side of the TS to a load beam. The counterweight cables are attached to the load beam, too.

Table 1, Vehicle Transfer Span Force Analysis

VEHICLE TRANSFER SPAN FORCE ANALYSIS Description		Normal Op. (no snow)	Normal Op. (with snow)	HS-25 Vehicle LH	HS-25 Vehicle RH
Dead load, Transfer Span weight, ea side, [lbf]	DL	15000.00	27500.00	15000.00	15000.00
Non-Normal operating loads, HS-25, Cw't, Jamming [lbf]	NL	0.00	0.00	45000.00	15000.00
Load on one side of Transfer Span [lbf]	L=DL+NL	15000.00	27500.00	60000.00	30000.00
Counterweight weight [lbf]	CW	67000.00	67000.00	67000.00	67000.00
Counterweight rope efficiency (0.965 per Rasmussen)	h	0.97	0.97	0.97	0.97
No. of active sheaves on Cwt.	NW	1.50	1.50	1.50	1.50
Friction on Cwt. Ropes [lbf]	WF=CW*(1/h^NW)-1)	3677.93	3677.93	3677.93	3677.93
No. of active sheaves on hoist block	NH	9.00	9.00	9.00	9.00
Friction on Block ropes [lbf]	BF=(L+WF)*((1/h)^NH-1)	7060.55	11785.75	24071.26	12730.79
Rope pull on Drum [lbf]	P=(L+WF+BF)/NH	2859.83	4773.74	9749.91	5156.52
Rope 1" dia IWRC, XIP breaking strength [lbf]	RS	103400.00	103400.00	103400.00	103400.00
Rope's Safety Factor	Rn=RS/P	36.16	21.66	10.61	20.05
Drum diameter [in]	D	48.00	48.00	48.00	48.00
Torque on Drum [in-lbf]	DT=P*D/2	68635.96	114569.82	233997.85	123756.59
Holding brakes Service Factor	BSF	2.00	2.00	2.00	2.00
Required brakes, two at each drum, capacity each: [N-m] (use USB 3-III Ed30 1/6 on 44.5" avg. dia.)	BC=BSF*DT*0.113/2	7755.86	12946.39	26441.76	13984.49
Required reducer output torque [in-lbf]	OT=LHDT+RHDT	137271.92	229139.63	357754.43	
Falk 425 Reducer output torque rating [in-lbf]	GT	385892.00	385892.00	385892.00	
Reducer Service Factor	GSF=GT/OT	2.81	1.68	1.08	
Reducer ratio	r	187.10	187.10	187.10	
Reducer efficiency	Gh	0.922	0.922	0.922	
Input torque to reducer [in-lbf]	IT=OT/r/Gh	795.75	1328.30	2073.86	
Motor rpm, (estimated slip), rpm	rpm	874.00	852.00	822.00	
Total required horsepower [hp]	HP=IT*rpm/63024	11.04	17.96	27.05	

Table 1, Continued

Description		Counterweight Loss LH	Counterweight Loss RH	One Side Jammed LH	One Side Jammed RH
Dead load, Transfer Span weight, ea side, [lbf]	DL	15000.00	15000.00	15000.00	15000.00
Non-Normal operating loads, HS-25, Cw't, Jamming [lbf]	NL	67000.00	0.00	151683.73	0.00
Load on one side of Transfer Span [lbf]	L=DL+NL	82000.00	15000.00	166683.73	15000.00
Counterweight weight [lbf]	CW	0.00	67000.00	67000.00	67000.00
Counterweight rope efficiency (0.965 per Rasmussen)	h	0.97	0.97	0.97	0.97
No. of active sheaves on Cwt.	NW	1.50	1.50	1.50	1.50
Friction on Cwt. Ropes [lbf]	WF=CW*(1/h^NW)-1)	0.00	3677.93	0.00	3677.93
No. of active sheaves on hoist block	NH	9.00	9.00	9.00	9.00
Friction on Block ropes [lbf]	BF=(L+WF)*((1/h)^NH-1)	30997.29	7060.55	63009.07	7060.55
Rope pull on Drum [lbf]	P=(L+WF+BF)/NH	12555.25	2859.83	25521.42	2859.83
Rope 1" dia IWRC, XIP breaking strength [lbf]	RS	103400.00	103400.00	103400.00	103400.00
Rope's Safety Factor	Rn=RS/P	8.24	36.16	4.05	36.16
Drum diameter [in]	D	48.00	48.00	48.00	48.00
Torque on Drum [in-lbf]	DT=P*D/2	301326.11	68635.96	612514.15	68635.96
Holding brakes Service Factor	BSF	2.00	2.00	2.00	2.00
Required brakes, two at each drum, capacity each: [N-m] (use USB 3-III Ed30 1/6 on 44.5" avg. dia.)	BC=BSF*DT*0.113/2	34049.85	7755.86	69214.10	7755.86
Required reducer output torque [in-lbf]	OT=LHDT+RHDT	369962.07		681150.10	
Falk 425 Reducer output torque rating [in-lbf]	GT	385892.00		385892.00	
Reducer Service Factor	GSF=GT/OT	1.04		0.57	
Reducer ratio	r	187.10		187.10	
Reducer efficiency	Gh	0.922		0.922	
Input torque to reducer [in-lbf]	IT=OT/r/Gh	2144.63		3948.55	
Motor rpm, (estimated slip), rpm	rpm	819.00		830.00	
Total required horsepower [hp]	HP=IT*rpm/63024	27.87		52.00	

After a few iterations the design parameters closed the loop both statically and dynamically. The static results are shown above in Table 1: Vehicle Transfer Span Force Analysis, please refer to it as needed to illustrate the following discussion of each component features. The dynamic results have been omitted.

Electric Motors

The main electric motor is a Reuland Electric 15 hp, 8 poles, 460 Volt, 3 ph, 60 Hz, single speed, design NEMA D, 5-8% slip, TENV, 60 minute duty, class H insulation, winding thermostat, foot mounted, single end shaft, frame 364T, condensation drains, marine duty, with integrally mounted 230 lbf-ft Stearns brake 40 degree C ambient. The second electric motor is similar to the first one except 5 hp, frame 256T, no brake.

The power required to lift the TS under every day's operation is only 11 hp, thus a 15 hp rated motor is sufficient. The 15 kVA available from the boat is enough to drive the motor and the other electric loads consisting in emergency lighting and controls. Under snow conditions the required lifting power is 18 hp. A second 5 hp motor can provide the additional needed power. The arrangement is transparent to the operator: a current sensor notifies the PLC of an overload and it starts the 5 hp motor automatically in case it is needed. One 20 hp motor would have been adequate, but due to a low power factor the current requirements exceeded the available KVA from the boat, making it difficult to make it work transparently to the operator. The load sharing between the motors was predicted and found acceptable in the worst case of slip discrepancy.

The nominal low speed of the motors together with a reduced starting torque allows a reduced contribution of inertia forces to the machinery in case a counterweight or the TS jams. During an oversize wood chip truck emergency scenario described earlier the required power is 27 hp during over a maximum 10 seconds; both motors can provide the required lifting power without damage. As the 15 hp motor will work under any circumstance, it is the one that is provided with a disk brake that is hard wired directly to the motor circuit with no intermediate relays or any other device that might slow down its operation. This brake works as a service brake.

Gear Reducer

The gear reducer is a quadruple reduction, parallel shaft, with two input and two output shafts with a nominal capacity of 385892 lbf-in input torque, 28.8 hp, 187.1:1 reduction ratio, Falk type A, size 425. The output shaft can share the full output rated torque in any combination of loads for both sides. An oil pan under the reducer with full gear box oil capacity and a cover on both is provided to collect the oil in case of a leak and avoid water contamination.

Although the elevator code is not fully applicable to our case, the man-rating of the reducer and other hoist components has been met by providing the parking brakes, attached directly to the drums, which can work in case of emergency, tripped by an over speed detector through the PLC.

For a nominal input torque from both motors the output torque would be 270,000 in-lbf. The governing factor in the selection of the reducer is the torque required for the chip truck load case, 357754 in-lbf for a service factor of 1.08, while lifting. This is an infrequent load case, not emergency. The emergency case of a lost counterweight would need 369,962 in-lbf for a service factor of 1.04. In case one side jams, the total required torque flowing from the motors through the reducer would be 681150 in-lbf, giving a service factor of 0.57, once the machinery comes to a stop. This is above the 0.5 service factor allowed for static overloads.

Couplings and Floating Shafts

The motor couplings are Falk grid type for the motors and gear type Falk 1050G51 for the floating shafts. The floating shafts are made of AISI 1026 ASTM A519 seamless pipe with solid bars at the ends with gear couplings in FRRF arrangement.

Wire Rope Drums

Although the specifications allow the cable to stack in multiple layers on the drum, there are reasons not to do it. First and most important is that the best way to support the cable is to provide an adequate seat for it. The lower layers must have a uniform pattern that provides good support to the upper ones. For the first layer to have a uniform spread the drum must be grooved, otherwise as more layers stack on top, the irregularities increase and provide less and less support to the upper layers to bear the load. The grooving pattern of a multiple layer drum is complicated to manufacture, especially at the ends where the cable must go through adequate ramps from a lower layer to an upper one. The cable life is greatly reduced if a good support is not provided.

Although the necessary wire rope length for the specified lift would require only 14.3 wraps, a total of 21 were provided including 3 full extra at the attaching clips, 14 for lifting, 3 due to standardization spread and one at the full side. Standardization is discussed in greater detail later.

The fleet angle analysis led us to design a 48 in diameter instead of a smaller one; the specifications allow a ratio of 19:1. Each drum has been designed to take the full unbalanced load from the TS, to allow one cable to break while the TS is supported by the other. Bronze bushings are used at both sides of the drum; no antifriction bearings. A full oil pan is provided under each drum to collect the cable lubricant. A brake disk has been attached to each wire rope drum, not to the shaft. Drum's material is A36 and no hardening was specified for this application to protect the wire rope. The shaft material is AISI 4340, quenched and

Reducer to Drum Floating shaft being aligned during Off-site Construction tempered to 302 Brinell for 120 ksi tensile and 90 ksi yield, 15% minimum elongation. Drums are left and right hand grooving, with flanges at the sides to protect against rope falling off. The shell has the required thickness with no need for internal reinforcements to avoid fatigue cracking.

Parking and Emergency Brakes

There are four brakes, two at each drum: Hindon USB-3-III with disk scraper and position limit switches with Hindon Ed 301-6 thrustor. They operate on a 565 mm radius on a 1250 mm outer diameter by 40 mm thick disk of ASTM A572 Gr. 50 steel. The nominal maximum torque with _=0.35 is 281,900 in-lbf.

The only way to guarantee that the TS will not fall during a machinery failure is to provide braking means at the drums. The brakes are intended to work as parking brakes, with a delay after the current to the motors has been cut. In an emergency these brakes can stop the TS from falling in case one of the hoist's mechanical components fails. A total of four brakes, two on each drum, both drums on the same shaft, have been installed, with a braking torque capacity of 127,400 N-m (1,127,600 lbf-in). The static load on the shaft is 137,272 lbf-in at normal operation, 357,754 lbf-in during an HS25 vehicle lifting and 369,962 if a counterweight is lost.

The brakes' capacity is enough to stop the TS in case of a failure and it has accelerated to a speed higher than the operating speed. The counterweighted system with about 15% imbalance demonstrates its advantage in this failure mode. It has been calculated that in the case of an HS25 vehicle loading case for one foot of free fall at the load beam radius the braking system needs 0.345 ft of decelerating distance to stop the TS. This behavior is representative of all four brakes working at the drums' shaft, even in the case of a broken hoist cable. In case of a broken mechanical component that leaves two brakes out, and with an HS25 vehicle on the TS, 2.23 ft of decelerating distance would be necessary to stop the TS for one foot of free fall. It takes 0.313 seconds to fall one foot under this loading, time for the control system to react to an over speed. These brakes are supposed to work only as parking brakes, setting once the machinery has come to a stop; this is every day's operating mode. But they are to be used during an emergency stop, too. Above mentioned free fall distance and its stopping distance were checked and confirmed after the TS completion, during tests and after fine tuning at the manufacturer's plant. The HS25 vehicle was simulated with test weights.

Hoist Wire Rope

A 1.00 in diameter, IWRC, extra improved plow steel class 6x37 right regular lay wire rope with a nominal braking strength of 51.7 ton was selected.

Special attention was given to the cable life issues including the 48:1 drum to rope diameter ratio.

Sheave Blocks

60 ton capacity with quadruple 24 in sheaves, McKissick block was selected for both upper and lower blocks for a 9 part rope hoist system. They are attached to the side girders.

Live Load Bar Assemblies

Once the TS has been lifted and positioned as needed according to the tide and the vessel height, it is supported by two live load hanger bars, one at each side, with oblong holes, hanging from the overhead bridge. Each hydraulically operated pin is attached to a live load hanger assembly that keeps the bars properly aligned with the pins by rollers. The hangers are attached to the load beam under the TS.

The pin assemblies have position sensors to assure that the pins have been totally removed before the TS is moved or totally inserted before live load is allowed on the TS. In addition the hanger bars have redundant limit switches to detect any uplift movement of the bars if the pins have not been removed, to avoid buckling of the bars when lifting the TS.

Counterweights

Two counterweights are used at the TS, one at each side. They consist of a steel box to support the main lead weights and additional lead weights to adjust the unbalance. The nominal weight is 67,000 lbf each to provide a total unbalance of 30,000 lbf. They are supported by two cross reeved wire ropes of 1.75 in diameter. The counterweights are guided inside the towers by angle rails attached to the towers and polyurethane pads to rub against them. Lead was chosen for the counterweights because of the long lift height and the limited height of the towers due to architectural requirements.

The counterweights perform more than one function. First, they compensate for most of the TS dead weight, reducing the hoist capacity. Second, they provide a controlled imbalance to assure that the TS will come down when the hoist pays out cable, overcoming the friction in each of the nine-part hoist rope reeving and of the counterweights cables themselves. It should be noted that the total friction under normal operating conditions is about 21.5 kips total. Third, by cross reeving, hanging them from two wire ropes, each one attached to opposite sides of the load beam, the counterweights add torsional stiffness to the TS in case that one side of the TS loses support due to a failure. Fourth, they add mass and reduce the accelerating torque in case of a total hoist failure, as in the case of a gear failure, reducing significantly the vertical acceleration. This increases the available time to detect an over speed and react by applying the parking brakes.

Other Components

Other mechanical components are conventional and do not need any comments.

Programmable Logic Controller

Once a decision has been made to install a Programmable Logic Controller (PLC), it takes charge of as much functions as possible. In our case, the PLC is in charge of the general Transfer Span sequence of operations, safeties, interlocks and other functions. Among them is the electric motors management verifying the load and starting the second motor, if needed.

A load cell has been located at the end of each load wire rope, at the sheave blocks. Its function is to detect a no load (slack) as well as an overload condition in the cable. It also takes care of over speed safeties, of the many functions related to obtaining the power from the vessel, of the management of the Live Load Pin switches and their operation and many more not mentioned here as the control system is not the subject of this paper. All these control operations are transparent to the operator.

Standardization

Standardization is the first idea that comes to mind when dealing with more than a few similar or identical installations as in our subject. It has many pros and apparently few cons. A ferry terminal presents particular problems.

Although the Transfer Span and its machinery have been standardized and are supposed to be identical, they should be adapted to sites with differing geographical, architectural, environmental and often political needs from one county to another. The dividing line between the standard and the custom-tailored must be set and conflicts can arise. In our case some dimensions had to be adjustable to allow the standard design to be used throughout. This is true in the case of the relationship between the towers heights, the trestle elevations, the high and low tides at particular terminals around the Puget Sound, the different ferry vessel deck heights and how these factors combine with the hoist and counterweight cables and with the live load bars lengths. The first attempt to standardization made at Port Townsend terminal presented some problems and the solution required changing some parts in the standard design (such as the load bars and other parts), to comply with the particular needs.

Transfer Span and Lift System during Off-site Construction

If standardization is to prevail, no improvements can be made to a single terminal or even a few terminals.

Another aspect of the standardization is how long it will be active. The specified life of our project's machinery is 40 years. Very few components defeat obsolescence in more than few years nowadays. It's hard to imagine that most components will still be available 20 years from now. In other words, the standardization is highly desirable but very difficult if not impossible to achieve.

Modular Construction

The entire hoist machinery is mounted on a headframe, allowing the machinery to be assembled with counterweights and towers, aligned and tested in the shop and then shipped to the jobsite

When the idea of a standard transfer span that could be used in all the terminals around the Puget Sound took shape, additional thoughts led WSF to make minor changes to the basic concept and adapt the design to allow the transfer span to be assembled, aligned, tuned and tested at the manufacturer's plant and then transported in barges to their final location. The advantage of carrying out the assembly, adjustments and fine tuning in the manufacturer's shop is indisputable. The time and cost of erection at the jobsite is greatly reduced, reducing the total lead time and overall cost. This was actually done in the first two terminals: Friday Harbor and Shaw.

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