Primary Chilled Water Loop Retrofit

By **Lucas B. Hyman, P.E.,** Member ASHRAE and **Fred R. Bockmiller, P.E.** Member ASHRAE

The University of California at Irvine (UCI) near Los Angeles has a mild Mediterranean climate, with a summer design temperature of 90°F (32°C) and winter design temperature of 38°F (3°C). The 1,500 acre (607 ha) campus has approximately 6.5 million ft^2 (604 000 m^2) of developed facilities (3.5 million ft^2 [325 000 m^2] is served chilled water from the central plant). Many of the campus facility buildings are 100% outside-air science buildings.

The campus central plant produces chilled water, high-temperature hot-water (HTHW), and compressed air distributed via piping mains to the campus buildings. Piping mains to the buildings are routed primarily in underground walk-through utility tunnels although some mains are direct buried. Chilled water is used for space conditioning, lab process, and reactor cooling. HTHW is used for space heating, domestic hot water (DHW) production, industrial hot water (IHW) production, swimming pool heat, and local building steam generation.

The central plant operates continuously with only occasional (once or twice a year) brief maintenance shutdowns. Cooling is required year-round. Shutdowns never last more than a day or two.

Four, natural gas-fired boilers produce 225-psig (1653 kPa) saturated steam for use in the steam-to-HTHW heat exchangers and a steam-turbine centrifugal chiller.

The central chiller plant consists of five large centrifugal chillers, concrete cooling towers, and a stratified chilled water thermal energy storage (TES) tank. Four of the chillers are electric-driven and one chiller is steam-turbine driven.

Chilled water is pumped with a primary-secondary pumping arrangement. The primary loop pumps circulate chilled water through the chillers, and the secondary pumps distribute chilled water to the campus.

The original pumped primary-only chilled water pumps circulated chilled water through the chillers and to the campus buildings via the campus chilled water distribution system. This required the pumps to have relatively high heads and large electric motors.

UCI expanded its central plant in a series of steps over the last 35 years. The Step 3 central plant expansion added chillers, additional cooling towers, and a TES tank. This expansion project converted the central plant's chilled water system from a pumped primary-only system to a primary-secondary arrangement. With this modification, the original pumps became the primary loop chilled water pumps. The expansion also added new secondary pumps to distribute chilled water to the campus. Because the original pumps no longer were required to distribute chilled water to the campus, the pump head requirements were lowered significantly. The Step 3 expansion did not, however, modify or replace those chilled water pumps with properly sized pumps.

Background

The original primary-only pumping system at the central plant used two pumps (75 hp [56 kW] each) dedicated to the two original 750-ton (2638 kW) steam-turbine chillers. When the third steam-turbine chiller (1,750 tons [6155 kW]) was added in 1969, the system was modified so that the chilled water pumps discharged to a common header. The 1969 retrofit added two 150-hp (112 kW) chilled water pumps. With the common header, any of the chilled water pumps could serve any of the chillers.

In 1989, UCI added two 800-ton (2814 kW) electric drive chillers and an additional chilled water pump to the central plant. The new pump also discharged to the common supply header.

In 1992, UCI demolished one 750-ton (2638 kW) steam turbine chiller and added a 1,000-ton (3517 kW) electric chiller, an ice-harvesting TES system, and two additional chilled water pumps (125 hp and 150 hp [93 kW and 112 kW]). These chilled water pumps also discharged to the common header.

In 1996, as part of the Step 3 central plant expansion, UCI installed two 1,000-ton (3517 kW) chillers in place of the second 750-ton (2638 kW) steam turbine chiller, and a 4.5-million gallon (17 million liter) stratified chilled water TES tank. Step 3 did not add new primary pumps, and the central plant chilled water system was converted to a primary-secondary system.

About the Authors

Lucas B. Hyman, P.E., is vice president of Goss Engineering, Corona, Calif.

Fred R. Bockmiller, P.E., is the energy engineer at UC Irvine, Irvine, Calif.

Advertisement for the print edition formerly in this space.

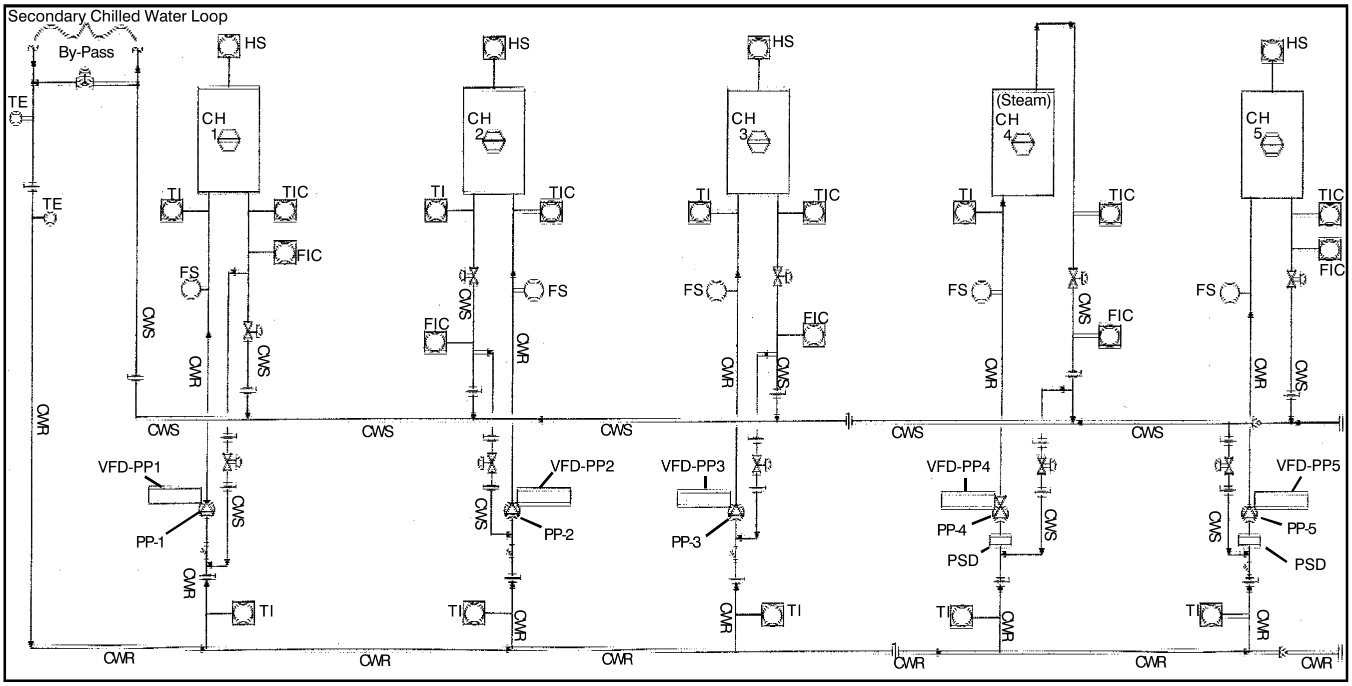

Figure 1: University of California at Irvine's central plant primary loop schematic.

This change added six variable frequency drive 125 hp (93 kW) secondary chilled water pumps to circulate chilled water to the campus. The existing chilled water pumps became (without any modifications) the primary loop pumps.

In 1997, UCI purchased and installed a 2,500 ton (8793 kW) electric chiller in the bay that previously had housed the two 800-ton (2814 kW) chillers, which had performed poorly at the 39°F (3.8°C) evaporator leaving water temperature required for the new TES system. The new 2,500 ton (8793 kW) chiller provides more capacity in the same footprint, and performs better at the required lower chilled water supply temperature.

The new 2,500-ton (8793 kW) chiller had a different set of design conditions. The new chiller was designed to operate at 16°F (8.8°C) ΔT at 3,750 gpm (237 L/s) (55°F [13°C] CWR and 39°F [3.8°C] CWS). The chiller, however, also would operate acceptably at 5,000 gpm (315 L/s) and at 12°F (6.6°C) ΔT. This was valuable as the campus often suffered from lower than design ΔT. This problem plagues many central chiller plants, particularly during low cooling load periods. Low ΔT is a particular problem with thermal storage systems, because it reduces the amount of cooling that can be stored, and reduces the chiller capacity when lower than design return chilled water is returned to the chillers.

Equipment Tag	Flow (gpm)	Design Head (ft)	Design Water hp	Motor hp
P–1	3,000	170	129	150
P–2	3,500	170	150	150
P–3	3,500	170	150	150
P–4	4,000	170	172	200
P–5	2,000	185	93	125
Total	16,000			775

Table 1: Primary chilled water pumps design conditions prior to retrofit.

Summer ΔT at the plant has reached 18°F (9.9°C). However, winter night ΔTs can be as low as 8°F (4.4°C). Low winter ΔTs are due in part to the many 100% outside-air science buildings. This retrofit project allowed for ΔTs in the range of 8°F to 20°F (4.4°C to 11°C).

Table 1 summarizes the design conditions of the primary chilled water pumps following the Step 3 expansion of the central plant.

Some of the electric motors of the pumps were too small for the primary chilled water pumps and easily overloaded. Therefore, UCI facilities staff operated the pumps with severely throttled discharge valves.

When Step 3 made the primary-secondary modifications, the required head for these pumps was significantly reduced. The pumps weren't replaced, the impellers weren't trimmed, and the motors would overload. Therefore, the reduced head requirements were accommodated by adding head by throttling balancing valves on the pump discharges. This resulted in the pumps operating under the same conditions and the same high horsepower requirements as before the primary-secondary system was installed.

Of course, the data in *Table 1* is the design point and only represents one point on a pump curve. Any pump operating at constant speed will operate at a lower head point than design, but will operate with higher flow by simply riding down on the pump curve to meet the system curve. This offered a potential solution of simply operating fewer pumps than were used before. This caused the pumps to overload and unfortunately, the frames could not accommodate larger electric motors.

If the pump motors were not undersized, the needed pump performance could possibly have been met by allowing the pump(s) to ride down the pump curve. The lower head/higher flow point is at a less efficient point on the pump's curve, so greater savings can be obtained by a pump that produces the proper flow and head at its maximum efficiency point.

In addition, far more net-positive suction head is required at high flow and low head of a pump selected for lower flow and higher head performance. This often causes pump manufacturers not to show or guarantee operation at the lower part of the pump curve.

Another challenge of the existing system was the requirement for manual selection of primary chilled water pumps. The plant operator had to mix and match pumps to serve the various combinations of chillers required to meet the load from the campus and to charge the TES tank. To improve the reliability of the system, most operators tended to run more pumps than required to provide the necessary flow.

Chiller Tag (Refrigerant)	Capacity (tons)	Design Flow (gpm)	ΔT (°F)	Min CHWS (°F)	Head Loss (ft H_2O)
No. 1 (R-11)	1,000	2,000	12	42	11
No. 2 (R-123)	1,000	2,000	12	39	9
No. 3 (R-123)	1,000	2,000	12	39	9
No. 4 (R-12)	1,750	2,730	16	40	24
No. 5 (R-123)	2,500	3,750	16	36	11

Table 2: Chiller evaporator design conditions.

Equip Tag	GPM	Head Loss	Pump Efficiency	BHP	EHP	kW
PP-1	1,800	28	84%	15.2	17.8	13.3
PP-2	2,000	33	84%	19.8	23.3	17.4
PP-3	2,000	25	84%	15.0	17.7	13.2
PP-4	2,730	57	86%	45.7	53.8	40.1
PP-5	3,500	23	87%	23.4	27.5	20.5
TOTAL	13,530			132	155	116

Table 3: Installed individual chiller pumps.

Another important issue was that the chillers had different design ΔTs, different evaporator flow rates, different evaporator pressure drops, and different minimum allowable evaporator leaving chilled water temperatures. *Table 2* shows the chiller evaporator design conditions at the beginning of the retrofit project.

Given these problems and the obvious oversize of the primary loop chilled water pumps, an opportunity existed to reduce energy consumption by reducing the power consumption of the pumps. While trimming the pump impellers could reduce the head produced by the pumps at design flows, that solution would not provide maximum energy savings nor address the other problems discussed earlier. An ideal solution also would improve operation and control. Because of these and other factors, neither the solution of trimming impellers nor operating fewer pumps was selected.

A variable-primary-flow (VPF) system was not considered because the secondary distribution pumps were already installed (therefore, no capital cost savings could be gained) and because the older central plant chillers (and their controls) could have trouble with variable evaporator flow.

System Design Issues

Reliability and durability of the pumps was an important issue as there was not enough space to install redundant pumps for each chiller. The design criteria established a minimum 30-year life. The following were also important issues:

1. Accomodating the different chiller evaporator parameters in *Table 2*.
2. Accomodating for varying campus ΔTs between summer and winter periods and still keep the chillers fully loaded.
3. Achieving the minimum chilled water supply temperature leaving each chiller.
4. Accomodating varying primary loop differential-pressures and accommodating potential increased primary loop differential-pressures as the central plant is expanded.
5. Accomodating varying flow rates through the chillers.
6. Keeping the chilled plant operating in both TES and non-TES modes during construction.
7. Making the project self-funding from energy savings with the projected saving resulting in no more than a seven-year payback.
8. Fitting into the limited space available and the piping constraints of the existing plant installation.

The Solution

Based on discussions with UCI staff, the authors determined that providing a single chilled water pump for each chiller would provide for operational simplicity and was the most desirable solution based on the earlier discussion. Pumps on a common header would need pump heads high enough for the chiller with the highest pressure drop (approximately 2.7 times the minimum chiller pressure drop). Spacing and cost constraints only allowed for a single pump per chiller. UCI Facilities keeps complete spare rotating-assemblies in stock so the staff can quickly repair a pump that fails. The design incorporated variable frequency drives to maintain a constant selected flow rate regardless of changing header differential pressures, and to accommodate the need for different chiller evaporator flow rates at different times of the year.

To accommodate as wide a variation in chilled water return temperatures as possible and still maintain the minimum allowable leaving chilled water temperature, the retrofit installed an automatic modulating bypass control valve at each chiller. The bypass maintains the leaving chilled water supply temperature by returning part of the chilled water supply to the chiller inlet. For example, if the CWR temperature to chiller number No. 2 is 55°F (13°C), normally, at the maximum ΔT, the CWS temperature would be 43°F (6°C). To obtain the desired 39°F (3.8°C) CWS, some CWS is recirculated to the chiller inlet for a blended chiller entering water temperature of 51°F (10.5°C). *Figure 1* provides a schematic diagram of the designed system. *Table 3* provides the design operating conditions of the new primary chilled water pumps.

Pump No.	Estimated Hours	Measured Power (kW)	Total Energy (kWh)	Total Cost
P-1	3,500	90	315,000	$18,900
P-2	3,000	112	336,000	$20,160
P-3	3,000	112	336,000	$20,160
P-4	3,500	90	315,000	$18,900
P-5	2,500	90	225,000	$13,500
TOTALS	15,500		1,527,000	$91,620

Table 4: Annual pump energy use before retrofit.

Pump No.	Measured Hours	Avg Power (kW)	Measured Energy (kWh)	Total Cost
PP-1	3,063	8.6	26,342	$1,581
PP-2	3,507	12	42,084	$2,525
PP-3	3,290	13.2	43,428	$2,606
PP-4	206	42.8	8,817	$529
PP-5	4,144	29.8	123,491	$7,409
TOTALS	14,210		244,162	$14,650

Table 5: Annual pump energy after retrofit.

Because energy savings were an essential factor behind the decision to use new pumps, UCI maximized design point pump and motor efficiencies. UCI specified factory performance testing and individual calibrated pump curves.

UCI selected double-suction horizontal split-case pumps as the basis for design. With the existing pipe geometry, this style of pump allowed for the simplest installation with the minimum number of elbows.

Automatic two-position control valves, linked to the plant's digital control system, allowed for automatic start-up of the dedicated pumps with each of the chillers.

Construction occurred during the winter months and UCI staged the work to allow half of the plant to remain on-line during the retrofit project.

The university pre-purchased the pumps, two-position control valves, modulating bypass control valves, and temperature transmitters. This allowed maximum control over the selection of the equipment. The university was able to acquire all of the equipment before releasing a contract for construction. This eliminated any construction delays due to equipment delivery.

Performance and Energy Savings

The pumps are operating successfully and the measured power usage is less than or approximately equal to that predicted with the exception of chiller No. 5. This is likely because the chilled water pump often is operated at 5,000 gpm (315 L/s) versus 3,500 gpm (221 L/s).

Before the retrofit work, UCI measured the actual average power kilowatt draw of each of the existing throttled primary chilled water pumps. The annual pumping energy cost for the existing throttled pumps was estimated by multiplying the average measured electric power by the estimated number of annual operating hours by UCI's calculated annual average cost of electricity ($0.06 per kWh based on SCE's TOU-8 rate schedule). The estimated energy cost for throttled primary chilled water pump operation was approximately $91,600 per year.

One year after the completion of the project, UCI read the operating hours and kilowatt-hours from each of the pump's VFDs. The average kilowatt draw and the estimated cost of operation was calculated as approximately $14,600. The estimated first-year savings is approximately $77,000. *Table 5* and *Table 6* provide these findings.

Conclusion

This system has provided UCI not only energy savings and a payback of approximately six years, but also the ability to better control chiller operation. Now, regardless of the campus chilled water return temperature and varying header differential pressures, the central plant operators have the ability to:

- Keep the chillers fully loaded.
- Maximize each chiller's ΔT by setting the minimum evaporator flow rate.
- Provide the minimum allowable leaving chilled water supply temperature.
- Easily operate the chilled water plant without complex decisions of how many pumps to run.