Energy Savings From Pump Impeller Trimming

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A recent study assessed the extent of pump oversizing in commercial office building heating and cooling systems. The study was used to estimate the potential energy savings made possible by trimming the pump's impeller to match the actual system head requirements. The objective of estimating the potential energy savings was to provide both design engineers and building owners with data on the benefits of pump impeller trimming, thereby demonstrating the feasibility of this approach as a means of improving building operating efficiency.

The study was based on data collected from 14 large commercial office buildings in the city of Madison, Wis. In each building, measurements were taken to determine the extent of pump oversizing. This information was used to estimate the cost and energy savings associated with pump impeller trimming.

The results of the study from these buildings show that pumps in large commercial office buildings are oversized and that substantial savings can be realized if the pump impeller is trimmed to the proper size. The average electrical demand savings for each building during the summer months was approximately 6 kW, while the average annual energy savings were close to 28,000 kWh. The resulting annual cost savings came to $1,220, and with an average trimming cost of $1,540 per building, the simple payback was 1.3 years.

In addition to the short payback, pump impeller trimming is an attractive energy conservation opportunity because it is easy to implement, and the work can be carried out without disruption to the building's normal operation. In the study, most systems had 100% standby capability, hence one pump could continue operating while the other is modified.

Pumps in Heating and Cooling Applications

Heating systems are generally of the variable flow type, employing two-way modulating control valves at their terminal units. Cooling systems are usually designed to maintain a constant flow through chillers. Constant flow is achieved by using three-way automatic control valves on the condenser side and a primary-secondary arrangement on the chilled water side. The primary pumps maintain constant flow through the chillers whereas the secondary side is variable flow. The secondary pump is often fitted with a variable speed drive, essentially eliminating the need for trimming impellers in these pumps. In this study, three building pumps were evaluated for energy savings from impeller trimming: heating pumps, primary chilled water pumps and condenser pumps.

Pump Oversizing Practices

Pump oversizing often stems from prudent engineering practices such as:

• Incorporating safety factors into the design to accommodate unexpected field changes that may arise during construction and that may increase the pump's system resistance requirements. Pipe routing changes or equipment substitution with an "or equal" are common changes that can also increase the total system head requirements.

• Future expansion plans, which often result in oversizing for immediate needs.

The oversized pumps resulting from such design considerations then must have their excess capacity counterbalanced in the field. Balancing valves located at the pump discharge are typically used to achieve this goal. The balancing valve introduces an artificial head in the system, which causes the pump to "ride" up its head-capacity curve or pump curve until design flow is attained. This wastes energy due to excessive pressurization of the hydronic system and losses at the balancing valve. Specifying the design flow to be set by trimming the pump impeller instead of throttling the balancing valve can significantly reduce these losses.

The design engineer then has an ideal situation: continuing the practice of specifying pumps that are large enough to cover any unforeseen contingencies and achieving energy-efficient pump operation. When facility expansion is anticipated a few years later, the energy savings from the trimmed impeller in the interim period, before the expansion, will usually more than pay for a larger impeller later.

Field Data

Building selection for the study was influenced by the willingness of building engineers to participate in the study and

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the size of the facility. The building sizes were limited to over 60,000 ft^2 (5 570 m^2) because smaller buildings often use packaged rooftop cooling units that do not have any pumps. Larger buildings typically use central systems that were better suited to this study. They also represent a major portion of the total office building space in the United States. The Energy Information Administration's Commercial Buildings Energy Consumption Survey data estimates that there are some 38,000 buildings in the United States with an average floor space area greater than 50,000 ft^2 (5 570 m^2).[1] These buildings represent 53% of the total office building floor space area built in the United States.

Field data collection centered on measuring the pressure generated across the pumps used in each system. Portable Bourdon-type pressure gages and pressure test plugs were used for taking pump suction/discharge pressure readings. Some cooling tower systems required the use of a 5 to 30 in. Hg (17 to 101 kPa) vacuum gage to take the suction pressure readings. The remainder of the readings were taken with a 0 to 100 psi (0 to 690 kPa) pressure gage.

Shut-off head and full-flow head readings were taken for each pump. The shut-off head reading was taken with the balance valve fully closed and then used to establish the actual pump impeller size. The shut-off head reading was plotted on the pump manufacturer's factory-verified pump curves. The impeller size is the one that corresponds to the shut-off head.

The full-flow head pressure is obtained by setting the balancing valve and all of the automatic control valves in the system to the 100% open position. This was done to simulate the system's actual resistance at full-flow conditions. The automatic control valves were opened by relieving the control air to the normally open heating valves, and sending maximum control air to the normally closed cooling valves.

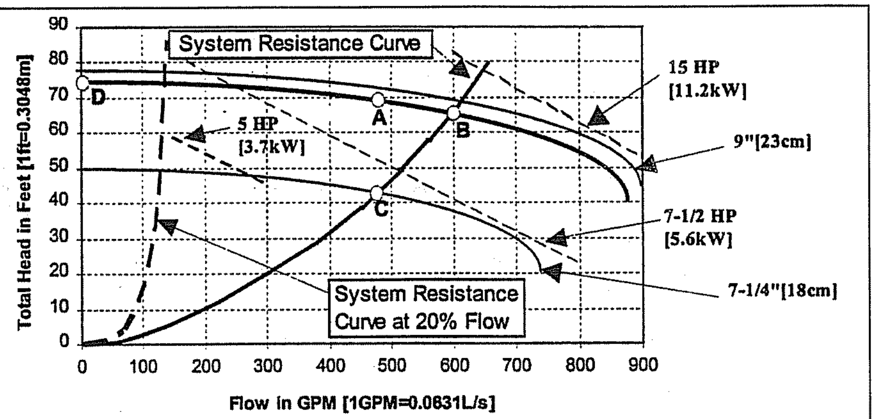

***Figure 1:* Pump head-capacity curve for Building Two.**

(Note: this figure is for illustration purposes only and does not represent a real pump curve.)

A = Engineering (Design) Pump Operating Point: 475 gpm [30L/s] @ 70 ft head [209 kPa].

B = Measured (Actual or full-flow) Pump Operating Point: 600 gpm [37.9L/s] @ 64.7 ft head [193 kPa], ~12 bhp [9 kW].

C = Optimum (Desired) Pump Operating Point: 475 gpm [30L/s] @ 40.5 ft head [121 kPa], ~6 bhp [4.5 kW].

D = Shut-off Head: Establishes pump's impeller size and thus the actual pump curve.

Point C is obtained by the intersection of the design flow requirement, which is a vertical line at 475 gpm [30 L/s], and the system resistance curve, which passes through the actual operating point at 600 gpm [37.9 L/s] @ 64.7 ft head [193 kPa] and is obtained from pump affinity laws:

$$(h_B/h_C) = (Q_B/Q_C)^2 \quad ====> \quad h_C = 0.00018(Q_C)^2$$

where,

h_B, h_C = Head at Points B and C respectively

Q_B, Q_C = Flow rate at Points B and C respectively

Energy Demand Savings at Full-Flow Conditions

= (12–6) bhp x 0.746 kW/bhp = 4.5 kW

Assuming Part Loading at 20% (see text for explanation):

Demand Savings = 2/3 x 4.5 = 3 kW

Because Building Two's pump serves a water source heat pump system, it operates 8,760 hours per year:

Annual Energy Savings = 3 kW x 8,760 hours/year = 26,280 kWh/year.

Energy Savings Calculations

The first step in making the energy calculations was to plot the field measurements onto the pump's factory-verified pump curve data. The pump's shut-off head reading was used to sketch the actual pump curve on the manufacturer's factory-verified pump curve data (*Figure 1*, point D). The pump curve is then used to determine the operating point. The pump curve and the pump head value at full-flow intersect at the operating point (*Figure 1*, point B).

The flow rate at the operating point is read off from the horizontal axis and is compared with the design or nameplate flow rate to determine if the pump is oversized. If the flow rate at the operating point at the full-flow condition indicates a flow greater than the design or nameplate flow (as shown in *Figure 1* by point B), then the pump is oversized and the impeller can be trimmed.

Next, the amount of required trimming is established from the intersection of the design flow and the system resistance curve (*Figure 1*, point C). The system resistance curve is drawn using the values of flow and head at the full-flow operating point. The system resistance follows the pump affinity equations:

$(Q_1/Q_2)^2 = (h_1/h_2)$, where

Q_1, h_1 = The actual measured flow rate and pump head

Q_2 = The selected flow rate

h_2 = The head corresponding to Q_2

Where the system resistance curve meets the design flow value determines the new impeller diameter and also the head's actual resistance in the system at design flow (*Figure 1*, point C). This represents the optimum or desired operating point. Once the desired operating point is established, the energy savings can be calculated.

Building		Heating Pumps			Chilled Water Pumps			Condenser Pumps		
No.	Year Built	Pump Size (hp)	Demand Savings (kW)	Energy Savings (kWh/yr)	Pump Size (hp)	Demand Savings (kW)	Energy Savings (kWh/yr)	Pump Size (hp)	Demand Savings (kW)	Energy Savings (kWh/yr)
1	1994	5	1.6	5,790	7.5	0.1	400	15	8.9	26,600
2	1991	15	3	26,410	–	–	–	–	–	–
3	1952	7.5	0.5	1,690	15	2.9	8,600	–	–	–
4	1966	5	0.7	2,410	–	–	–	–	–	–
5	1994	7.5	1.9	6,750	–	–	–	–	–	–
6	1972	15	4.5	16,400	25	4.5	13,400	25	7.7	18,800
7	1974	30	3.4	12,300	–	–	–	–	–	–
8	1986	–	–	–	50	13.5	40,600	–	–	–
9	1989	15	5.5	19,780	20	3.2	9,600	40	6.5	19,400
10	1981	–	–	–	15	2.3	12,240	25	8.9	47,880
11	1989	7.5	1.3	4,820	10	0	0	25	4.3	12,440
12	1970	10	1.2	10,560	20	5.5	16,400	25	5.5	16,400
13	1984	5	0.5	1,690	–	–	–	–	–	–
14	1972	25	3.7	13,270	15	3.8	12,352	25	6	19,500
Average:	–	–	**1.7**	**8,710**	–	**2.5**	**8,110**	–	**3.4**	**11,500**

Table 1:* *Summary of electrical demand and annual energy savings by building.

The savings from the heating pumps and the cooling pumps are calculated separately. The annual energy savings for the heating pumps are calculated with the assumption that they are operating at a part load condition for a significant portion of the time. This is mainly due to two factors:

• Pump flow rate is non-linear with respect to terminal heat output.[2] Therefore, in a low-temperature water system that has a supply temperature of 180°F (82°C), the terminal heat output is about 90% of the design capacity at 50% of design flow rate. Since buildings are normally at part load conditions, the actual flow rates in the system is considerably lower than design.

• Good design practice suggests that internal gains such as lighting should not be used to discount the building's design heat loss. In actual operation the building load is less than the design and results in the building operating at part load during occupied hours.

Because of these factors, it was reasoned that the heating system operates at part load conditions for a significant portion of the season. These part load factors were incorporated into the energy savings calculations by examining how the boiler hp or demand savings—estimated at full-flow conditions—varied with the building heating load. Using this observation, a demand savings estimate was produced that was representative of the season.

The difference in boiler hp, or demand savings, was empirically observed on the pump curve data and found to vary with the pump flow. When the pump flow went from full-flow down to 20%—corresponding to a building-heating load of 20%—the demand savings decreased by approximately 33%. Therefore, to make conservative energy savings estimates, the heating pumps were assumed to be loaded 20% on average. The demand savings at this loading were selected as representative of the whole season and used in the energy savings calculations.

The cooling system boiler hp savings measurements did not require adjustments for part load conditions because both the primary chilled water pumps and the condenser pumps have a constant load characteristic. Their flow rates are independent from building load changes.

Finally, the annual hours of operation for the heating pumps used in *Table 1* were determined by asking the building engineers how long they ran the pumps, and in some cases, estimates were made from the outside temperature setting used for automatically turning the heating pumps on/off. In *Table 1* the heating season energy use was based on 3,600 annual operating hours. The annual hours of operation for the cooling system pumps were obtained by dividing the chillers' cumulative hour log data by the number of seasons the chiller had been in operation. *Table 1* calculations for cooling energy savings used 2,000 annual operating hours.

Results

The results of the energy savings analysis are summarized in *Table 1*. In some buildings no pressure test taps were found

[illegible]		[illegible]			[illegible]		[illegible]		
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	Energy Savings ($/Yr)	Demand Savings ($/Yr)	Total Cost Savings ($/Yr)	Total Cost Trimming [illegible]	Simple Payback (years)
1	188	67	14	4	899	312	1484	2400	1.6
2	856	126	–	–	–	–	982	800	0.8
3	55	21	291	102	–	–	469	1600	3.4
4	78	29	–	–	–	–	107	800	7.5
5	219	80	–	–	–	–	299	800	2.7
6	531	189	453	158	635	270	2236	2400	1.1
7	398	143	–	–	–	–	541	800	1.5
8	–	–	1372	473	–	–	1845	800	0.4
9	641	231	324	112	656	228	2192	2400	1.1
10	–	–	414	81	1618	312	2425	1600	0.7
11	156	55	–	0	420	151	782	1600	2
12	342	50	554	193	554	193	1886	2400	1.3
13	55	21	–	–	–	–	76	800	10.5
14	430	155	417	133	659	210	2004	2400	1.2
Average:	**$282**	**$71**	**$274**	**$88**	**$389**	**$119**	**$1,220**	**$1,540**	**1.3**

Note: 1. For calculating the demand charges it was assumed that the heating season is from the beginning of October to the end of April, and the remaining months comprise the cooling season. Further, the heating pumps are only operated in the heating season, and the CW and condenser pumps operate only during the cooling months.

Table 2:** **Summary of total cost savings and the simple payback of Building One.

at the pumps, whereas in others it was difficult to simulate the full-flow condition through the control system. Therefore, as *Table 1* shows, data from only 12 heating pumps, nine chilled water (CW) primary pumps and seven condenser water pumps were collected, representing 67% of the total pumps.

The total average savings in summer electrical demand per building were approximately 9 kW and the average annual energy savings per building were 28,000 kWh. These averages reflect savings based on only 67% of the total pumps. Greater savings can be expected on average if the limitations mentioned earlier are overcome. If a pump's life is 10 years, then the total savings will be 280,000 kWh.

Discussion of Findings: Several observations were made with respect to the operating efficiency of heating and cooling pumps in the 14 buildings:

• It was discovered that the balancing valves on almost all the pumps were in the fully open position with the result that the actual flow rate was greater than the design requirement (*Figure 1*, Point B). Although these pumps may have been balanced when the building was new, the balance valves were no longer at their balanced settings. This situation results in more energy consumption by the pumps than if the balancing valves are throttled to achieve design flow (*Figure 1*, point A).

• In two cases the pumps were so oversized that trimming the impeller down to the smallest diameter recommended by the manufacturer still would have left the pumps generating more head and flow than design requirements. A better solution is to replace the existing 1750 rpm motor with a smaller size low-speed 1150 rpm motor. This can be accomplished with minor field modifications to the motor base. In addition, the motor efficiency would improve slightly because the smaller motor will be more fully loaded.[3]

• One building had recently implemented a major lighting energy conservation retrofit. This significantly reduced cooling and thus CW flow requirements, which meant that the original design cooling system flow rates were no longer valid and the new flow rate requirement at the decreased load needed to be determined. The new flow requirement can be determined by observing the maximum loading on the chiller in the first cooling season following the lighting retrofit and comparing this with the minimum flow rate limit of the chiller. The greater of the two establishes the new CW flow requirements of the system. When the new CW flow rate has been established, the pump impeller can be trimmed to meet that flow.

Economics of Pump Impeller Trimming: Two factors affect the economics of pump impeller trimming: the trimming cost and the electric rates. Trimming cost consists primarily of labor. The total labor time involved will vary for each site because of factors such as the accessibility of the pump, the size of the pump, the location of the building site from the contractor's machine shop and the amount of trimming needed. The time needed to balance 5 to 75 hp (3.7 to 56 kW) pumps in accordance with the Hydraulics Institute's Centrifugal Test Code is eight hours.[4] In the study, a two-person per day labor estimate was preferred. With a labor rate of $50 per hour the total trimming cost is $800 per pump. This trimming cost was used in the calculations in *Table 2*.

The electric rates were obtained from the local utility.[5] The utility rates are segregated by season and are slightly lower in the winter, which is a non-peak season. The winter rates comprise energy charges of $0.032 per kWh consumed with a demand charge of $6 per kW. Similarly, the summer rates are $0.034 per kWh and $7 per kW. For purposes of this study, the seasonal changeover dates for the utility's rates were assumed to coincide with the building pumps. This simplified the energy savings calculations. It is worth noting that these electric rates are much lower than the national average electric rates. In comparison the national average electric rates for commercial use are $0.076 per kWh,[6] and so the energy savings in other parts of the country may be higher on average.

A simple payback was calculated for each building and is shown in *Table 2*. Ten of the 14 buildings had a simple payback of less than two years, two buildings had a payback of less than four years and two had paybacks greater than five years. The total savings ranged from less than $100 per year to a maximum of $2,400 per year. Using an average cost savings of $1,220 and assuming the building pumps have a life of 10 years, the total average savings that can be expected over the life of the pump are $12,200.

Conclusion

Pump impeller trimming has several advantages, the first of which is a short simple payback (often the criteria driving project implementation decisions). The simple payback was discovered to be less than two years for most buildings. Another advantage is that the impeller trim balancing procedure is relatively simple to implement. It is based on well understood—albeit seldomly applied—engineering principles.

The implementation requires commonly available pressure measuring instruments, such as a differential pressure gage or Bourdon pressure gages with pressure test plugs. The actual impeller trimming process involves two steps: trimming the impeller and balancing it, usually with a dynamic or static balancing apparatus. The necessary equipment is commonly available in machine shops.

Finally, all of the systems in the study were found to have a 100% standby pump so that impeller modifications in a retrofit application could be carried out in mid-season without disrupting building operations or comfort.

Pump impeller trimming could be economically beneficial to large end-users such as universities, school districts, and state and federal buildings, which have a large number of pumps in use.

The operating efficiency of pumps can be improved in existing buildings and new construction projects. Existing buildings exhibit a potential for energy savings. In this study more than 90% of the 27 pumps studied showed a potential for saving energy and reducing pump operating costs.

Impeller trimming can also be used on new projects. In fact, the benefits in new projects may be greater than in an existing building, since the incremental cost of the impeller trimming will be lower than in a retrofit application. On new projects, the engineer can specify the balancing contractor to recommend the proper diameter impeller and specify that throttling the balancing valve is unacceptable. Specifications must require the impeller to be statically or dynamically balanced after trimming. It may be best to have the trimming done through the pump manufacturer to ensure quality for the pump's performance. The manufacturer can also furnish an updated nameplate tag that reflects the as-built condition of the pumps.

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