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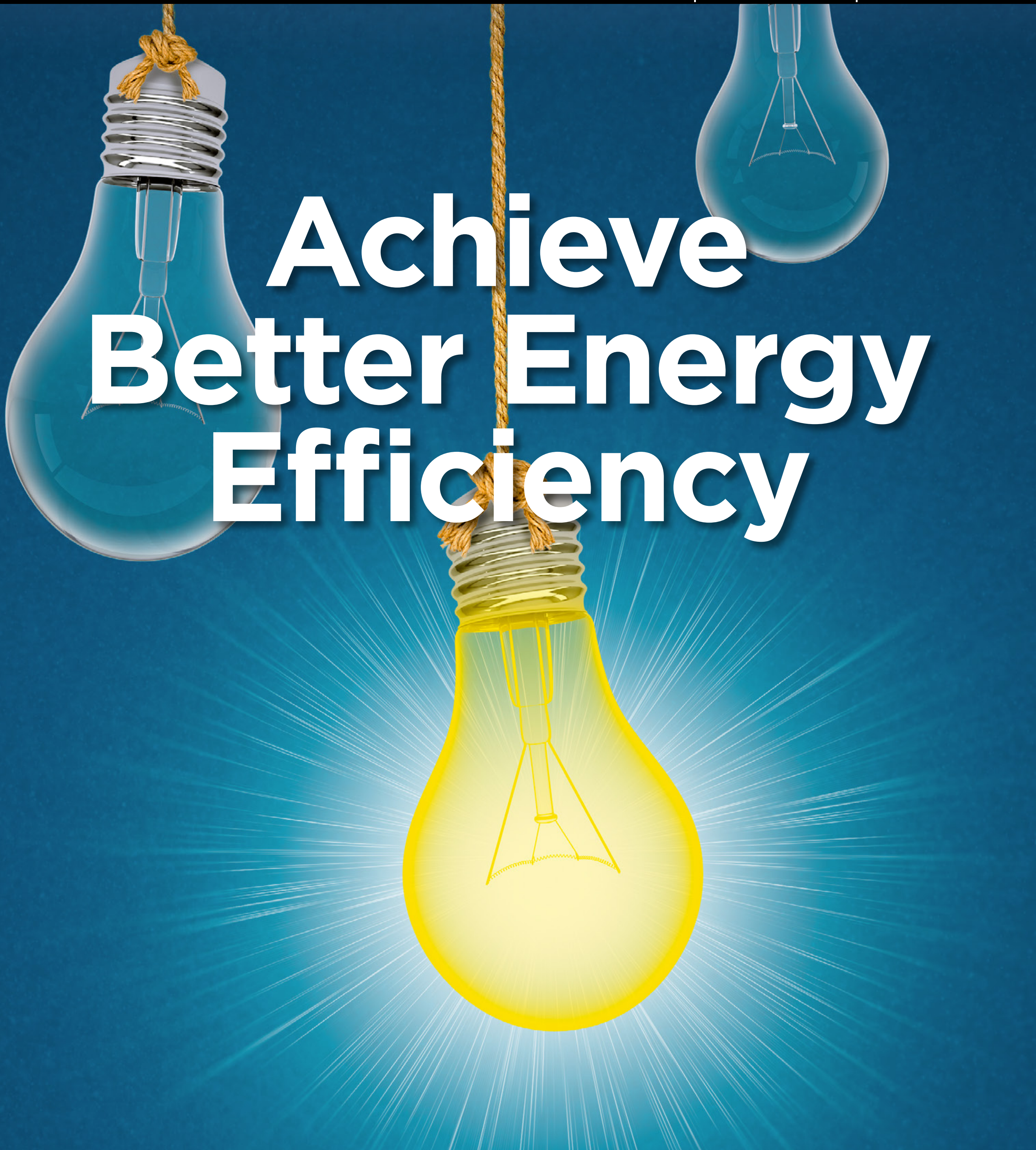

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Take a Closer Look at Equipment Performance

Modern instrumentation packages reduce the need for testing — but not always

By Earl Clark, Energy Columnist

To performance test or not performance test, that is the proverbial question. Jake often was asked this question. He knew the answer wasn't easy or general enough to cover all situations.

When he first began performance testing, a code for field performance testing existed, and instruments usually were calibrated pressure gauges and NBS [National Bureau of Standards] reference mercury-filled thermometers. Orifice plates were removed prior to testing and inspected, measured and replaced, if necessary. His mentors had drilled into him the need to assess measurement errors. In addition, there was the human element. To properly test in those days would require a cadre of personnel. Thhis human factor then became part of the measurement error. Thermometers were selected so they could be read when they sat in the well. Unfortunately, sometimes the thermometer would need to be pulled part of the way out to read it.

Then, came temperature recording boxes. RTDs or thermistors were connected to the box and Jake read in one place all of the temperatures at one time. Jake took great care in calibrating the individual temperature elements before a test and then maintaining an ice bath during the test to do calibration checks. He built all the steps into his methodology.

Next, the manufacturer of his box developed an interface that would hook up to his portable computer. This allowed Jake to develop a program to monitor the test while he conducted it. The prospects elated Jake.

Do you know how your equipment is operating?

What he found, though, gave him pause. When he had not been able to measure the discrepancies in data, he was oblivious to the accuracy of his tests. Now he found himself asking the question, “What are we getting when we field performance test a unit?” (In a future article, we will attempt to clarify and show examples of what this question raises.)

CFC PHASE-OUT EXAMPLE

Jake and TJ had worked together for years when the chlorofluorocarbon (CFC) phase-out came around. As part of the corporate CFC phase-out program, they were charged with retrofitting, removing or replacing all of their CFC-containing refrigeration equipment. The task spanned several years. Each of the group’s engineers received a list of refrigeration equipment to evaluate.

TJ was responsible for a -40°C system with two large refrigeration machines and one small machine used primarily during plant startup, and then run sporadically as a topping unit when production was high. The nearly 30-year-old unit still ran, but it was questionable as to whether it would be retrofitted. All of the units operated on CFC-12. As part of the project, each machine had to be present-state performance-tested, meaning engineers tested the unit as it stood — with no tuneup or maintenance touchups. Also, each equipment manufacturer provided a reselection evaluation for HFC-134a operation. TJ had the reselection in hand when he set out to do the performance test.

During testing, TJ kept looking at the portable computer. Something did not seem right. He reran the test and the results came up the same. He recalibrated the RTDs and reran the test. TJ was stymied; the results were the same. He called Jake and said, “I need to talk about this test. It is a head scratcher!”

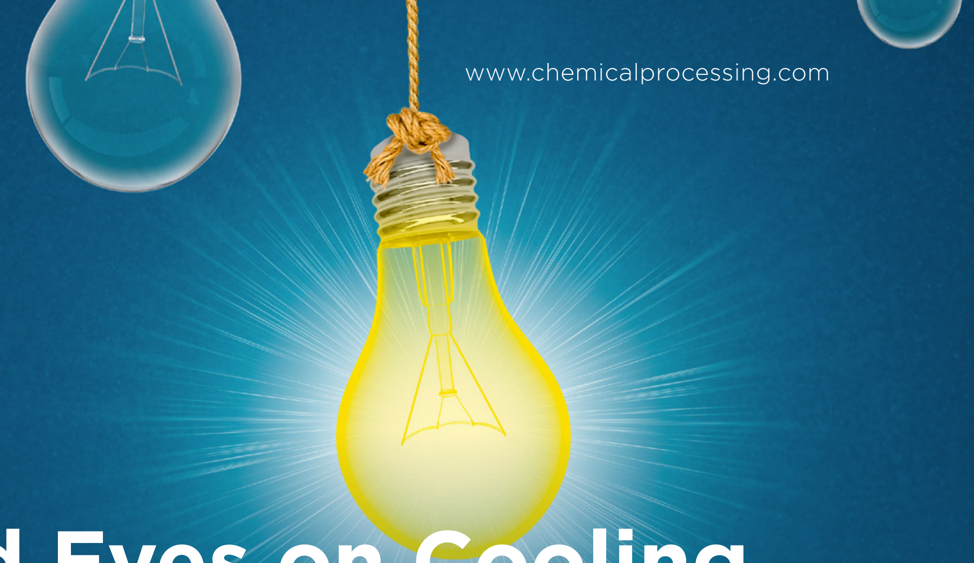
“Jake, can you tell me where I am going wrong on this test?” TJ asked when they met. They reviewed test data, design data and the reselection. It appeared the compressor was operating below 45% efficiency. Digging into interstage pressure and temperature details, they found the second stage of the three was at 20%. Jake wondered if there was an obstruction. They pulled out the reselection. Dimensions for each stage as built were included in the manufacturers report. Usually, in a multistage compressor, each higher stage has a smaller exit wheel width because the compressed gas has a lower specific volume at

Improvements in technology for performance testing now allows seeing so many things probably missed in the past.

the higher pressure. They found the second stage's width was nearly double in size, resulting in inefficient performance; it may have been operating in surge on occasion. TJ wondered, "How had this been missed for so many years?" Jake explained that improvements in the technology for performance testing now allows seeing so many things probably missed in the past. Because this was an auxiliary machine and operated sparingly, it just languished.

While this is just one example, it raises questions. Do you know how your equipment is operating? Have you done performance testing recently? Did you perform an acceptance test on your equipment when it was installed? In the next few columns, I will cover more on this topic, including performance testing reviews, performance test versus acceptance test, online testing, modern field testing and daily performance monitoring. ■

EARL M. CLARK is engineering manager for Global Energy Systems Group, and energy columnist for *Chemical Processing's* monthly Energy Saver column. He can be reached at eclark@putman.net.



Cast Cold Eyes on Cooling System Interactions

Take a look at the cooling tower, refrigeration compressor and condenser

By Earl Clark, Energy Columnist

Plant personnel often asked Jake if they should install variable speed drive motors on their cooling tower fans. He usually replied, “I have normally found that one horsepower saved on the fan requires three horsepower more in compressor power.” Jake’s experience on multiple field tests and computer optimizations led him to this rule of thumb.

However, in the late 80s and 90s, as the phase-out of chlorofluorocarbons began, refrigeration manufacturers tightened up not only the containment of their equipment but also the performance. Prior to that, power consumption varied within the 0.7 to 1.2 kW/ton range. Steady improvements brought chiller power consumption down in the 0.4 to 0.6 kW/ton range. So, Jake’s old rule of thumb probably didn’t work anymore.

To check this, Jake first thought about how individual component acted in the system and each’s constraints. He broke it down to the cooling tower, the refrigeration compressor and condenser. He then identified how these would interact.

For the cooling tower, Jake looked at some readily available samples of manufacturers’ curves. For optimal cooling, the fan blades are fixed at an airflow that maximizes the available motor horsepower. The curves are set to correlate to water flow and ambient air wet bulb temperature (WBT). The constraints on the tower include the air’s ability to absorb water vapor for the

While they didn't achieve the full potential of 22%, they were able to reduce energy by 15%.

incoming water and the sensible temperature rise of the moist air leaving the tower. As the ambient air WBT increases, the air's ability to absorb more water vapor — as well as the sensible temperature — decreases. Towers are designed with a set approach temperature, typically the difference between the WBT and the exiting water temperature, of 7–10°F. However, as the tower's heat load decreases with high WBT, the approach temperature will fall, but not in proportion to the tower's airflow. So in effect, air power doesn't show full benefit versus the exiting water temperature. At that point, reductions in airflow will result in less power, which won't impact the refrigeration condenser.

In the condenser, the design conditions dictate the design kW/ton. That is but one point on the refrigeration compressor map. The kW/ton can vary considerably, depending on the evaporator and condenser operating conditions. The compressor has to lift the refrigerant gas from the evaporator to the condenser where it is condensed at a temperature largely dependent on the cooling tower exiting water temperature. The lower that temperature is, the lower the lift on the compressor, and the lower the compressor hp/ton. Generally, each 1°F reduction in condensing temperature trims compressor power 1.5 to 2.5%.

The major constraint on the condenser is a minimum pressure drop across the thermal expansion device. This could be a fixed orifice, a float valve or a control valve. Falling below the pressure drop could restrict flow to the evaporator thus increasing refrigerant liquid level in the condenser, a condition known as stacking. This would reduce the surface area in the condenser, raising the pressure and eventually slightly increasing flow.

The second constraint is the compressor operating curve. This can be modified by using inlet guide vanes to change the capability of the compressor to lift the gas. A variable speed drive either with a variable speed motor or a turbine drive also can serve to optimize compressor efficiency.

Jake recognized this analysis was more complicated than his old guidance. He developed a computer model to accurately assess the lowered condenser temperature impact versus the reduced cooling tower airflow impact. He picked several good test candidates where the tower and the con-

Compare your various options and the capital required to the savings achieved. Use this to make an informed decision on how to proceed.

denser were uniquely connected; parallel towers and chillers would make the job difficult.

What he found was it “usually” made more sense to run the cooling tower to minimize condenser temperature and pressure. In his case, the exceptions were high ambient WBT with light loads, off peak season operations where lowering the exit cooling tower water temperature came up against minimum required condenser pressures, and a few others. The results surprised Jake. He attributed it to the advances made in refrigeration equipment efficiencies.

So, start collecting your data on the systems under consideration. Develop a model that looks at the constraints on the chiller as well as the cooling tower. Look at the interactions of the cooling tower, the refrigeration compressor and condenser. Compare your various options and the capital required to the savings achieved. Use this to make an informed decision on how to proceed. Happy energy hunting. ■

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Don’t Generalize Complex Systems

Deviating from standard design temperatures requires careful evaluation

By Earl Clark, Energy Columnist

Nearly all of us have designed heat exchangers. We use standard temperatures dictated to us by our seniors. The standard at Jake’s company was 10°F. The approach temperature for the column to cooling fluid was 10°F. The delta T for the cooling fluid was 10°F. The approach of the refrigeration evaporator to the cooling fluid was 10°F and the delta T for the cooling fluid was 10°F. The approach of the cooling tower water to the refrigeration condenser was 10°F. The delta T of the cooling tower water was 10°F. The cooling tower operated with a range of 10°F and the approach to ambient wet bulb was 10°F. Oh, and the design fluid velocities were all 7 ft/s.

Jake eventually moved from the design job to a plant. There, he discovered a completely different world. He constantly found operators violating his strict upbringing on design temperatures. When he asked why, they responded, “We found a sweet spot and the process runs better there!” Also, “If we could just get more flow we could improve the process yield.”

As he gained more experience in the field, Jake realized the operators were settling on about 70% of the design delta T that he had used. They also were operating the cooling fluids at about 10 ft/s instead of 7 ft/s. This seemed to be pretty universal whether it was the process area or the HVAC systems. He coined Jake’s law: operations will run the heat exchangers at 70% of design delta T.

So what are the implications of Jake’s law? Are there extra costs? Does the increased yield pay for the added expense?

Operators were settling on about 70% of the design delta T.

The first thing to note is the flow rate increased by nearly 50%, so naturally, pumping costs went up. But by how much? The increased flow raises pump energy proportionally. However, there may be an increase in pressure. Because the velocity increased by 50%, the incremental pressure drop went up by the square of 1.5 or 2.25. Flow × head (or pressure) = pump energy, so it rose by a factor of 3.37. Obviously, there's a base pump head, so this would cause a smaller overall pumping cost increase. Depending on where the online pumps fall on their pump curves, another pump may be needed. Also note that this additional energy is passed on through the system as increased energy load and will eventually be dumped to the cooling tower.

Next, Jake reviewed the resultant log mean temperature differences (LMTD) and their effects through the system. The change in process temperature results in a lower LMTD in the evaporator, which in turn reduces the evaporator pressure, raising the lift and the power required. The slight rise in pumping power is extracted through the evaporator, also increasing power needs.

Finally, if the process load is raised as a result of the process improvements, this also will require more power.

So, with all the generalizations — potentially with negative results, Jake was perplexed. To solve his dilemma, Jake developed a model to determine what would actually happen. He modeled each heat exchanger in the system. He added a decision box to use actual fouling factors. He used pump curves and a selection process for pumps on line. He modeled the compressors in the refrigeration machines. He also added a selection box for each refrigeration machine in the plant.

The result? Jake concluded you should never use generalizations when dealing with a complex system. The height of the columns added a static load to the overall pump pressure that masked the increased pressure across the heat exchangers. The slight increase in pressure moved the pumps to a better efficiency point on the curve. Jake also found that operations had one more pump online than needed. First savings achieved. The process heat exchangers were a lot cleaner than expected; actual LMTDs were better and resulted in

Develop models for complex interacting systems nd test your assumptions — and the generalizations!

improved performance compared to the conservative design points.

The refrigeration machines were being operated in a partially unloaded mode decreasing the lift; reducing the number of machines to only those necessary to handle the load cut the energy consumption. In the end, Jake was able to show a 10% reduction in energy while production increased by about 5%.

So, be careful using generalizations. They can lead you astray. Develop models for complex interacting systems and test your assumptions — and the generalizations! Happy energy hunting!

EARL M. CLARK is engineering manager for Global Energy Systems Group, and energy columnist for *Chemical Processing's* monthly Energy Saver column. He can be reached at eclark@putman.net.

Consider Refrigeration Condenser Pressure

Operating at design condenser pressure could wind up wasting energy

By Earl Clark, Energy Columnist

Condenser pressure control on refrigeration machines is often not well understood. Minimizing the pressure can result in significant savings. Most refrigeration machines are designed for worst-case condenser pressures, which usually occur during peak summer temperatures. Most are cooled be a cooling tower, which may struggle during peak summer conditions. Several key operating points can help minimize energy consumption.

First, the lower the compression ratio is, the lower the energy consumption. Most chilled water systems operate with about 60°F temperature difference between the condenser and the evaporator. A rule of thumb is that each °F equates to 1/60 or about 1.5% energy use. Specific systems may vary from this, but this is a quick way to estimate how much energy you could save by reducing condenser pressure.

If you operate a low temperature system, the change is less dramatic. However, you have to remember that the horsepower required is also higher so a small change in temperature can result in large energy savings.

The caveat is that you must have enough differential pressure between the condenser and the evaporator or intercooler/economizer so that the required flow will still pass from the condenser to the evaporator to maintain system capacity. Otherwise, refrigerant will stack in the condenser and tubes in the evaporator will be above the liquid level, resulting in higher superheated temperatures

While they didn’t achieve the full potential of 22%, they were able to reduce energy by 15%.

in the evaporator and reduced capacity in the condenser due to flooding.

Various experts have suggested variable speed fans on cooling towers to minimize their energy consumption. What these experts seem to miss is that the higher fan cost associated with the cooling tower are more than offset by the refrigeration machine’s reduced energy. Let me back up to say that there’s a tipping point in the cooling tower. At some point, more airflow and more fan power won’t decrease cooling tower water temperature proportionately and the extra power is wasted.

JAKE’S TEST

Jake had been called to a site for a boiler problem. The plant took its cooling water flow from the bottom of Lake Michigan. While on site, Jake also observed the operation of the low temperature chillers used to cool reflux columns in the process. The low temperature brine was supplied at about -60°F and cooled by a cascade refrigeration system utilizing two separate refrigerants in two separate refrigeration machines [We should be consistent in using one or the other! So, I’ve changed the temps to °F. This makes some statements look a bit odd, like using a 3.6°F increment instead of 2°C. Anyway, ask Earl, which he prefers and ensure we use the same units throughout]. The upper stage cooled the lower stage at about 15°F and was itself cooled by the lake water.

Jake asked why the condenser temperature on the upper stage was near summer conditions while the lake water temperature was at about 40°F. The operator indicated that he had been told to maintain design condensing temperature. This was based on peak design during the summer when lake temperatures climbed to near 70°F.

Jake visited the plant engineer and inquired why they were running such high temperatures when low temperatures were available. The plant engineer stated they needed the higher temperature because the machine wouldn’t run unless it was at the higher condenser pressure. Jake did a quick calculation. The temperature differential was about 120°F. Each 1.2°F would result in a 1% energy reduction. A 27°F reduction might result in 22% decrease in energy consumption. The plant engineer

If you have a refrigeration machine that's operating at a design condenser pressure even though colder water is available… You might be able to reduce your energy bill significantly!

decided the savings were worth testing. Jake and the plant engineer decided to do the test in 3.6°F increments. Because the old machines essentially were controlled manually, they decreased the condenser pressure on the upper stage and then trimmed the set point for the condenser/evaporator between the two stages. The reduced steam-driven compressor speed lessened the differential pressure between stages. This reduced and balanced the differentials on each stage. They were careful to make sure that process conditions were met at each phase of testing. While they didn't achieve the full potential of 22%, they were able to reduce energy by 15%.

So, if you have a refrigeration machine that's operating at a design condenser pressure even though colder water is available, start asking questions. You might be able to reduce your energy bill significantly! ■

EARL M. CLARK is engineering manager for Global Energy Systems Group, and energy columnist for *Chemical Processing's* monthly Energy Saver column. He can be reached at eclark@putman.net.

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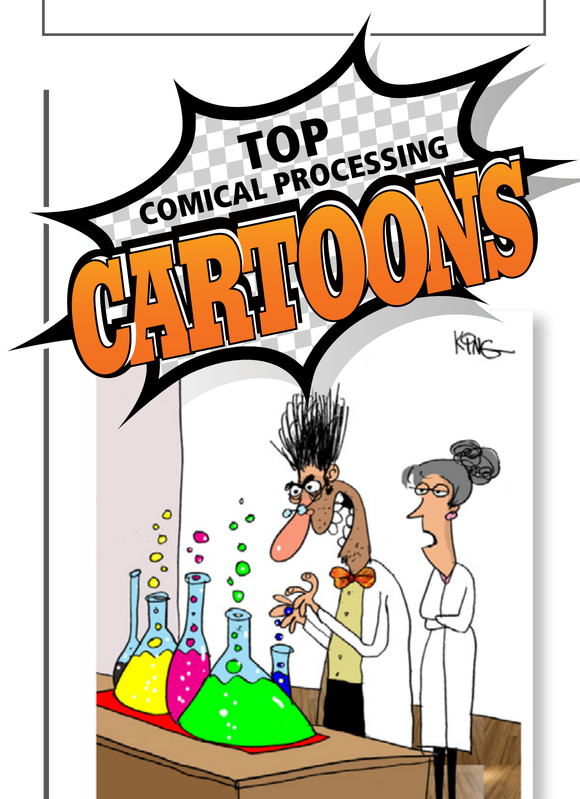

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