CHEMICAL ENGINEERING

September 2012

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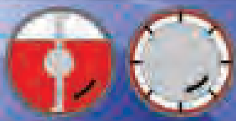

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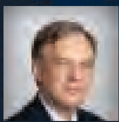
KEYNOTE PRESENTATIONS

The Evolving Shale Landscape, Wednesday, November 14

Two industry experts address the environmental and regulatory issues, as well as the economic impacts of unconventional oil and gas drilling one week following the Presidential election.

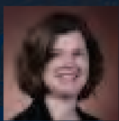
Moderator: Rebekkah Marshall, Editor-in-Chief, *Chemical Engineering*

SPEAKERS:



The Shale Gas Prize and Its Politics: The Risks and Rewards for our Environment

John Hanger, Special Counsel, Eckert Seamans



Shale Gas: A Game Changer Impacts for American Chemistry, Manufacturing & U.S. Economy

Martha Gilchrist Moore, Sr. Director for Policy Analysis & Economics, American Chemistry Council

Plant Manager Roundtable, Thursday, November 15

Moderator: Rebekkah Marshall, Editor-in-Chief, *Chemical Engineering*

PANELISTS:

Jim Armstrong, Operations Manager, Rhodia, Inc. - Baton Rouge

Paresh Bhakta*, Site Director, Clear Lake Plant, Celanese Ltd.

Jim Hull, Vice President - Manufacturing, Georgia Gulf Chemicals and Vinyls, LLC

** Denotes Invited Panelists*

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BPA

Opinions on ChE education

Last month, we ran a letter from reader, Jason Makansi, commenting on our June story, Chemical Engineering Education Evolution (*Chem. Eng.*, June 2012, pp. 22–25), which highlighted, among other things, the need for a more-meaningful connection between academia and industry. Makansi's letter was passionate in itself, expressing frustration at the slow pace of progress over the past 30 years or more. Equally passionate was the response that Makansi's letter has prompted from the rest of our readership. While each opinion differs somewhat with regard to a solution, it is safe to say that most practicing engineers agree that the typical ChE graduate has always lacked some basic practical skills.

In Makansi's letter, he illustrates the central problem by explaining that when he graduated with his B.S.Ch.E., he did not know "which way to turn a valve, or, for that matter, anything else about an actual process plant." Other readers, such as those whose letters appear on p. 6 of this issue, graduated with similar deficits in their practical know-how, with one admitting that "as a newly graduated engineer from a respected university, I didn't even know what a pipe flange was."

Of course, this is not the first time that the gap between academia and industry has been raised. Each one of you might have his or her own illustration to offer. I, myself, often lament that the main thing I recall from my process-control class is learning how to perform Laplace Transforms — a skill I never had the opportunity to retrieve in practice and probably never will.

The question is not whether graduates lack hands-on experience; the question is what should be done about it. For nearly 30 years, Makansi's suggestion has been that engineering schools should offer two tracks — one for students likely to continue through graduate school and one for students who would be seeking jobs in industry.

Meanwhile, one reader, Roger E. Blanton, points out that adding any standalone practical-training courses would exacerbate the difficulty that most students find in completing their ChE degrees in four years. His suggestion is to make more use of summer internships.

Another reader, James S. Bloss, essentially makes the argument that nothing should be done about it. In his opinion, the chemical process industries are far too broad to incorporate enough hands-on specifics to make a significant difference, and core lessons in discipline and problem solving can go much farther.

In my opinion, all three readers make good points. Problem solving is indeed a foundational skill that can be applied across industries and functions. And, like other readers, I often wonder "whatever happened to co-ops?", which used to be a common route for a ChE student bound for industry.

Like Makansi, though, I think that the education experience should be more tailored to the student's professional goals. A twist on his two-track idea could be an applied-ChE degree option. Meanwhile, I, personally, do not object to longer, five-year programs. After all, I transferred from architecture, where that is an accepted state of affairs. Admittedly, however, not everyone embraces a five-year degree option. So, with that in mind, I think it is important to consider that practical learning does not necessarily mean more coursework. It is more of a distinction in the method of learning itself. If students had more opportunities to turn valves, see heat exchangers and work with actual process control systems, I would argue that absorption of the academic fundamentals would be more rapid in the first place. ■

Rebekkah Marshall

Letters

Re: Education needs a reality check...

I would like to pass on a few comments on Jason Makansi's letter titled "Education needs a reality check", which appeared in the August 2012 edition (p. 6).

I too, like Jason, began my now 36-year career not knowing which way to turn a valve but being fully trained in transport phenomena. Did my university fail me? No. My university's ChE program (B.S.Ch.E. 1976, University of Akron) provided a very essential engineering skill-set: discipline and problem solving. Sure, I would like to see new hires have more practical knowledge, but the engineering field is so broad that I think it is difficult for any university to meet the need of every industry.

In my field, power generation, we utilize engineers who excel at transport phenomena and finite element analysis (FEA), as well as engineers who excel in skills developed through the sweat of their brow, such as welding, machining or equipment repair. The common trait I observe is that successful engineers have the "knack", a curiosity about how things work and how they can be improved. This is a teachable skill. I know. I was able to teach it to my son, a young electrical engineer. It constantly amazes me how his mind never stops working, investigating and inventing. This is the skill engineering programs need to focus on.

James S. Bloss, P.E., Sr. Principal Engineer
The Babcock and Wilcox Co., Barberton, Ohio

...internships should fill the gap

I read with great interest Mr. Makansi's letter in the August 2012 *Chemical Engineering* periodical. It struck a chord with me when I recalled that as a newly graduated engineer from a respected university, I didn't even know what a pipe flange was. However, I was fortunate enough to be hired by a world-class air-pollution control equipment manufacturer. There, I quickly learned and became an "expert" as I was sent out to start up newly installed systems. I'm not saying this is ideal, but it is reality. Many students cannot complete an engineering curriculum in four years. Adding any practical training courses would exacerbate that goal.

The solution may lie in summer intern jobs where students can gain practical insight into how their education can be applied, and learn about necessary topics not covered in the classroom, like how to turn a valve. In addition, it provides a company an opportunity to try out a potential employee and target those students that fit their organization at graduation. I believe that an engineering degree doesn't provide everything a graduate needs to know to function in our industry, but it is an indicator to an employer that the individual has the capacity to learn. If we work together with educators and provide opportunities to supplement the academic curriculum with real-life experiences in summer positions, we can all win.

Roger E. Blanton, P.E., Business Development Manager
John Zink Co., LLC, Tulsa, Okla.


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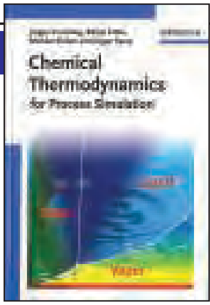


Chemical Thermodynamics for Process Simulation. By Jürgen Gmehling, Bärbel Kolbe, Michael Kleiber and Jürgen Rarey. Wiley & Sons Inc., 111 River Street, Hoboken, NJ 07030. Web: wiley.com. 2012. 760 pages. $135.00.

Reviewed by David Hill,
Chemstations Inc., Houston

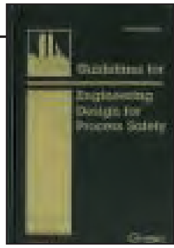
The authors of this text are leaders in chemical engineering thermodynamics. Based on their status, I recommended this book in a blog entry before even seeing a review copy — and I'm glad I did. The book is aimed at both students and professionals who work with process simulators. It assumes a prior understanding of thermodynamics, and is a more advanced text that focuses on how complex thermodynamic situations affect the use and application of process simulation.

Many aspects of applied thermodynamics are not explored in undergraduate courses. Users of process simulation are often left to learn this material on their own. Gmehling's book assembles many of these topics in one text, and supplies examples and sample problems. It is more approachable and extensive than the out-of-print thermodynamics text by Stanley Walas (Phase Equilibria for Chemical Engineering, 1985).

In the book, mathematics — some quite complex — is interspersed with clear explanations of what the mathematics indicates. This is helpful because many practitioners may require a refresher for the mathematics, and many students will not be aware of practical applications that the mathematics can explain.

Effective explanations are provided for why there are different models for physical properties and vapor-liquid equilibria (VLE). Background on some of the models and equations is also discussed, as are the strengths and weaknesses of the methods. The explanations of why we have different types of models (activity methods and equations of state) for VLE are very helpful.

Many topics that routinely confuse users of process simulators are discussed and explained. There is an excellent explanation on the often confusing concept of reference state of enthalpy. Modeling options for "special" systems, such as polymer, acetic-acid-water, or formaldehyde-water, are discussed. Both the theoretical basis and practical implications of azeotropes are also well explained, and the book includes one of the best available explanations of residue curve maps. The explanation on how simulators calculate heat of reaction is also very helpful.



Although English is not the first language for the authors, the text is generally easy to read, despite an occasional clumsy phrase. The text's authors are involved with DDBST products (Dortmund Data Bank, the Dechema database), and contributed heavily to the development of the PSRK (Predictive Soave-Redlich-Kwong) and Volume-Translated Peng-Robinson (VTPR) models. I felt the book promoted these methods and tools too heavily. That being said, I am a strong proponent of the PSRK VLE model and would agree that these tools and methods are powerful. Sample Mathcad files are available, but it would have been helpful to see the examples in Matlab also.

This text explains many of the concepts that my company's technical support group regularly explains to users of our process simulator, and we recommend this book to our customers.

Guide to the Business of Chemistry. By the American Chemistry Council, 700 Second St. NE, Washington, D.C. 20002. Web: americanchemistry.com. 2012. e-book. $385.00.

Guidelines for Engineering Design for Process Safety, 2nd ed. By the Center for Chemical Process Safety. John Wiley & Sons Inc. 111 River St., Hoboken, NJ 07030. Web: wiley.com. 2012. 440 pages. $125.00.

Practical Spray Technology: Fundamentals and Practice. By Charles Lipp. Lake Innovation LLC, P.O. Box 3596, Lake Jackson, TX 77566. Web: lakeinnovation.com. 2012. $385.00.

Practical Thermocouple Thermometry, 2nd ed. By Thomas Kerlin and Mitchell Johnson. The International Society of Automation. 67 Alexander Drive, Research Triangle Park, NC 27709. Web: isa.org. 2012. 181 pages. $89.00.

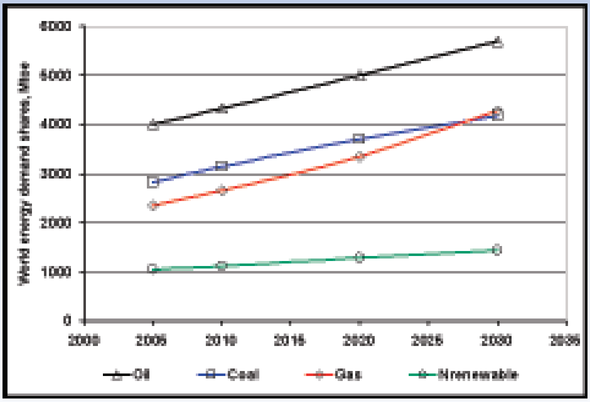
Colour Design: Theories and Applications. Edited by Janet Best. Woodhead Publishing Ltd. 80 High St., Cambridge, CB22 3HJ, UK. Web: woodhead-publishing.com. 2012. 672 pages. $315.00.

CRC Handbook of Chemistry and Physics, 93rd ed. Edited by William Haynes. CRC Press, part of Taylor & Francis Group, 6000 NW Broken Sound Parkway, Suite 300, Boca Raton, FL 33487. Web: crcpress.com. 2012. 2,664 pages. $169.95. ■

Scott Jenkins

Process Insight: *Optimizing CO_2 Capture, Dehydration and Compression Facilities*

The removal of CO_2 by liquid absorbents is widely implemented in the field of gas processing, chemical production, and coal gasification. Many power plants are looking at post-combustion CO_2 recovery to meet environmental regulations and to produce CO_2 for enhanced oil recovery applications. The figure below illustrates actual data of fuel consumption in 2005 and an estimate of energy demand for various fuels from 2010 to 2030. The world energy demand will likely increase at rates of 10–15% every 10 years. This increase could raise the CO_2 emissions by about 50% by 2030 as compared with the current level of CO_2 emissions. The industrial countries (North America, Western Europe and OECD Pacific) contribute to this jump in emissions by 70% compared to the rest of the world, and more than 60% of these emissions will come from power generation and industrial sectors.

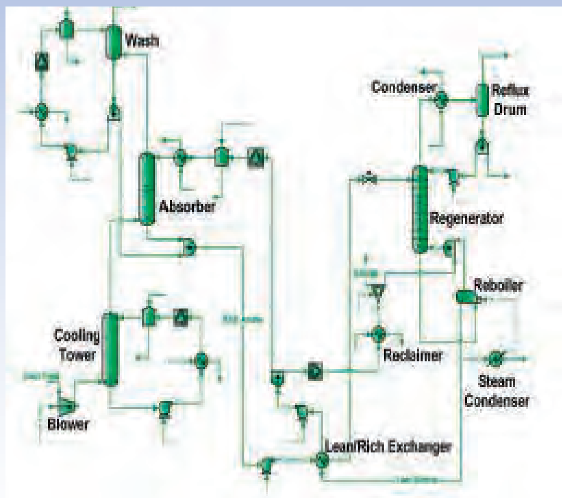


Despite the strong recommendations from certain governments, there are very few actual investments in CO_2 capture facilities geared toward reducing greenhouse gas emissions mainly because of the high cost of CO_2 recovery from flue gas. CO_2 capture costs can be minimized, however, by designing an energy efficient gas absorption process. Based on the findings of recent conceptual engineering studies, HTC Purenergy estimated the production cost to be US$ 49/ton CO_2 (US$ 54/ tonne CO_2) for 90% CO_2 recovery of 4 mole% CO_2 content in the flue gas of NGCC power plants. A separate study showed the cost for 90% CO_2 recovery of 12 mole% CO_2 from a coal fired power plant to be US$ 30/ton CO_2 (US$ 33/tonne CO_2). The cost of CO_2 recovery from coal power plant flue gas is substantially less than that of NGCC power plant flue gas due to the higher CO_2 content in the feed.

The energy efficiency of a CO_2 capture plant depends primarily on the performance of the solvent and optimization of the plant. In traditional flue gas plant designs, MEA was the primary solvent and was limited to 20 wt% to minimize equipment corrosion. Recent developments in controlling corrosion and degradation has allowed an increase in the solvent concentration to about 30 wt% thus decreasing the required circulation and subsequent steam demand. A recent DOE study shows the steam consumption for an existing CO_2 plant using 18 wt% MEA (Kerr McGee Process) is 3.45 lb of steam per lb of CO_2 for amine regeneration. A modern process that uses 30 wt% MEA is expected to use 1.67 lb of steam per lb of CO_2 for amine regeneration. The HTC formulated solvent is a proprietary blend of amines and has a lower steam usage than the conventional MEA solvent. Based on the material and energy balances for the plant designed in the recent study, the reboiler steam consumption is estimated at about 1.47 lb steam/lb CO_2 using the proposed formulated solvent without implementing any split flow configurations. This is much less than the reported steam usage for the MEA solvent.

The design of a facility to capture 90% of the CO_2 from the flue gas of a coal fired power plant is based on the specified flue gas conditions, CO_2 product specifications, and constraints. Using the ProMax® process simulation software from Bryan Research & Engineering, CO_2 capture units can be designed and optimized for the required CO_2 recovery using a variety of amine solvents. The following figure represents a simplified process flow diagram for the proposed CO_2 Capture Plant.

The table below presents the main findings for CO_2 capture from the coal fired power plant and the NGCC power plant, each designed to produce about 3307 ton per day (3,000 TPD metric). To produce the same capacity of CO_2, only one train with smaller column diameters is required in the case of the coal power plant and two trains with larger column diameters are required in the NGCC Power Plant case. This is mainly



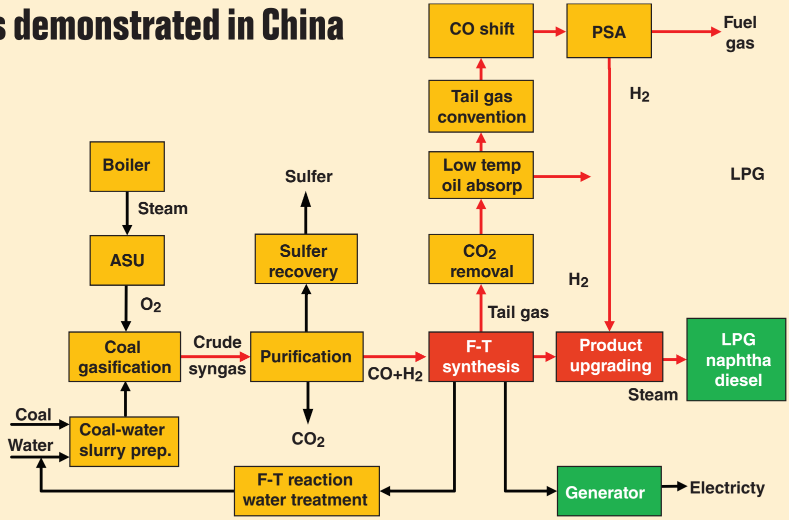
due to processing a larger flue gas with lower CO_2 content in the NGCC power plant. Consequently, a substantial reduction in the capital and production cost was reported for the coal fired power plant CO_2 recovery facility.

Parameter	Coal Power Plant	NGCC Power Plant
CO_2 Production Capacity, ton / day	3307	3307
CO_2 Recovery, %	90	90
CO_2 in flue gas, mol%	12	4
Number of trains	1	2
Flue gas rate, MMSCFD	528	920
Absorber diameter, ft	32.8	39.4
Regenerator diameter, ft	19.7	19.7
Capital Cost, million US$	165	227
Operating cost, million US$	25	51
Production cost, US$/ton	30	49

For more information about this study, see the full article at **www.bre.com/support/technical-articles/gas-treating.aspx**.

Edited by Gerald Ondrey September 2012

A high-temperature F-T process demonstrated in China



Synfuels China Technology Co. (Beijing; www.synfuelschina.com.cn) says it has developed a high-temperature slurry Fischer-Tropsch (F-T) process (flowsheet) that produces mainly clean diesel, naphtha and liquefied petroleum gas (LPG) products, with unique features including: high productivity of proprietary catalyst; efficient recovery of reaction heat by producing 2.5–3.0-MPa(g) steam, which can be used to generate electricity; a uniform temperature distribution in the slurry bed; and easy integration with different synthesis-gas- (syngas) production technologies.

To design the F-T catalyst, the company first used quantum chemistry to calculate the molecular energy levels of different crystal faces, and then achieved the desired catalyst structure by strictly controlling the preparation conditions. This approach resulted in high productivity of the catalyst.

The company's technology has been successfully applied in the Yitai 160,000-ton/yr, and the Luan 160,000-ton/yr demonstration coal-to-liquids (CTL) plants, located at Ordos City in the Inner Mongolia Autonomous Region.

Features of the Yitai plant include: a uniform temperature distribution across the 5.3-m slurry-bed reactor; the ability to easily separate the catalyst from the wax, and replace it online to ensure the continuous and stable operation of the plant; and the utilization of the F-T heat of reaction by producing 3-MPa byproduct steam, which can be used to generate electricity for the power supply of the CTL plant itself.

The Luan plant's features include: nitrogen from the air-separation unit and hydrogen from the pressure-swing-adsorption (PSA) unit are used for the synthesis of ammonia, which is used to produce urea with the high-purity CO_2 from the purification and F-T synthesis units; the fuel gas from the PSA unit is used for integrated gasification combined cycle, improving the total heat efficiency of the plant.

Based on its experience with the demonstration plants, Synfuels China is now working on a few large-scale CTL projects.

A commercial move toward algae farming for CO_2 recovery

Advanced Algal Technologies (Sydney, Australia; www.advancedalgal.com) has signed a $100-million deal with Fuzhou Xiangli Enterprise Management Consulting (Fuzhou, China) — an LED-lighting manufacturer — for a license to build 500 algal-farming-conveyor modular units per year. These units will be used to reduce carbon dioxide emissions in manufacturing plants in China.

Traditionally, algae have been grown either in open ponds, or in bioreactors (see, for example, *Chem. Eng*, September 2008, pp. 22–25). Both entail serious limitations: open ponds are subject to external conditions such as weather, pollutants and wildlife; bioreactors may require expensive sun tracking equipment and involve problems in harvesting the algae. Meanwhile, bioreactors still rely on contact of the gas bubbles in the water to dissolve the CO_2.

In the patented system, developed by Advanced Algal Technologies, the algae are grown on a fabric inside an insulated building, within a precisely controlled environment. The algae can grow to a density of 100 g/m^2 per day. The algae are in constant contact with the correct levels of CO_2 in the atmosphere of the growing structure, and the use of a wet mat technology provides a high molecular transfer rate of CO_2 in solution.

The company said the system allows maximum algae yield. For example, a 3,000-m^2 warehouse fitted with the system can produce more than 200 ton/d of algae or more than 50,000 L of algal oil and 150 tons of high-protein animal feedstock. Such a facility would consume more than 130,000 ton/yr of CO_2.

Artificial photosynthesis

Scientists at Panasonic Corp. (Osaka, Japan; www.panasonic.com) have developed an artificial photosynthesis system that utilizes sunlight to convert CO_2 to organic materials (mainly formic acid) at what is said to be the highest efficiency yet achieved (0.2%). The efficiency level is comparable to that of plants. The keys to the Panasonic system are a nitride semiconductor, which is used as a photo-electrode for the reduction of CO_2, and a metal catalyst to accelerate the reaction and increase selectivity. The research was presented at the International Conference on the Conversion and Storage of Solar Energy in July. The company hopes to use the development in a system to capture and converte waste CO_2 from incinerators, power plants and other industrial processes.

Ion exchange

The RecoPur system from Eco-Tec Inc. (Pickering, Ont.; www.eco-tec.com) improves the performance of the ion exchange process for treating produced water at oil drilling locations. The system features an ion-exchange resin with finer particles than conventional resins, and resin beds that are

(*Continues on p. 12*)

Note: For more information, circle the 3-digit number on p. 72, or use the website designation.

A step toward the production of fuel for nuclear fusion

Masaru Nakamichi, leader of the Blanket Technology Group, Naka Fusion Institute, Japan Atomic Energy Agency (JAEA, Ibaraki, Japan; www.naka.jaea.go.jp) has developed a new fabrication technology that enables the mass-production of beryllium intermetallic compounds (beryllides), which are advanced, efficient neutron multipliers that could be used for nuclear fusion reactors. JEAE is said to be the first to fabricate beryllides with 1-mm dia., and it expects the newly developed technology to be a big step toward establishing fuel-production technology for the nuclear-fusion demonstration (DEMO) reactor.

At temperatures higher than 600°C, JAEA has found that the beryllide $Be_{12}Ti$ expands by only 3% compared to 50% expansion by Be itself. Also, hydrogen generation from the reaction of steam at high temperatures — a potential hazard in fusion reactors — is 1,000 times higher with Be compared to $Be_{12}Ti$.

JAEA has optimized the fabrication method at the DEMO Design and R&D Coordination Center of International Fusion Energy Reactor Center (IFERC; Aomori; www.naka.jaea.go.jp), where it has produced ductile and easy-worked beryllides rod, using beryllides raw powders with surfaces cleaned by a plasma sintering method. In the past, such materials prepared by alternative methods have been too fragile and unworkable to make powders. By using the beryllides rod as the electrode in a "rotating electrode" process, researchers have fabricated the world's first beryllides fine particles with 1-mm dia. This achievement will contribute to the fuel production test at the International Thermonuclear Experimental Reactor (ITER; naka-www.jaea.go.jp), and also toward establishing a fuel-production technology for the nuclear fusion DEMO reactor.

The scientists also believe that this technology could be applied for the fabrication of aluminum-based alloys that are lightweight, and heat and wear resistant, for use as mechanical parts in high-performance automobile engines.

(Continued from p. 11)

shorter, allowing higher flowrates in a more compact footprint. RecoPur is designed to efficiently remove calcium and magnesium from produced water at oil drilling sites so that the high-salinity water can be used as feed water for once-through steam generators (OTSG) in steam-assisted gravity drainage (SAGD), and other enhanced oil-recovery techniques. The RecoPur system consists of a strong-acid- and weak-acid-cation resin bed that can reduce total hardness from thousands of ppm to less than 0.1 ppm. The ion-exchange system also significantly reduces the use of salt for regeneration of the resin.

Sans organomercury

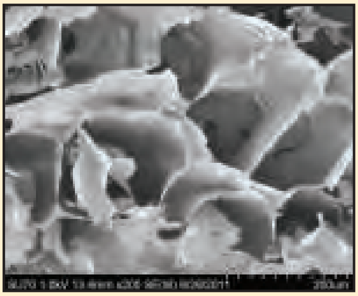
The Dow Chemical Company (Dow; Midland, Mich.; www.dow.com) has voluntarily completed a program to replace mercury-based catalysts used in a product portfolio acquired from another company. The results

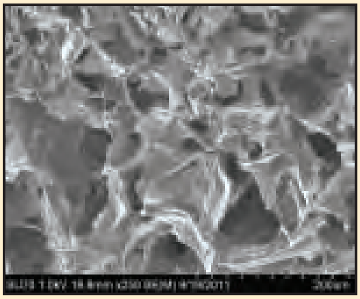
(Continues on p. 14)

Sterile ice fog improves control of freeze-drying

Linde Gases (Murray Hill, N.J.; www.linde-gases.com) has developed a novel cryogenic technology that promotes uniformity in the formation of ice crystals during freeze-drying (lyophilization) processes for proteins, vaccines and other injectable pharmaceutical products.

Lyophilization involves freezing a solution of the product, then reducing the surrounding pressure to allow the water to sublimate. Because of the high purity of the water and clean production environments used in pharmaceutical products, there is a lack of nucleation points for water to begin freezing, explains Prerona Chakravarty, Manager of Linde's project, known as VeriSeq Nucleation. The solutions become supercooled, which makes controlling the formation of ice crystals difficult, she adds. This lack of control lengthens operating cycles and reduces product yields.



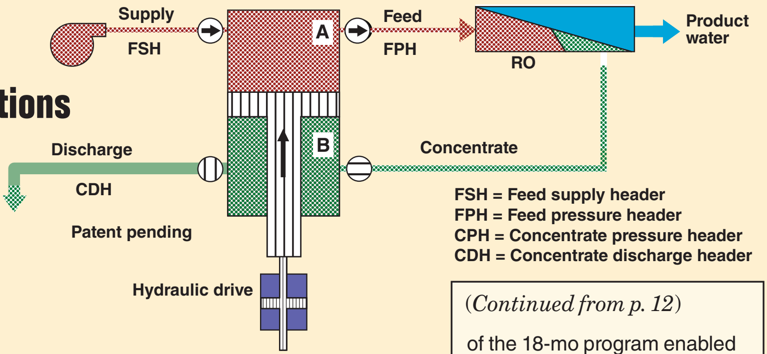


The Linde technology uses a proprietary mixing technique to blend liquid nitrogen with ultra-pure water to create a mist of fine, sterile ice crystals. The "ice fog" rapidly spreads throughout the lyophilizing chamber and causes all vials to freeze at the same time and at the desired temperature. Controlling the temperature at which a vial freezes — the ice nucleation temperature — produces the preferred ice structure within the product, which reduces processing time and minimizes product damage.

This can be seen in the top image, which shows the magnified pore structure of a freeze-dried product subjected to VeriSeq Nucleation. The pores are larger and more regular, which leads to improved drying properties. The image on the bottom shows the pore structure of a freeze-dried product without any nucleation control.

Linde's VeriSeq Nucleation technology can be retrofitted onto existing dryers, as well as incorporated into new installations, Chakravarty says, and initial customer feedback from early demonstrations of the technology, which was unveiled last June at the Achema 2012 tradeshow, has been positive.

This new pump system improves efficiency for desalination operations



A new hydraulically driven, positive-displacement-pump system can lower the energy requirements for pumping water by 10% in large seawater reverse osmosis (SWRO) desalination plants, where electricity typically represents the largest cost.

Recently launched by GE Water & Power (www.ge-energy.com) at the 2012 International Water Week in Singapore, the IPER (integrated pump and energy recovery) system is designed for plants with greater than 1,000-m^3/d capacity, where crank-driven positive displacement pumps are not practical because the crankshaft lengths become prohibitively large. Improvements to the conventional centrifugal pumps used for higher capacity SWRO plants have been able to deliver only incremental energy savings.

In the IPER system (diagram), the crankshaft is eliminated, and the piston is instead moved by a unique hydraulic-drive system that powers three double-acting pistons at much slower cycle speeds than traditional positive displacement pumps. The system consists of a hydraulic-pump-drive unit, a water-displacement cylinder, and a sophisticated control unit.

GE Water & Power has installed the system in a GE owned-and-operated facility in the Caribbean, and is in discussions regarding future installations. The company is also developing larger IPER units, says Erik Hanson, systems product management leader for water and process technologies at GE Power & Water, including one with a capacity of 2,500 m^3/d, anticipated for August 2013 and a 10,000-m^3/d unit in the third quarter of 2014.

(Continued from p. 12)

of the 18-mo program enabled Dow Formulated Systems to remove all organo-mercury catalysts from its polyurethane (PU) elastomer products prior to the E.U.'s formal recommendation to phase out these catalysts, and five years before such compliance is mandated. Dow launched its complete suite of re-engineered PU elastomer products earlier this year. This launch required eight different catalyst packages to replace just one that used a mercury-containing catalyst. The new Hy-

(Continues on p. 16)

Strain-hardening test method speeds HDPE pipe testing

A test method developed by SABIC (Saudi Basic Industries Corp.; Riyadh, Saudi Arabia; www.sabic.com) scientists drastically reduces the time needed to evaluate slow-crack growth resistance in high-density polyethylene (HDPE) that is used in pressure pipes. The strain-hardening test method allows HDPE producers to speed product development and improve quality control by offering an alternative to traditional approaches for collecting data on the long-term slow-crack growth performance of pressure pipes.

Traditional methods, such as the full-notch creep test (FNCT) involve subjecting material samples to constant stress, and crack-inducing liquid and measuring the time to failure, a process that can go on for months. The strain-hardening test consists of a tensile test carried out at 80°C in a few hours. When a polymer sample is highly stretched, it exhibits strain-hardening, a phenomenon that SABIC scientists have correlated with slow-crack growth in HDPE, so that the slope of the stress/strain curve at very high elongation rate — the so-called strain-hardening modulus — can be used to predict its resistance to slow-crack growth.

"The beauty of the method is that it can be carried out with non-specialized equipment right in the plant, rather than having to send samples out for testing at special institutes," says Ralph Handstanger, SABIC technical marketing engineer.

"So far the fundamental connections between strain-hardening and crack propagation have been worked out for the assessment and ranking of HDPE raw materials," Handstanger says. There is potential for the application of this approach to pipes and other areas, however.

This silicone coating keeps electrical insulators safe

Wacker Chemie AG (Munich, Germany; www.wacker.com) has commercialized a solvent-free coating for electrical insulators. Powersil 570 Plus is said to be the world's first insulator coating based on a patented silicone-in-water emulsion technology. Its viscosity displays a strong dependence on its shear rate, which makes it possible for spray applications. The one-component emulsion is applied by spraying and then it cures to form a water-repellent silicone coating. It adheres to ceramic and glass substrates, and passes the 1,000-h salt-fog test per IEC 62217.

Porcelain and glass are commonly used to insulate overhead powerlines, but in coastal and industrial areas, salt deposits and dirt can impair the insulating properties, which can lead to electrical discharges or so-called pollution flashovers. To avoid such scenarios, insulators must be cleaned regularly. Silicone coatings maintain the electrical insulating properties, and their hydrophobicity prevents the formation of "wet pollution layers." Thus, the typical flashover scenario — wet film layer with an increased leakage current, dry band arcing, bridging or individual insulator sheds and finally the electrical flashover — is impossible, says Wacker.

This osmium-based catalyst is rendered nontoxic by polymer incarceration

Professor Shu Kobayashi and his research group at The School of Science, University of Tokyo (Japan; www.chem.s.u-tokyo.ac.jp/users/synorg/index_E.html) have developed an osmium-based catalyst for performing asymmetric syntheses of drug precursors. While showing the same catalytic performance of its predecessor — a microencapsulated osmium oxide in a polymer — the new catalyst system is not dissolved by solvents, which makes it possible to recover the catalyst for reuse. The new, so-called polymer-incarcerated osmium (PIOS) system is nontoxic, nonvolatile and stable for months in air.

The catalyst is made by mixing OsO_4 in a styrene-based polymer solution for 72 h. Hexane is then added, which causes the OsO_4 to become microencapsulated by the polymer. After removing the solvent, the solids are heated to 110°C, which crosslinks the polymer into a matrix in which the OsO_4 is bound.

The catalyst enables a one-step reaction process instead of the three or four steps required by conventional catalyst technologies. For example, the chemists demonstrated that the PIOS system catalyzes the asymmetric dihydroxylation of alkenes into the corresponding diols with high yields and high enantioselectivities. They also prepared 1-mol quantities of a key intermediate for camptothecin (an anticancer drug) with 97% yield with the catalyst. They also confirmed that osmium was not detected to be absorbed in the organs of PIOS-dosed mice in acute toxicity tests.

The researchers are expecting that their achievement on immobilization and stabilization of a toxic system could be applied to other catalyst systems that show high activity but also suffer due to high toxicity.

A move toward bio-based malic acid . . .

Novozymes A/S (Copenhagen, Denmark; www.novozymes.com) has developed a strain of metabolically engineered *Aspergillus oryzae* fungus that is capable of producing the four-carbon, dicarboxylic acid compound malic acid. The chemical building block malic acid is used as a flavor enhancer in the food industry, and can be converted into 1,4-butanediol (BDO), which serves as a precursor to many other chemical derivatives used in resins and polymers. Developed in partnership with Archer Daniels Midland Co. (ADM; Decatur, Ill.; www.adm.com), the robust fungus strain generates high yields of malic acid and has the potential to achieve production economics similar to petroleum-derived malic acid, says Novozymes vice president Rasmus von Gottberg. Novozymes is looking to out-license the fungus strain and associated technology to partners who would commercialize the bio-based malic acid and its derivatives.

. . . and acrylic acid . . .

Meanwhile, BASF SE (Ludwigshafen, Germany; www.basf.com), Cargill (Minneapolis, Minn.; www.cargill.com) and Novozymes signed a joint R&D agreement to develop an industrial biotechnology-based production process for acrylic acid. Since 2008, Novozymes and Cargill have collaborated to develop microorganisms that convert renewable feedstock into 3-hydroxypropionic acid (3-HP), which is a possible precursor to acrylic acid. BASF has now joined forces to develop a process to convert 3-HP into acrylic acid. BASF plans to first use the bio-based acrylic acid to manufacture superabsorbent polymers, which are used in diapers and other hygiene products.

. . . and butadiene

Also last month, Invista (Wichita, Kan.; www.invista.com) and LanzaTech (Roselle, Ill.; www.lanzatech.com) signed a joint development agreement focused on bio-based butadiene. According to the agreement, Invista and LanzaTech will collaborate on projects to develop one- and two-step technologies to convert carbon monoxide into butadiene.

Initial commercialization is expected in 2016. The collaboration will initially focus on the production of butadiene in a two-step process from LanzaTech — CO-derived 2,3-butanediol (2,3 BDO). A direct single-step process will also be developed to produce butadiene directly through a process of gas fermentation. ■

(*Continued from p. 14*)

perplast and Diprane products ensure that users can already be confident to have non-mercury catalysts that comply with future REACH requirements, says the company.

Phenol recovery

Last month, Rhein Chemie Rheinau GmbH (Mannheim, Germany; rheinchemie.com) was awarded first prize in this year's Responsible Care competition — held by the Baden-Wuerttemberg branch of the German Chemical Industry Assn. (VCI; Frankfurt am Main, Germany; www.vci.de) — for a process to recover phenol from mixed wastewater streams generated in the manufacture of a plastics additive. The process removes phenol and recycles it back to the production process, thereby reducing the amount of phenol to be disposed of by around 150 metric tons per year.

Phosphorus recovery

Researchers from the Fraunhofer Institute for Interfacial Engineering and Biotechnology (IGB; Stuttgart, Germany; www.igb.fraunhofer.de) are developing a patented process for the recovery of struvite (magnesium-ammonium phosphate) from sewage sludge. The electrochemical process precipitates struvite (from a solution containing nitrogen and phosphorus) as crystals that can be directly used as a fertilizer, without the need for further processing.

The process is now being tested in a mobile pilot plant, which features a 2-m tall electrolytic cell with a sacrificial magnesium anode and a metallic cathode. Water is split into OH^- at the cathode, and oxidation takes place at the anode to form Mg^{+2} ions, which migrate through the water and react with PO_4^{-3} and NH_4^+ to form struvite.

So far, the required power for the process has never exceeded 70 W-h/m^3 for all types of wastewater tested, and the researchers have demonstrated that the phosphorous concentration is reduced by 99.7% to less than 2 mg/L, which is lower than the maximum allowable concentration permitted by the German Waste Water Ordinance (AbwV). ❑

Newsbriefs

Cheaper chemicals from algae, farms & forest may be possible, report says

The bio-based materials and chemicals industry needs to tap newer, non-food sources of biomass and cellulosic material and raise volumes of feedstock before it can emerge as an economically viable alternative to petroleum-based products, according to a Lux Research (Boston, Mass.; www.luxresearchinc.com) report. Currently, the high cost of capital and operations limit bio-based materials and chemicals to a few facilities located where corn and sugarcane are plentiful and inexpensive.

"Bio-based materials and chemicals manufacturers need syngas [synthesis gas] and sugar to fuel their growth. Gasification and enzymatic hydrolysis are key technologies for securing vast amounts," says Mark Bunger, research director and lead author of the report, "Pruning the Cost of Bio-Based Materials and Chemicals."

Lux Research analysts studied cost drivers in gasification, enzymatic hydrolysis of cellulose and algae cultivation to find opportunities where new technologies can turn them to profit. Among their findings:

- Algae remains a cost-intensive loser. In Lux Research's model, algae cultivation yields a 48% loss, calling into question its long-term prospects. The problem lies in the high capital costs for growing algae at industrial scale
- Syngas fermentation has great new product potential. The many products of syngas fermentation proven at laboratory scale, or larger, include ethanol, butanol, acetic acid, butyric acid, 2,3-butanediol and methane. Leading startups in this domain include ZeaChem, which is collaborating with Procter & Gamble, and LanzaTech
- There is hope for cellulosic biomass, but costs need to fall further. Enzymatic hydrolysis is being commercialized at new facilities like GraalBio's plant in Brazil, using the latest enzymes from Novozymes and DSM. However, many parts of the process need to improve, including harvesting and collecting biomass, which adds $15/ton, or $0.21/gal, in costs

CHEMICAL PRODUCTION SEES DECLINE

Overall production of chemicals in the U.S. fell by 0.1% to 86.6% of its 2007 level during July, but the overall decline masked gains in specialty and bulk chemicals, according to data from the U.S. Federal Reserve Board that was analyzed in the August 17 Weekly Chemistry and Economic Report from the American Chemistry Council (ACC; Washington, D.C.; www.americanchemistry.com). U.S. specialty chemical production was up 0.4% in July, said the report, while basic chemical production gained 0.2%. Production fell in agricultural chemicals and consumer products.

EPA identifies substitutes for toxic flame-retardant chemical

In an effort to identify possible substitutes for a toxic flame-retardant chemical, known as decabromodiphenyl ether (decaBDE), the U.S. Environmental Protection Agency (EPA; Washington, D.C.; www.epa.gov) has released a draft report on alternatives. This assessment, developed under EPA's Design for the Environment (DfE) program, profiles the environmental and human health hazards on 30 alternatives to decaBDE, which will be phased out of production by December 2013.

DecaBDE is a common flame retardant used in electronics, vehicles and building materials that can bio-accumulate in humans and animals. The technical assessment can help manufacturers identify alternatives to decaBDE. This draft report is the latest in a series of actions the agency is taking to address flame retardants made with bromine.

CSB deploys to fire and explosion at Chevron refinery

A seven-member investigative team from the U.S. Chemical Safety Board (CSB; www.csb.gov) was deployed to the scene of an explosion and fire that occurred August 6 at the Chevron Refinery in Richmond, Calif. The team is headed by Donald Holmstrom, director of the CSB's Western Regional Office in Denver. Board member Mark Griffon accompanied the team. According to company officials, the explosion and fire involved the release of vapor that found an ignition source. Four workers sustained minor injures.

CSB chairman Rafael Moure-Eraso said, "CSB investigations examine a wide range of safety issues such as effective process safety management and mechanical integrity."

Progress on a waste-to-fuels process reported at ACS meeting

A process for converting municipal waste, algae, cornstalks and similar material to gasoline, diesel and jet fuel shows promise in larger plants, the developers reported at the National Meeting & Exposition of the American Chemical Society (ACS; Washington, D.C.; www.acs.org), which took place in Philadelphia from August 19–23.

The technology, termed integrated hydropyrolysis and hydroconversion (IH^2), was developed by the Gas Technology Institute (GTI; Des Plaines, Ill.; www.gastechnology.org). GTI scientist Martin Linck anticipates multiple demonstration-scale facilities in operation by 2014.

IH^2 technology involves the use of internally generated hydrogen and a series of proprietary catalysts, which jump-start chemical reactions that otherwise would happen slowly or not at all.

Linck said it differs from other biofuel technologies in that it produces a finished, ready-to-use liquid hydrocarbon fuel, rather than crude intermediate substances or substances that contain oxygen, which must be processed further.

CHEMICAL PLANT SECURITY: GATING MORE THAN THE PERIMETER

While fences and locks have been helping to secure the CPI for some time, the vulnerability of industrial control systems has only recently been demonstrated in ways that make everyone take notice. Cyber and physical security are now considered interdependent

Traklok

FIGURE 1. TrakLok integrated the physical security of a lock with a sophisticated system that monitors the security of cargo and instantaneously reports security breaches via cellular and satellite networks

It was clear at the recent Chemical Sector Security Summit (Baltimore, Md., July 31-Aug. 1) that awareness of and activity in security for the chemical process industries (CPI) are growing. The annual summit, now in its sixth year, was held at its largest venue ever and brought together about 650 participants from industry, academia and the government to network with other security professionals, share best practices and learn more about chemical security regulations.

One of the strong messages from this summit, as well as from additional sources, is that more attention than ever before is being given to cybersecurity of industrial control systems. In fact, in her introductory comments at the summit Suzanne Spaulding, deputy undersecretary for the National Programs and Protection Directorate (NPPD) for the U.S. Dept. of Homeland Security (DHS; Washington, D.C.; www.dhs.gov) emphasized an increased awareness of the interdependence between cyber and physical security and said that the "chemical sector has been a leader to bring together those two disciplines."

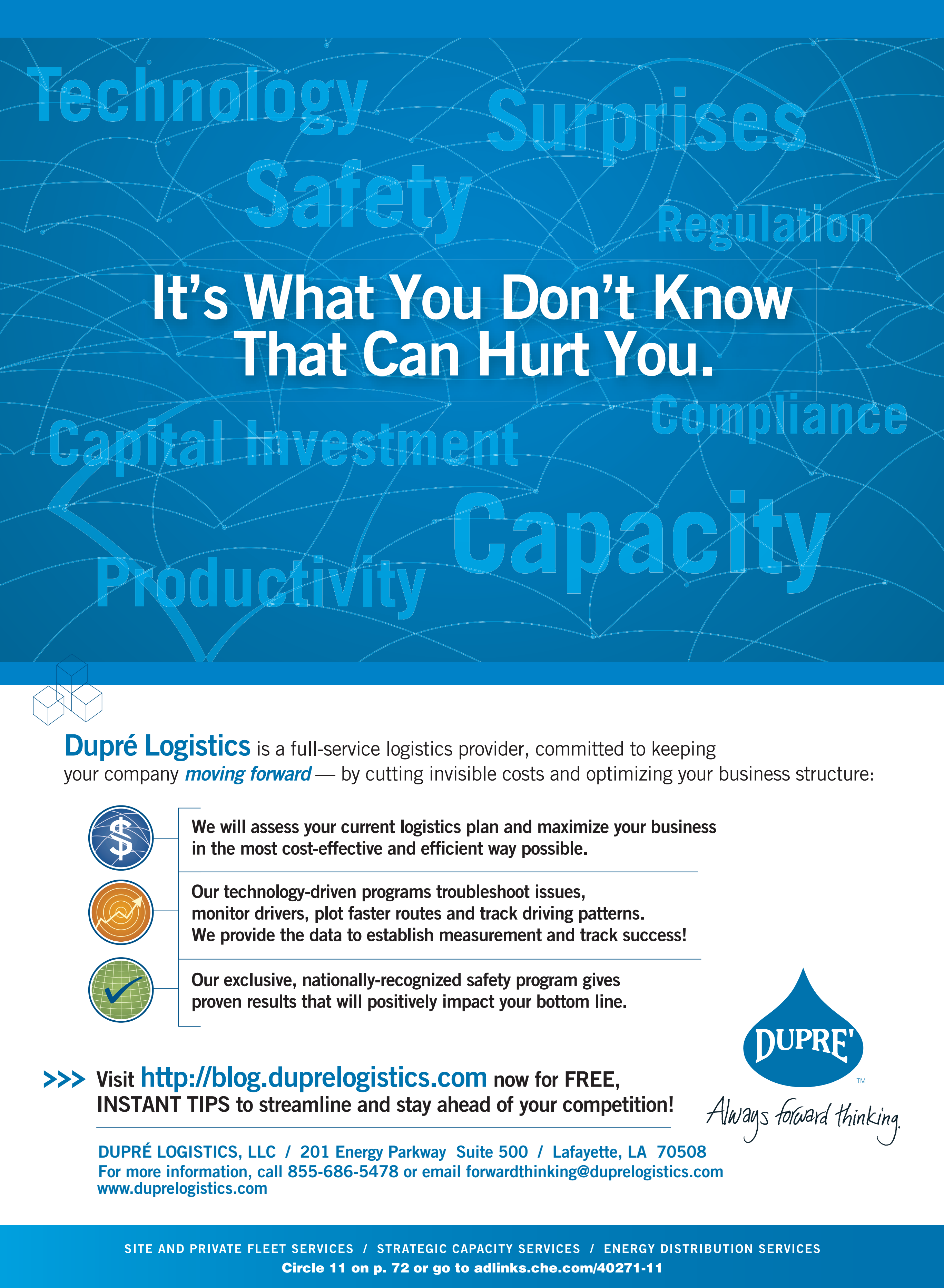
Meanwhile, in his keynote address at the summit, undersecretary for the NPPD, Rand Beers, spoke of a new appreciation of owners and operators across all sectors, including the chemical sector, for the need for cybersecurity. He said that in the larger picture, security requires teamwork — across the government, private sector and the public.

That type of teamwork was evident in the mix of participants at the summit. Evidence is also seen in a number of working groups that are collaborative efforts to reduce the risks to industrial control systems. A good example of this is the Industrial Control Systems Joint Working Group (ICSJWG; www.us-cert.gov/control_systems/icsjwg/).

CFATS

The CPI are considered part of the critical infrastructure in the U.S., and as such are regulated by the DHS under the Chemical Facility Anti-Terrorism Standards (CFATS; for details on CFATS see Chemical Plant Security, *Chem. Eng.*, pp. 21–23, September 2009).

While CFATS continues to be a driver for much of the heightened activity in security, the program itself has recently been confronted by serious challenges. Last year at this time, with strong support in both the House and the Senate, CFATS was poised to be reauthorized by Congress for multiple years (as opposed to the annual reauthorization it had been receiving). However, an internal assessment of the CFATS program in the last months of 2011 brought to light some serious concerns about the management and implementation of the program. A multi-year reauthorization now seems to be unlikely in the foreseeable fu-


Technology
Surprises
Safety
Regulation
It's What You Don't Know
That Can Hurt You.
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Capital Investment
Capacity
Productivity
Dupré Logistics is a full-service logistics provider, committed to keeping your company moving forward — by cutting invisible costs and optimizing your business structure:
We will assess your current logistics plan and maximize your business in the most cost-effective and efficient way possible.
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ture. While authorization inspections have resumed as of July 2012, the program is currently facing the possibility of drastic funding cuts.

During a "Congressional Perspectives" panel at the Security Summit, Monica Sanders, council for the House Committee on Homeland Security, shed some light on the CFATS situation. She explained that with a new program like CFATS, you want to have one-year authorizations until the program is "mature", at which time it is extended for multiple years. "Congress thought that CFATS was at that point, but the late 2011 memo was a game changer", she said, and heavy Congressional oversight is still needed on the CFATS program.

The panel members, which also included Chris Schepis, senior professional staff member for the House Committee on Homeland Security; and Jerry Couri, senior professional staff member for the House Committee on Energy and Commerce, seemed to agree that they expected CFATS would face a budget cut. The panel also commented, however, that Congress wants to see CFATS succeed as the kind of program envisioned in 2006 when it was first authorized.

While CFATS has been a driver for much of the heightened activity in security over the past few years, the goal of securing the CPI is not dependent on the standard. As Lawrence Sloan, president and CEO of the Society of Chemical Manufacturers & Affiliates (SOCMA; Washington, D.C.; www.socma.com) said in his opening remarks at the summit "… the reason we meet here is not merely because of the presence of a regulation, but because of a need for mutual collaboration ... industry's commitment to securing its most hazardous chemical products is stronger than ever, as evidenced by the overwhelming presence of so many attendees here today from the private sector. Others who are here who may not fall under CFATS or the Coast Guard's MTSA [Maritime Transportation Security Act] program are also here to learn because, whether you are regulated or not, securing your assets and the communities in which you operate makes good business sense."

Physical security

"Perimeter security is the first line of defense," says Ryan Loughin, director of Petrochemical & Energy Solutions at Tyco Integrated Security (TycoIS; Boca Raton, Fla.; www.tycois.com), "and Tyco Integrated Security offers many solutions including advanced fiber-based technology, thermal imaging/infrared, analytics software and remote video products that help satisfy the CFATS Risk-Based Performance Standards".

Indeed, most of CFATS addresses the critical need to physically secure chemical plants, with one of the major goals being to prevent theft and diversion. Diversion is a form of misappropriation, and refers to the acquisition of a material by means of deception, by someone who should not possess the material. While numerous forms of protection to avoid theft and diversion are employed at CPI sites, Tom Mann, CEO of TrakLok Corp. (Knoxville, Tenn.; www.traklok.com) sees a vulnerability during transportation. "Manufacturers of high-value and high-risk products have enormous security at their facilities," says Mann, "However, when they ship product, they have had no other options than a padlock and a tracking device."

Mann's company, TrakLok, has introduced a new truck and container security device (Figure 1) that combines physical security with realtime global monitoring and notification. The lock integrates the physical security of a lock with a sophisticated system that monitors the security of cargo and instantaneously reports security breaches via cellular and satellite networks. TrakLok also offers a geofencing capability prohibiting the lock from opening if it is outside a prescribed area.

Cybersecurity

"Adversaries are becoming more stealth," said Lisa Kaiser, operations lead, CyberSecurity Implementation, ICS-CERT, Office of Cybersecurity and Communications for the DHS, at the 2012 Chemical Sector Security Summit. In addition to the advanced complexity of threats, Kaiser said that there has been more than a 400% increase in the number of reported cyber-related incidents from 2010 to 2011. This statistic is from the Industrial Control Systems Cyber Emergency Response Team (ICS-CERT) Incidence Response Summary Report[1], which shows that the number of reported and identified incidents impacting organizations that own and operate control systems associated with critical infrastructure rose from 41 in 2010 to 198 in 2011.

The threats. Kaiser cited an increased use of so-called spear-phishing involving major U.S. corporations including oil-and-gas, and water companies. Spear-phishing is a technique where malware-containing files are attached to very credible-looking emails, often seemingly from trusted sources. Clicking on the attachment unleashes the malware, which then opens the gates to the sender to access the computer system of the unsuspecting victim.

While most of the spear-phishing to date has been directed at information gathering, which in itself is reason to defend against it, the threat of being able to do more, such as take control of an industrial control system (ICS) is real and has been demonstrated.

Most users of ICSs are now aware of the Stuxnet worm, which was a real game changer for ICS security in that it was highly sophisticated and deliberately targeted to attack a

1. Available at http://www.us-cert.gov/control_systems/pdf/ICS-CERT_Incident_Response_Summary_Report_09_11.pdf

control system (for more on Stuxnet, see Securing the CPI, *Chem. Eng.*, pp. 17–20, September 2010; and Cybersecurity for Chemical Engineers, *Chem. Eng.*, pp. 49–53, June 2011). New Stuxnet-like malware has since appeared, including Duqu, identified in October 2011 and Flame, identified more recently this year.

Even if CPI control systems are not specifically targeted by this type of malware, "the fact that Stuxnet was wildly successful opens the door to collateral damage," says Rick Kaun, business manager for the Industrial IT Solutions Group of Honeywell Process solutions (Phoenix, Ariz.; www.becybersecure.com). In addition to spear-phishing through social media like emails, Kaun says that the use of USB sticks and other devices with unknown content in work systems is also a big concern for ICS.

Andrew Ginter, director of Industrial Security at Waterfall Security Solutions (Calgary, Alberta, Canada; www.waterfall-security.com) echoes the concerns about collateral damage saying, "Thus far the intent [of the malware] is industrial espionage, but it is clear that they are capable of much more." Industrial practitioners should not feel "secure" because they cannot point to a specific incident of harmful control, says Ginter. He explains further that if someone who shouldn't have access to your control system does, that in itself is unacceptable. The ability for unauthorized remote control has been demonstrated and is a threat. Even without malicious intent, an uninformed hacker who has control can do damage.

Security experts. The suppliers of ICSs and security systems for them are continuously working on their arsenals to help prevent against cybersecurity threats.

Waterfall's Ginter says that one approach to address advanced threats is to use hardware-enforced one-directional communication (Figure 2) so that information can be shared out of the industrial control network to the business side, but no information can get back to the process control side. Oftentimes ICSs are connected to business networks to help track raw-material and product inventories, schedule maintenance and replacement part ordering and more. Advanced threats can, however, penetrate the firewalls that are used between the business and control systems. Unidirectional gateways are a hardware and software solution that permit the flow of information out of isolated networks, but not back into them.

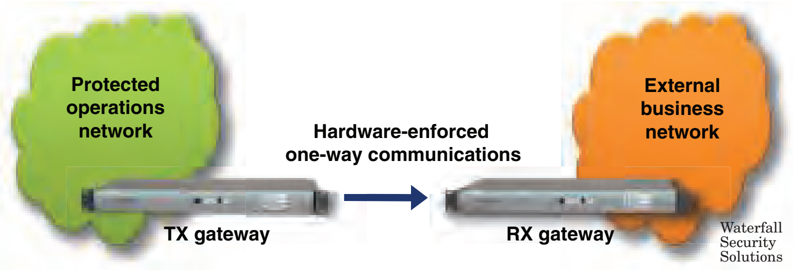

FIGURE 2. Unidirectional communication prevents communication from the business network back to the process control network

Another tool that has been used in business IT environments but is relatively new to automation control systems is "whitelisting". Honeywell's Kaun explains that whitelisting means that "only predefined applications and processes are allowed on the ICS systems". It protects from unwanted intrusions by allowing only those applications and files that have been placed on an approved list, and everything else is blocked. Anti-virus (AV) solutions on the other hand, are referred to as "blacklisting", since they operate differently by blocking what is already known to be bad, but otherwise allowing everything else. Honeywell now offers a fully certified application and whitelisting solution to help combat malware from attacking its Experion Process Knowledge System.

"The end users of large companies are asking us to implement more security, including AV, whitelisting, firewalls etc.," says Graham Speake, principal systems architect for Yokogawa Corp. of America's (Newnan, Ga.; www.yokogawa.com/us) IA Global Strategic Technology Marketing Center in Dallas, Tex. "Some are also looking to have some or all of these services outsourced and managed by the vendor," he says. Notifications of patches for various systems from various vendors can be confusing for end users to coordinate and implement correctly. Part of this, says Speake, is due to the fact that there has not been a definition of what to put into a "patch". There is, however, an ISA99 (International Society of Automation; Research Triangle Park, N.C.; www.isa.org) working group that is addressing this need and preparing a draft standard for patch management in the ICS environment.

Speake says that Yokogawa offers users a field service where they will come in and implement upgrades and patches on a regular basis (quarterly, for example). While not yet widespread, this service is particularly useful for small-to-medium size companies who may not have the resources to keep their systems up-to-date. As security measures become more complex, more support will likely be needed.

No quick fixes. Security experts agree that the solutions to help thwart cybersecurity threats are not quick fixes. And, that while awareness of the vulnerabilities and threats is increasing, there is still a way to go. Looking forward, Yokogawa's Speake says "the drive for products and devices that have security designed and built into them will become more intense. It is easier for vendors to build security in than for end users to add it on afterwards. It is the vendors' responsibility to enable this security, and then the end user must take responsibility for keeping these products up-to-date and provide or outsource the services needed to monitor the security devices."

"It [security] is a *discipline* much like safety and maintenance," emphasizes Honeywell's Kaun. The day he drives up to a plant and sees a sign that reads "So many days since a cybersecurity incident" next to the sign that says "So many days since a safety incident" is when he will feel that we have reached a good level of awareness. ■

Dorothy Lozowski

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Equipment News Roundup

AN OPEN AND SHUT CASE

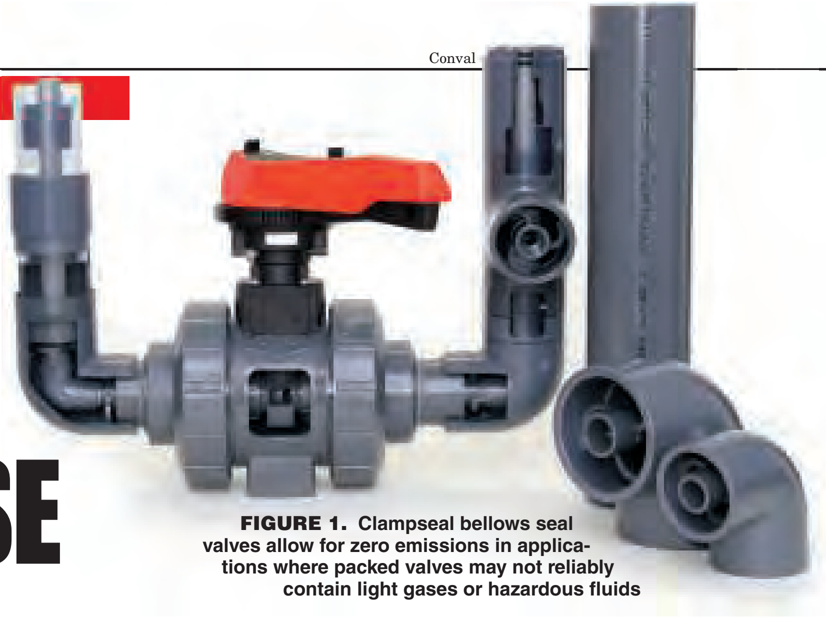

Conval

FIGURE 1. Clampseal bellows seal valves allow for zero emissions in applications where packed valves may not reliably contain light gases or hazardous fluids

Improvements in valve technology allow processors to improve performance while reducing leaks

Though times are changing, many of the traditional challenges associated with running a chemical processing plant remain the same. For years processors have been asking for leak-proof, high-performance valves, but it is only recently that valve manufacturers are employing new technologies, designs and materials that improve performance and reduce leaks.

Improved C_v values

"Traditionally, processors running batches of chemicals or feeds used ball valves because they typically offered the best C_v [valve flow coefficient] values without pressure drop," says Dave Vollaire, product manager for valves and actuation with GF Piping Systems (Tustin, Calif.; www.piping.georgfischer.com) "However, one of the drawbacks of ball valves occurs when the line goes dry or chemical is allowed to crystallize. This can jam up the ball valves and cause damage to stems and seats and lead to leakage."

To combat this issue, GF Piping now offers a line of diaphragm valves, the Georg Fischer 5 Series manual and pneumatic diaphragm valves, which feature a central union nut, unlike conventional valves that include four metal screws to hold the top assembly on the valve body. This non-corrosive connection is characterized by homogeneous temperature behavior, even surface pressure and a high-pressure rating. In addition, the flow geometry offers double the C_v values.

Another way to increase C_v value while decreasing leaks is to replace ball valves with a newer technology. For example, GF Piping also offers a new-generation, cone check valve with a spherical shaped design that offers improved flow characteristics and leak-proof operating performance. The aerodynamic, spherical-shaped cone ensures tight closing and 100% leak-proof operation, says the company, while providing improved C_v values over standard ball-type check valves.

An added bonus of this, says Vollaire, is that while most ball valves require about 10 diameters of straight run pipe after a flow disturbance (like an elbow or pump), this model requires half that, which allows use in smaller areas. "Users don't need as many straight runs, and this is a big benefit because typically, in a pipe run, it's difficult to find anywhere that is clear and free of obstruction," he says.

Reduced leaks

In most situations in the chemical, petrochemical and petroleum-refining industries, leaks are to be avoided at any and all costs, which is what makes packing integrity so important, says Michael Hendrick, vice president of sales and marketing with Conval (Somers, Conn.; www.conval.com). "We make heavy process-industry valves for refineries and large plants that are scrutinized by the Environmental Protection Agency," he says. "So this is a big, big issue for our customers and for us."

Conval started using graphite packings some time ago because the tem-

A NOD TO AUTOMATION

In addition to making leak-free, high performance valves, industry experts say actuation is growing considerably. "I believe the increase in actuation is related to labor savings in that by automating the systems, users are able to see, from a remote monitoring station, what's going on with the valves — whether they're engaged, on or off, having issues and so on," says Dave Stewart, director of marketing with Hayward Flow Control (Clemmons, N.C.). "In a vast refinery or chemical plant that is not automated, this involves a lot of time spent getting to valves in remote areas, but by automating them you save a lot of time. Actuators serve as a way to monitor the status of the valves in a facility and, as more plants and refineries are being updated, actuation is being added."

GF Piping System's Dave Vollaire agrees: "Automation is on the rise and we see people moving away from manual systems and using positioners and more feedback devices that reveal the actual position of valves in a system." ❑

FIGURE 2. **The Double-See vinyl piping system provides a pressure-rated double-containment system for transporting hazardous liquids**

perature limitations of plastics and rubbers are more restrictive, so the new and improved graphite packings work better, last longer and resist pitting, making them more suitable for critical applications with temperatures over 500°F.

However, to provide even more leak-proof capability, Conval developed the Clampseal Packing System (Figure 1), which puts compression on the packing in an effort to maximize packing life in the demanding high-pressure and high-temperature applications.

The uniform, single-piece gland allows the graphite packing to be loaded uniformly with a one-piece gland, which eliminates the potential for stem damage from gland cocking, while the integral gland wrench provides immediate gland/packing adjustment capability. "This feature makes it easier to adjust the packing and gives more uniform compression of the packing, which helps reduce leaks," says Hendrick.

The Clampseal technology can be found on many of the company's valves, including the new zero-emissions bellows seal valves, which are suitable for applications where packed valves may not reliably contain light gases or hazardous system fluids, due to leakage in the stem/packing seal or stuffing box, wall/packing seal area.

Also to prevent leaks of hazardous liquids, many facilities are double containing their valve and piping systems, says Vollaire. "This means they run the main carrier line of chemical, and around that line there is a second pipe," Vollaire says. "In our case, we put an outer valve, like a dummy valve that throttles and closes the valve inside that is carrying the chemical. This type of double containment is especially helpful for lines that carry chemicals overhead, but have the potential to leak, because the secondary pipe contains the chemical leak from falling down."

GF's Double-See (Figure 2) is used as a pressure-rated, double containment system and offers versatile installation options, assembled and tested fittings and a pipe cut-length guidance system that provides a simplified approach to installation. The valve-in-valve design allows full-containment-pressure ratings. ❑

MORE VALVE PRODUCTS

Wafer check valves for critical water applications

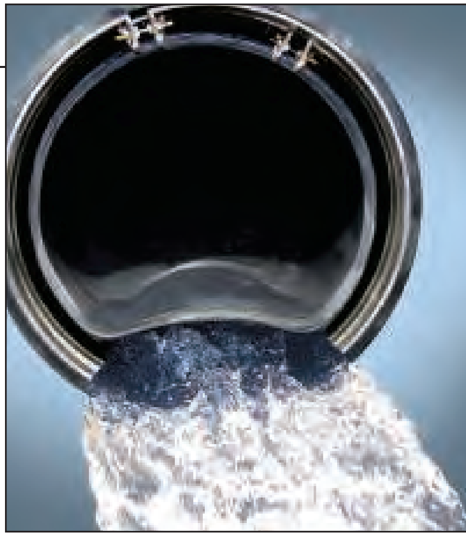
These 10- and 12-in. PVC wafer check valves (photo) are installed on the discharge side of pumps to prevent backflow flooding that could damage pump systems. The body, disc and stopper assembly are machined from solid PVC plate stock, which conforms to ASTM D1784 Cell Classification 12454A. This material offers chemical resistance. The valve is easily installed by slip fitting the valve between two mating flanges. The valve body automatically centers on the mating flanges, once the stud pack is installed. A directional flow arrow on the valve body indicates the upstream and downstream sides of the valve. The lack of bolt pattern allows one valve style to accommodate many mating flange dimensions. — *Asahi/America, Inc., Malden, Mass.*
www.asahi-america.com

Asahi/America

An inline check valve for backflow prevention

The CheckMate's (photo) custom-engineered, all-rubber, uni-body design eliminates backflow, while the elastomer-fabric reinforced-design reduces maintenance. The valve offers low head loss and can open to a near-full pipe diameter to maximize the flow capacity of the outfall. The valves are available in 4–72-in. sizes. — *Tideflex Technologies, a Division of Red Valve Co., Inc., Carnegie, Pa.*
www.tideflex.com

Red Valve



Hand valves and manifolds for instrumentation applications

Hand valves and two-, three- and five-valve manifolds in carbon steel (photo, p. 27) provide a solution for natural gas

Parker Hannifin

Hayward Flow Control

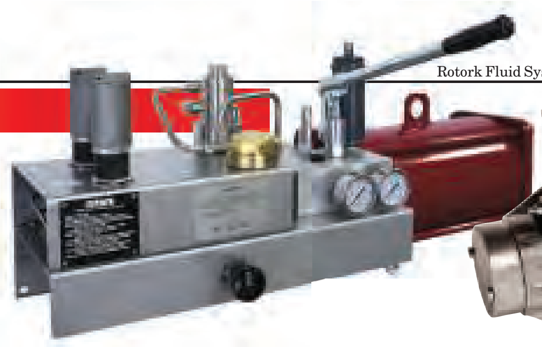
transmission line's, and other chemical process industries' instrumentation applications that don't require the extreme performance of stainless steel. The products are pressure rated up to 6,000 psig (414 bar) to provide system flexibility when selecting process valves. Designed to work with a variety of fittings, the valves and manifolds feature an external adjustable gland that allows for full adjustment of the stem packing without removal of the valve from the system. Handles are color-coded to quickly and easily identify valve functions. — *Parker Hannifin Corp., Huntsville, Ala.*
www.parker.com

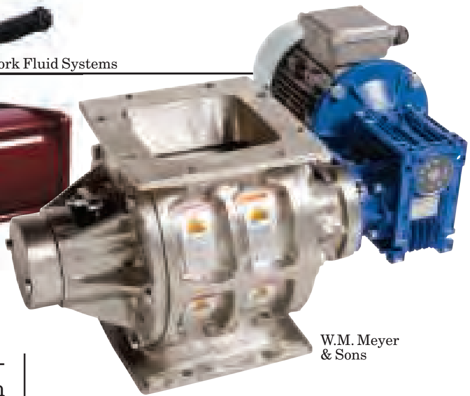
Glass-filled PP makes this valve more durable

The platinum GF-PP (glass-filled polypropylene) TB Series ball valves (photo) feature a full port design with true-union threaded or flanged end connections and reversible seats. The valves have a maximum pressure rating of 250 or 150 psi with threaded or flanged ends, and a maximum service temperature of 240°F. The platinum GF-PP material provides high impact resistance and is suitable for more robust chemical services, abrasive applications and water distribution. Features include an adjustable seat retainer, double O-ring stems and easy actuation. Options include handle lockouts, 2-in. square operating nut, stem extensions, pneumatic or electric actuation, manual limit switches and spring return handle. — *Hayward Flow Control, a Division of Hayward Industries, Clemmons, N.C.*
www.haywardflowcontrol.com

Improve safety and reliability with this ball valve

The KTM Unibody EB700 Series ball valve is designed and certified to meet the standards and service requirements of the petrochemical, chemical and oil-and-gas industries through improved pipeline integrity, performance and safety characteristics. The one-piece, cast body minimizes potential leak paths, making it safer and more environmentally friendly than split body designs. With dual body-insert seals and shaft seals, this valve


Rotork Fluid Systems


W.M. Meyer & Sons

provides sealing integrity, says the manufacturer. The valves feature a PTFE/PFA co-polymer "E" seat. The valves in sizes above DN25 (NPS 1) also provide an additional safety feature incorporating cavity self-relieving seats to relieve potential ball cavity over-pressure when the valve is in the closed position. — *Tyco Flow Control, Princeton, N.J.*
www.tycoflowcontrol.com

This dust-duty valve creates a quality air seal

The Dust Duty Valve (DDV; photo) comes with a robust cast housing designed for greater capacity and is suitable for dust collection. It will service applications where a quality air seal is necessary in light-duty, non-abrasive applications under baghouse, cyclone and dust collections with a maximum pressure differential of 60 in. wc.. The maximum temperature is 300°F. The valve is available in cast-iron or stainless-steel construction with square or round flanges in sizes 6–14 in. — *W.M. Meyer & Sons, Inc., Libertyville, Ill.*
www.wmwmeyer.com

Spring-return actuators for infrequently operated valves

The ManPower range of products (photo) reduces installation time and expense since no electrical-, hydraulic- or pneumatic-power supply is required. Suitable for a range of ball, butterfly and plug valves, the scotch-yoke actuators are equipped with a compact, self-contained, manually operated hydraulic-power pack. A hydraulic hand pump on the power pack is used to operate the actuator and compress the failsafe spring, holding the valve in the desired open or closed position until a failsafe signal is received, at which point the spring will immediately drive the valve to the safe position. Failsafe operation can be triggered by electrical signals, high- or low-pressure pilots or fire sensors, enabling the successful fulfillment of many ESD (emergency shutdown) and pressure-related protection duties in the oil, gas, petrochemical and pipeline industries. — *Rotork Fluid Systems, U.K.*
www.rotork.com ■

Joy LePree

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Emerson Process Management

FOCUS ON
Flowmeters

McCrometer

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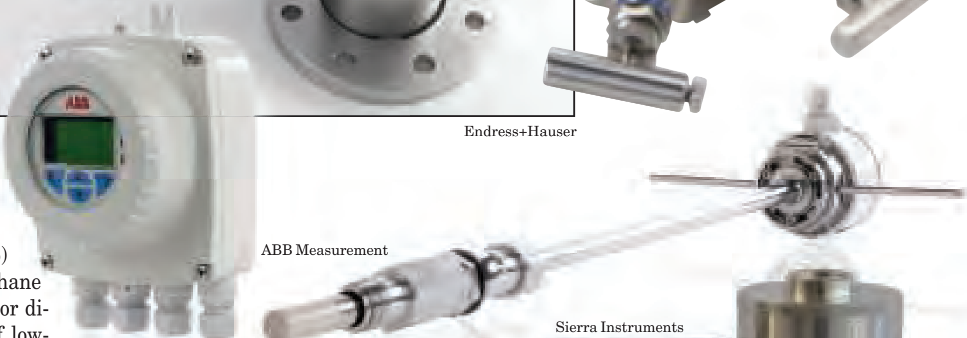


ABB Measurement

Sierra Instruments

A flowmeter designed for biogas

The Proline Prosonic B200 ultrasonic flowmeter (photo) measures flowrate and methane content of wet biogas, landfill or digester gas, and other types of low-pressure, wet or contaminated gas. The company has optimized transit-time ultrasonic technology for biogas applications. The B200's body is made from 316L stainless steel and is suitable for wet, dirty or corrosive gases. The flowmeter is available with either an aluminum or stainless-steel transmitter housing. The instrument can also measure and display direct methane fraction as a function of sound velocity and temperature, without the need for other instruments. The B200 operates in temperatures of 32 to 176°F and pressures up to 145 psi. The accuracy of its volume flow is 1.5% of the reading. — *Endress+Hauser Inc., Greenwood, Ind.*
www.us.endress.com

A wide choice of power solutions for this flowmeter

The AquaProbe insertion electromagnetic flowmeter range (photo) now has an extended range of power source and transmitter options. The AquaProbe series is designed to be used with this company's WaterMaster and AquaMaster 3 transmitters, and can now be powered with main and battery power, as well as renewable power. The new, renewable power version can draw power from either solar or wind sources. The enhancements make it ideal for clean water applications, the company says. The unit also features intelligent installation technology that performs a self-configuration sequence automatically on initial installation. — *ABB Measurement Products, Warminster, Pa.*
www.abb.com/measurement

Minimize pressure drop with this mass flow device

The SmartTrak 140 (photo) is a mass flow controller for high-performance mass-flow-control applications where minimal pressure drop is a key consideration for cost savings and efficiency. The SmartTrak 140 controls gas mass flow up to 500 standard L/min, with an ultralow pressure drop of 4.5 psid. This level is considerably lower than other available mass flow controllers, which typically offer pressure drop values of 25 psid, the company says. The SmartTrak 140 combines two of this company's existing technologies: its laminar-flow element, sensor and digital electronics, and its low-pressure-drop valve, which has a large flow coefficient. The SmartTrak 140 also offers the ability to set zero, span and full-scale for ten different gases independently in the field. This company also manufacturers the Chlorine-Trak 760S mass flowmeter, which is specifically designed for cost-effective flow measurement in chlorine injection processes found in wastewater treatment applications. — *Sierra Instruments, Monterey, Calif.*
www.sierrainstruments.com

This flowmeter has a direct-mount configuration

This company's Wafer Cone flowmeter (photo) now has a direct-mount configuration where the transmitter is directly

Note: For more information, circle the 3-digit number on p. 72, or use the website designation.

Flexim Americas

Parker Hannifin

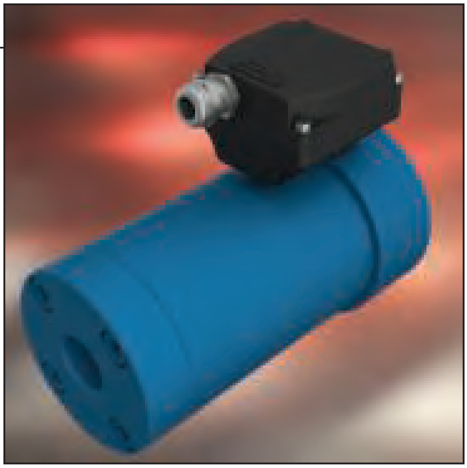
mounted, eliminating impulse lines. Eliminating impulse lines with the direct mount lowers installation costs, and reduces potential leak points by 50%, the company says. Simple plug-and-play mounting ensures the meter is installed correctly the first time. The advanced Wafer Cone flowmeter is ideal for gas or liquid service in line sizes from 1 to 6 in. The meter's flange-less design makes installation fast and easy, while the interchangeable cone accommodates changing flow conditions without the need for recalibration, because cones can be removed and replaced with a cone at a different beta ratio. — *McCrometer, Hemet, Calif.*
www.mcrometer.com

DP flowmeters that take advantage of the latest HART revision
Rosemount Model 3051 pressure transmitter products (photo, p. 29), capable of measuring flow with differential pressure (DP), have been enhanced to take advantage of the latest revisions to the HART communication protocol (HART 7). In addition, the company's range of DP flow and level instruments now feature Power Advisory Diagnostics, a predictive diagnostic tool that allows users to identify electrical loop issues at the instrument or anywhere in the loop before they cause a loss of measurement. Examples of these issues include water or corrosion in the terminal block or junction boxes, wiring problems or a failing power supply. — *Emerson Process Management, Chanhassan, Minn.*
www.emersonprocess.com

Handle severe duty with this flowmeter
The 3600 Series Digital Mass Flow instruments (photo) are designed specifically for severe industrial environments. Series 3600 devices are suitable for food and beverage, biotechnology and pharmaceutical and chemical processing applications that require frequent washdown, as well as chemical, petrochemical and industrial process applications where hazardous location certification is required. These stainless-steel flowmeters have digital control electronics that provide unparalleled accuracy, repeatability and control stablilty, as well as watertight construction that meets IP66 guidelines. — *Parker Hannifin, Hatfield, Pa.*
www.parker.com

Use this portable flowmeter for hazardous areas
The Fluxus F608 and G608 portable flowmeters (photo) are specifically engineered for use in hazardous areas. The flowmeter models, which can be used for liquids and gases, have been approved by FM International for use in Class I, Div. II areas. The F608 and the G608 are based on this company's widely used F601 and G601 models. The carbon-fiber housing is highly resistant to impact, as well as oil, water and a host of other liquids. The transducer has been designed for robustness. The portable F608 and G608 feature efficient battery management, and can be set up in less than five minutes, the company says. The units can be used in pipes as small as 0.4 in. (inner dia.) up to 255 in. and in temperature ranges from –40 to 390°F. — *Flexim Americas Corp., Edgewood, N.Y.*
www.flexim.com

Kral



These flowmeters are optimized with a new sensor system

Several series of this company's Volumeter flow measurement system (photo) are being optimized with a new sensor system that increases the maximum temperature for the OMG and OMH Series from 150 to 180°C and provides better resolution. The sensor is an "all-in-one" solution that measures flow amount, flow direction and temperature. Due to to miniaturization and system integration, a single sensor is now sufficient where three sensors used to be required, the company says. The new sensor is designed to make connecting to the Volumeter easier. — *Kral AG, Lustenau, Austria*
www.kral.at

These Coriolis meters are compact

The Sitrans FC430 is a Coriolis flowmeter said to be the most compact flow measurement solution on the market. The digitally based Sitrans FC430 has a short build-in length, and is suitable for any liquid or gas application in the chemical process industries. The instrument is ideal for multi-parameter measurement, and can be used effectively in applications such as fast-filling, batch control, blending and dosing, as well as for measurement of gases or fluids. In addition to the market-leading compactness, the Sitrans FC430 also can perform with 0.1% accuracy, low pressure loss and a stable zero point, the company says. — *Siemens AG, Industry Sector, Industry Automation Division, Nuremburg, Germany*
www.siemens.com

This magnetic flowmeter is for large sizes

The AdMag AXW series of magnetic flowmeters are available in sizes from 20 to 40 in., and are ideal for industrial process lines, water and other basic applications. The AXW series is designed for outstanding reliability and ease of use, and has a wide selection of liner materials, such as PTFE, hard rubber and polyurethane. The product comes standard with ASME, AWWA, EN, JIS or AS flanges. A submersible version is also available, the company says. The AXW also has a shorter face-to-face dimension for a smaller footprint and easier installation. — *Yokogawa Corp. of America, Newnan, Ga.*
www.yokogawa.com/us ■

Scott Jenkins

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CHEMICAL ENGINEERING
FACTS AT YOUR FINGERTIPS
Department Editor: Scott Jenkins

Gas Sparging

Sparging refers to the process of injecting a gas through a diffuser into a liquid phase. It is used in both physical and chemical process applications. In physical process applications, the gas is used to either dissolve gas into liquid (such as aeration and carbonation) or to remove contaminants from the liquid phase (such as oxygen stripping, volatile organic compound stripping and water stripping). In chemical process applications, sparging is used to dissolve reactant gases into a liquid phase for further reaction (such as in hydrogenation, oxidation, fermentation and ozonation reactions). The following information discusses some of the considerations required for setting up a sparging operation and selecting sparging equipment.

Spargers (diffusers) come in various sizes, configurations and materials of construction. They are chosen based on the design and operating conditions of the process. Metal spargers are used in high-temperature, corrosive or oxidizing conditions, whereas ceramic spargers are sufficient for mild conditions. The type and configuration of the sparger used depend on factors such as whether a process is continuous or batch, gas flowrate, tank size, mechanical agitation, operating pressure and temperature.

Mass-transfer rate

The main purpose of a sparger is to increase the gas-to-liquid mass-transfer efficiency, which is the ratio of the amount of active gas component (that is dissolved in liquid) to the gas injected. A low efficiency will result in an increased gas-injection rate and therefore increased cost to achieve the desired results. The liquid-phase mass-transfer resistance primarily controls the gas-to-liquid mass-transfer efficiency.

The gas-to-liquid mass-transfer rate per unit volume is given by $K_La(C^* - C)$. In this equation, K_L is the liquid-phase mass-transfer coefficient that is dependent on the diffusivity, liquid viscosity, temperature, and mixing; a is the interfacial area of gas bubbles in contact with liquid; C^* is the saturated concentration of the gas in liquid; and C is the concentration in bulk liquid. As the interfacial area a is increased by sparging small gas bubbles with high surface-to-volume ratio, the gas mass-transfer rate is improved. The mass-transfer driving force $(C^* - C)$ also has a big impact on the gas dissolution rate as the high-purity gas is used instead of the lower-purity gas. For example, the saturated concentration of oxygen in water from pure oxygen is five times higher than that from air, resulting in a large increase in the oxygen dissolution rate with pure oxygen.

Sparging equipment selection

There are several important factors to consider when selecting a sparger. The gas exit velocity at the sparger surface is an important design criterion for sparger selection. The actual gas volumetric flowrate for exit velocity is calculated using the pressure (P) that is the sum of tank headspace pressure ($P_{Headspace}$), liquid head pressure at the sparger (P_{Liquid}), and pressure drop across the sparger element (ΔP). The minimum sparger surface area is based on the gas exit-velocity limit for the process. The exit velocity limit is lowest for the static sparging operation when there is no mechanical agitation of the liquid phase. For agitated tank sparging and dynamic sparging, where liquid has high forced velocity along the sparger surface, the gas exit-velocity limits are significantly higher, requiring smaller spargers for the same gas flow. The exit-velocity limit for agitated tank sparging and dynamic sparging depends on the impeller speed and liquid velocity, respectively.

Agitation effects

Apart from using a properly designed sparger, it is important to focus on the mixing of gas and liquid. In chemical process applications, the reactor vessel is often closed so that the unreacted high-purity gases, such as hydrogen or oxygen, are not vented through the system. In these applications, specially designed mixing impellers are used, depending on the operating conditions of the reactor. Typically, one impeller turbine is located above the sparger to shear and disperse the gas bubbles. Agitation at the liquid surface may also be required to entrain the headspace gas into the liquid phase.

Contaminant removal

In physical process applications where a contaminant is removed from the liquid phase, it is important to estimate the gas required for the process, as well as other components volatilized from the liquid phase. Sometimes downstream processing equipment is required to recover the contaminant from the gas phase before venting the sparging gas to the atmosphere.

Biodiesel example

Many parameters have to be taken into consideration when estimating the gas flowrate and vent-gas composition in any sparging application. For example, when a gas is used to remove moisture from biodiesel, the key variables to estimate the gas requirement are temperature, pressure, moisture in influent gas, biodiesel composition, initial and final moisture concentration in biodiesel, batch size and batch time.

Figure 1 shows the effect of the operating temperature and pressure on the normalized gas flowrate to reduce the moisture from 2,000 ppm to 500 ppm in a batch process using pure nitrogen gas (~99.99% purity). The gas- and liquid-phase compositions can be estimated using the vapor-liquid equilibrium calculations.

Figure 2 shows the calculated moisture concentration in biodiesel at 10-psi and 14.7-psi operating pressures for processes at 200°F with 1-h batch time. The moisture concentration in both processes reduces from 2,000 ppm initial concentration to 500 ppm final concentration after sparging, but the process with 10 psi pressure requires 37% lower gas flow. This reduces the nitrogen consumption, as well as the biodiesel vented with the gas.

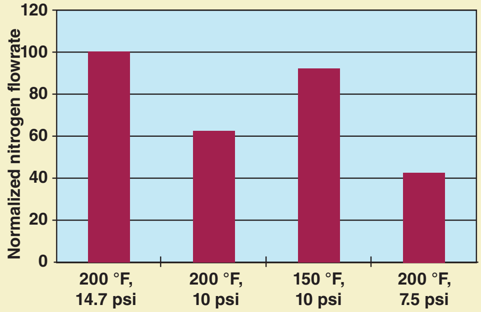

FIGURE 1. Normalized nitrogen gas flowrate for moisture removal from biodiesel

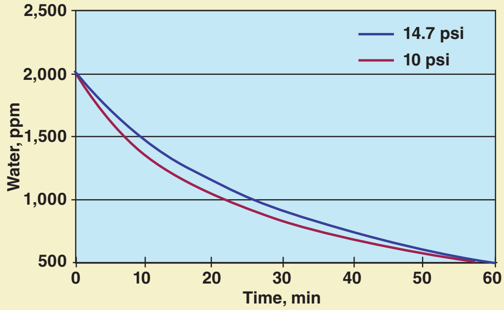

FIGURE 2. Moisture concentration in biodiesel with nitrogen gas sparging at 200°F

Editor's note: Content for this edition of "Facts at Your Fingertips" was contributed by Air Products (Allentown, Pa.; www.airproducts.com).

Cover Story

Condition Monitoring Methods for Pumps

Applying condition monitoring tests to pumps can save costs by optimizing overhaul scheduling

Ray Beebe
MCM Consultants Pty Ltd. and Monash University

Pumps are among the most commonly used machines in the chemical process industries (CPI). Condition monitoring tests and predictive maintenance can help pump operators determine when to overhaul pumps in a way that minimizes costs. Despite pumps' ubiquity and large energy demands, however, relatively little information is available on how to apply predictive maintenance approaches and condition monitoring to process pumps.

Pump overhauls may occur on a fixed time schedule or as a result of a specific breakdown, but neither case necessarily represents the most cost-effective policy. In cases where deterioration in the performance of a centrifugal pump causes a drop in plant production, pump overhaul is readily justified, because the cost of performing the overhaul is usually small compared to the losses from reduced production. However, when pump performance deteriorates due to wear and the only effect is increased power consumption (with no discernable effect on production), the question of when to perform an overhaul becomes important.

The right time to overhaul that minimizes cost can be calculated from predictive maintenance test results. The use of condition monitoring methods ensures that pump overhauls aimed at restoring performance are executed when they are actually necessary. This article describes several condition-monitoring tests for pumps, and discusses how to use these predictive-maintenance methods to estimate the increased power consumption that results from pump wear (Figures 1 and 2).

Experience in the field or with an original equipment manufacturer (OEM) may help pump operators determine when to undertake overhauls, but others may not have an idea of how to determine the optimal time, because many pump textbooks, even excellent and widely used ones, have failed to cover the application of condition monitoring to pumps in detail, and information in this area has been lacking until recently [*1, 2*].

It is hoped that the tools presented in this article will help asset managers and engineers improve management of pump assets to provide capacity for production, as well as improve energy efficiency and minimize greenhouse-gas emissions. Although the article focuses on pumps, the optimization approach described here can also be applied to other plant equipment where deterioration results in efficiency loss, and where energy consumption can be measured or estimated.

FIGURE 1. Pump wear, such as that shown on this pump impeller, can lead to reduced pump performance

FIGURE 2. The degree of pump wear is expressed as the percent reduction in total head

Choosing monitoring methods

The extent and specific effects of internal wear in centrifugal pumps vary according to the characteristics of the liquid being pumped, as well as the type of pump and its operating duty. Some pumps last for years, while others last for only months.

Pump monitoring methods (where justified) should be chosen that can detect each of the pump degradation modes that are either experienced or expected. Common pump-monitoring methods include the following:

- ***Vibration monitoring and analysis:*** probably the most widely applied method of condition monitoring for rotating machines in general, and suited to detect such faults as unbalance, misalignment, looseness
- ***Sampling and analysis of lubricants for deterioration and wear debris:*** relevant for bearings and

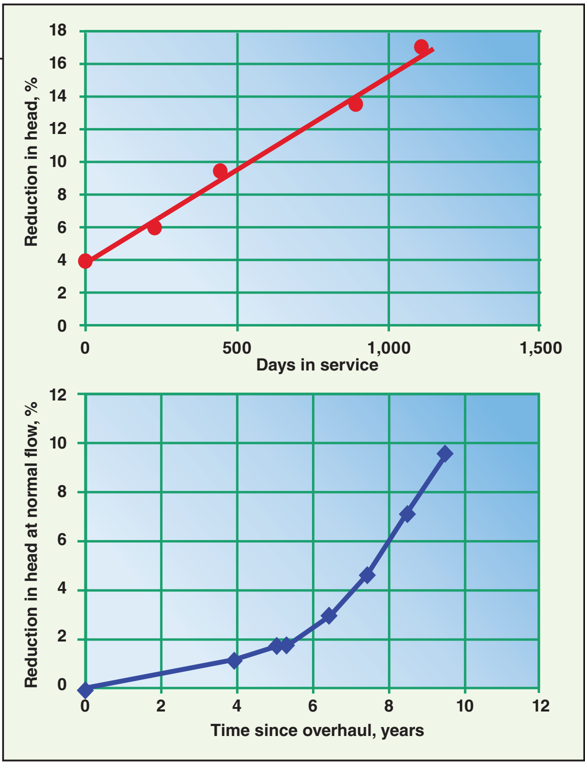

FIGURE 3. The degradation of pump performance can be shown by the percent decrease in total head at normal-duty flow, and can be determined by head-flow testing (Top graph, 230 kW; Bottom graph, 5,744 kW)

lubrication system faults
- ***Electrical plant tests:*** relevant for motor condition
- ***Visual inspection and non-destructive testing:*** particularly relevant for pump casing wear
- ***Performance monitoring and analysis:*** relevant for pump internal conditions

For critical machinery, the use of more than one method of condition monitoring in combination may be justified. This article demonstrates the use of performance analysis and includes some examples of condition monitoring in practice.

An understanding of basic pump performance characteristics is assumed, as is how to measure test data in a repeatable manner. For condition monitoring, repeatability is more important than absolute accuracy.

Head-flow method shows wear

The most useful condition-monitoring method is head-flow measurement, because in addition to pump deterioration, it also detects any changes in system resistance. This method can be used for all pumps where flow (or some repeatable indicator of flow) can be measured.

Figure 3 shows the trend in degradation of two pumps over multiple years. The degree of wear is expressed as the percent reduction in total head due to wear, compared with the pump's new condition, at duty-point flow. This is usually derived from head-flow tests near the duty point, but can also be obtained using the shut-off head test, where this test is allowable.

Regular interrogation of the plant historian can be used if a plant has a DCS (distributed control system). Figure 3 (bottom) shows data for a multistage boiler-feed pump (specifically, a constant-speed, four-stage pump; 5,853 rpm; 5,744 kW; 171 kg/s at 20.4 MPa). Data points are extracted every six months from a run of 10 days at a steady load.

Performance information such as that shown in Figure 3 can reveal the extent to which a pump has deteriorated, and pumps can be prioritized for overhaul on the basis of their relative wear. The question becomes whether an overhaul of the worst-performing pump is justified economically. (The article will revisit this question and provide a method for helping to make the decision later.)

Throttling the pump over its full flow range to obtain data points is not necessary for monitoring. Several points near the normal operating duty point are sufficient to reveal the effects of wear, usually shown by a shift in the head-flow curve towards the zero-flow axis by an amount equal to the internal leakage flow.

A series of test readings at steady conditions, at about 15-s intervals, is sufficient. Average values are plotted. Speed must also be measured for variable-speed pumps, and the head-flow data must be corrected to a standard speed using the affinity laws for pump performance.

Field tests sometimes yield results that are slightly different from the manufacturer's works tests because site conditions for flow and pressure measurement (as required by the various standards for pump testing) are rarely available. It is important to remember that for monitoring, relative changes are more important than absolute accuracy.

For condition monitoring, non-intrusive ultrasonic flowmeters are acceptable in many cases. A permanent flowmeter installed as part of a pump's minimum flow protection or for process measurement can be used, as long as the flowmeter's long-term condition is considered to be constant. Or, it can be inspected regularly.

Shut-off head method

The shut-off head method can be used for predictive maintenance of pumps. While measuring the head at zero flow is a relatively simple test, it is only possible where it can be tolerated by the pump. This is not the case for high-energy pumps, and for pumps of high specific speed where the power at shutoff is greater than that at the duty point. Pumps have exploded from a buildup of pressure when left running at zero flow.

To perform the shut-off head test, read the suction and discharge pressures (when steady) with the discharge valve closed fully for no longer than 30 s or so. To convert the pressure readings into head values, the liquid temperature is also needed to determine the density.

Wear on the outer diameters of pump vanes will show readily, because the head-flow curve of a worn pump shifts toward the zero-flow axis. To show wear of the sealing ring, the pump head-flow curve needs to be relatively steep. (Note that if the pump has a rising head-flow curve, internal leakage will initially give rise to an increase in shut-off head).

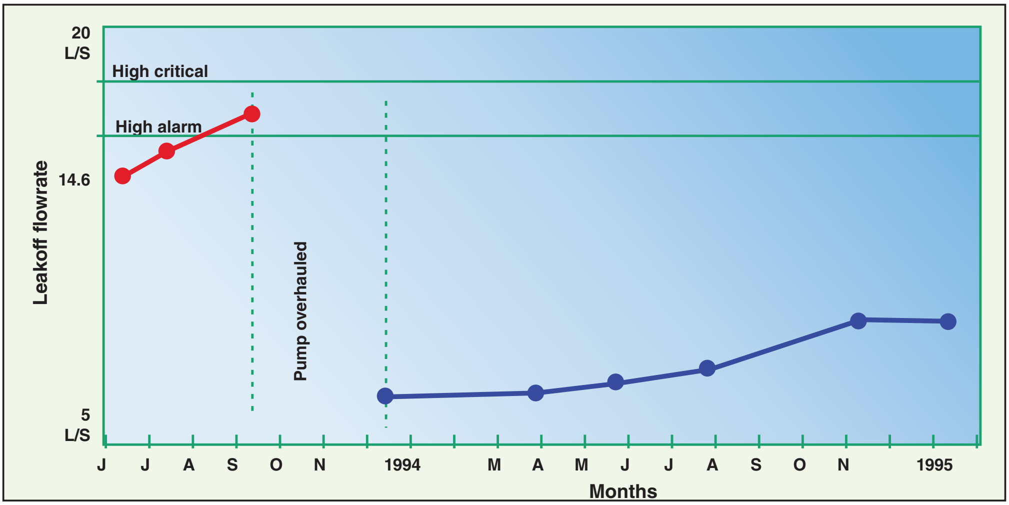

FIGURE 4. Condition monitoring of a high-energy multistage pump can be accomplished by measuring the balance-device leakoff flow. (Note: flows shown are corrected to a standard pump speed)

Thermodynamic method

Another method of pump monitoring is to measure the temperature rise of the liquid through the pump. This measurement is a reflection of the inefficiency of the pump. Because the differential temperature is small, great care is required to measure it. For example, effects of recirculation at the pump inlet and outlet must be eliminated and thermodynamic tests are not possible at low or zero flow. Pump efficiency can be calculated from the measured data of inlet temperature, the differential temperature and the head. If the efficiency changes over time, comparisons can be made on plots of efficiency versus head. For high-head pumps, an allowance must be made for the isentropic temperature rise, which occurs as a result of pressure increase [2].

Commercially available devices are widely used for the thermodynamic method, especially in the water industry [3]. For the installation of pressure and temperature probes, tappings for measuring suction and discharge must be two pipe diameters away from pump flanges. Tong-type detectors are placed to measure motor power. Pump efficiency is then found from the precise measurement of the head and temperature rise through the pump. By assessing motor losses, the power absorbed by the pump can be computed, and from these data, the pump flow can be found.

For condition monitoring, tests at around the normal operating point are usually sufficient. The thermodynamic method would be more attractive economically if no special tapping points were required. Research at Monash University (Gippsland, Australia campus) on high-head pumps using special semiconductor temperature probes on the outside surface of the piping (even when covered with insulation) gave usable results, provided the pump was allowed to run at steady operating conditions for 30 min to allow the piping temperature to stabilize [4].

The empirical expression shown below gives the percent efficiency for pumps at up to 54°C. The expression includes a correction for the isentropic temperature rise. Total head is in kPa, and temperatures are in °C [5].

$$\frac{100}{\left[1 - 0.003(InletTemp - 2) + 4{,}160\dfrac{Temp\,rise}{Total\,head}\right]}$$

Measurement of balance flow

Multistage pumps with their impellers facing in one direction usually have a balance disc or drum arranged such that the final-stage discharge pressure counteracts the axial thrust on the shaft line. Another method for pump condition monitoring is to measure the leakoff from the balance device [6].

The basis of the method is that if increased wear in the annular space of the balance device is evident from increased leakoff flow, then the interstage clearances must also be worn. Because the leakoff line is quite small compared to the pump main flow piping, a permanent flowmeter for this method is relatively inexpensive.

Figure 4 shows an example of using leakoff flow to schedule pump overhauls of boiler-feed pumps at a power plant. Flows are read manually, and trends plotted using a database program. Note that here the balance flow (15 L/s) corresponds to about 10% of the regular duty flow, and consumes about 250 kW of extra power. When added to the likely internal recirculation, this would mean that an even larger proportion of the power absorbed is being wasted.

The boiler-feed pumps are variable-speed units, and other tests show that the measured flows must be corrected in direct proportion to the speed. On a set of pumps of a different design at another location, both head-flow and balance flow were measured for some years, but no correlation was found between the two.

On still another pump type, of the 11 stages, the head-flow performance was determined to be well below the datum curve. As the pump was dismantled, measurements showed that the interstage clearances were not worn. A condition-monitoring credibility crisis was averted when the balance seat area was reached and found to be severely eroded from water leaking past the valve seat. Balance flow had obviously been very high. For the best monitoring, it is therefore desirable that both head-flow and balance flow should be measured, particularly if the balance area can be separately dismantled in the field.

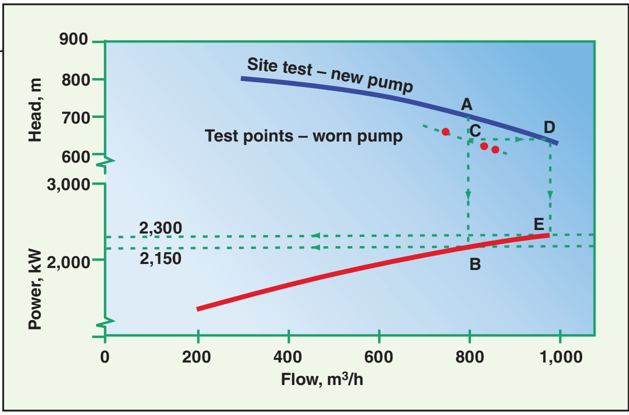

FIGURE 5. Head-flow-power characteristics of a new pump differ from the data points for a worn pump. Point C to D shows internal leakage flow

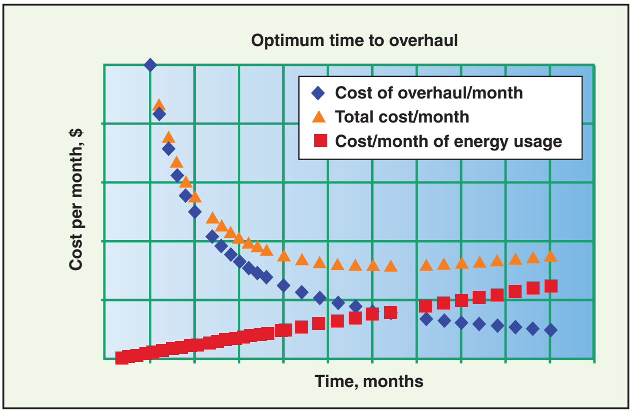

FIGURE 6. The point that optimizes the time to pump overhaul can be shown graphically

Calculating time to overhaul

The most economically cost-effective time to restore lost pump performance by overhaul varies with the circumstances. If the deterioration of pump performance is constant over time, then operators can conduct a cashflow analysis to ensure that investing in the pump overhaul will yield the required rate of return. This analysis is the same process as that used in deciding on any significant investment in plant improvement. If the deterioration rate is increasing with time, then the optimum time for overhaul occurs when the accumulated cost of the increased electricity consumption equals the cost of the overhaul itself.

The method for calculating the optimal time to overhaul can be applied to a number of pump-operating scenarios, which are described as follows:

Pump deterioration results in a reduction in plant production: Whenever the cost of the overhaul is small or insignificant compared to the cost of lost plant production, prompt overhaul is usually justified at any convenient "window."

Pumps that run intermittently to meet a demand: In a pumping installation such as topping-up a water supply tank or pumping liquid out of a tank or vessel, deterioration will result in the pump taking more time to do its duty. Therefore, the extra service time required of the pump results in increased power consumption, and that can be compared to the cost of conducting the overhaul.

Pump deterioration does not affect plant production: In some cases, where a pump is operated at constant speed with throttle-valve control, a deterioration in pump performance may not affect plant performance, at least initially. The internal wear does not cause any loss in production from the plant, because the control valve can open more fully to maintain pump output. Eventually, as wear progresses over time, pump output may not be sufficient to avoid a loss of production. Or the extra power consumed will exceed the motor rating.

Figure 5 shows the head-power-flow site-test characteristics of such a pump. Its output is controlled using a throttle control valve. The duty flow is 800 m^3/h, and the duty point in the new condition is represented in Line A. The power absorbed by the pump is read off the power-flow curve (2,150 kW; Line B). The power-flow curve should ideally be found onsite, but information from the works tests may have to suffice.

After some service, the data points marked on the diagram as "Test points — worn pump" indicate that internal wear has occurred. When worn to this extent, the operating point moves to Point C (the system resistance curve moves lower when the throttle valve is opened further).

The increased power required in the worn condition can be estimated by extending from the head-flow curve at constant head from the operating point to Point D, and then dropping straight down to the point where the the power-flow curve intersects for the new condition at constant flow: Point E. Next, follow the arrowed line in Figure 5. It is assumed that the original curve still represents the flow through the impellers. Less flow is leaving the pump to the system due to internal wear. (If the pump was motor-driven, it may be possible to measure the actual power by a test at extra expense).

In the example, the power required for this duty in the worn condition is shown in Figure 5 by the projection from the duty flow of 800 m^3/h to the test curve to find 640 m of head, then by moving across to the "site test — new pump" curve, and down to the power curve, to find 2,300 kW.

The extra electricity consumption is therefore 2,300 – 2,150 = 150 kW, which is then divided by motor efficiency (90%, in this case), to obtain 167 kW of extra consumption.

If the sealing clearances are known (either by previous experience of correlation with measured performance, or if the pump is opened up already), operators can estimate the extra power consumed that will likely be saved as a result of an overhaul [*1, 8*].

Finding the optimum time for overhaul from head-flow data. For this example (Figure 6), the test points were obtained following 24 months of service (the pump was known to be in new condition at the start). An overhaul would cost $50,000, with electricity costs of $0.10/kWh and with the pump in service for 27% of the time, on average. The increased power consumption is estimated at 167 kW (with motor efficiency included in the total). The test discussed here shows that the rate of increasing cost per month has reached 167 × 0.10 × 0.27 × 720

= $3,246/mo (using an average month as 720 h).

As the time that the pump has been operating now is 24 months, $3,246 ÷ 24 gives the average monthly cost rate of deterioration as $135/mo.

The optimum time for overhaul (T) is calculated [7] from Equation (1).

$$T = \sqrt{\frac{2 \times O}{C}} \quad (1)$$

where O = cost of overhaul and C = cost rate of deterioration.

This calculation yields an optimum time to overhaul of the pump of 27.2 months. It is better to calculate and plot the average total cost-per-month values for a range of times. Doing so will allow the cost impact of undertaking the repairs at another time (such as at a scheduled plant shutdown), to be seen clearly.

Average cost, month by month

For an example, use the time period of 22 months. The average cost of conducting the overhaul is now $50,000 ÷ 22 = $2,273/mo.

The average cost of extra energy consumed is then ($135 × 22) ÷ 2 = $1,485/mo. Total average cost per month is the sum of these two figures: $3,578/mo. Since pump wear progresses linearly, the cost per month also is linear and the average can be obtained by dividing by two (at the starting time, the value is zero, and the average over the time period is given by the ending value minus the start value, divided by two).

Repeat this calculation for several months (the use of a spreadsheet can be helpful) and look for the minimum total cost, which is at 27.2 months. If plotted as cost per month against time, the resulting curves will show that the cost per month of overhaul drops over time, and the cost of extra energy increases with time.

The time value of money could also be taken into account, if required. Usually the total cost curve is fairly flat for ±20% or so. The calculations can be easily set up using a spreadsheet.

If the overhaul was delayed until the 30-month mark, for example, then the accumulated cost of lost energy would have reached $135 × 0.5 × 30^2 = $60,750. At 27.2 months, the cost is $135 × 0.5 × 27.2^2 = $49,939. The cost of delaying overhaul is thus the difference between the two ($10,811).

This calculation is only correct if the wear progresses at a rate that increases uniformly with time. As Figure 3 shows, this is not unusual. Information may not be available to make any other assumption, but decision makers have to start somewhere. Other formulas apply for nonlinear rates of change [7].

Some notes and additional points to consider regarding these calculations are as follows:

- Some relatively small pumps may never fully justify an overhaul on savings in energy use alone, but the overhaul may be justified on reduced plant-production rate
- The method does not apply to pumps of high specific speed that show little change, or even a reduction, in power with increased flow
- If a pump varies in its duty, then the energy usage would be corrected in proportion
- The cost of electricity to be used here may vary with the power supplier's tariff structure. The cost may be less in stepped blocks with higher consumption levels for the plant, or a peak-demand charge may apply

Pump deterioration does not affect production (variable-speed control): In situations where a pump has variable-speed control, and performance deterioration does not (at least initially) affect production, the following steps can be taken. For a pump where the speed is varied to meet its desired duty, the effect of wear on power required is much more dramatic than for the case of a constant-speed, throttle-controlled pump. This is because the power usage increases in proportion to the cube of the speed ratio.

Unless the pump output is limited by the pump reaching its maximum speed, or by its driver reaching its highest allowable power output, then no production will be lost. However, power consumed will increase more dramatically for a given wear state than that for a constant-speed pump.

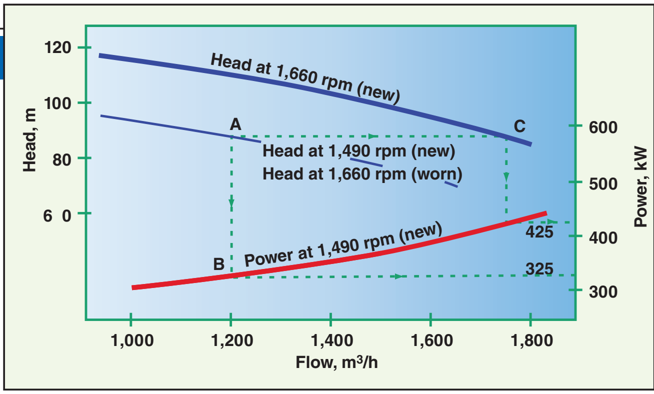

FIGURE 7 For variable-speed pumps, the effect of wear on power required for operation is more dramatic than constant-speed pumps

To estimate the power required in the worn state, the head-flow curve must be drawn for the current higher speed in the new condition. Select a head-flow point on the original, new-condition curve, and correct it to the higher speed by multiplying the flow by the speed ratio and multiplying the head by the square of the speed ratio. Repeat this for several other points at flows above regular-duty flow to draw the new condition head-flow curve.

Follow the same method and calculations as in the previous section to find the time for overhaul at minimum total cost. The operating point is projected from the worn curve to the new curve at the same speed as the worn curve.

Figure 7 shows the performance of a variable-speed pump. When new, operation at 1,490 rpm meets the desired duty flow, at operating Point A, requiring 325 kW of power (Point B). After some period of service, internal leakage has increased to the extent that the pump must run at 1,660 rpm to meet the required duty (still Point A).

To estimate the power required now, the head-flow curve must be drawn for the higher speed in the new condition. Several head-flow points are selected and corrected to the higher speed. To do this, multiply each flow by the speed ratio, and multiply each matching head by the speed ratio squared. This will result in the head-flow curve at 1,660 rpm in the new condition.

Project across from the head at the duty flow (Point A) to meet the head-flow curve at 1,660 rpm (new condition; Line C in Figure 7). Projection downward at a constant flow leads to the increased power required at 425 kW. The extra power is 31% more. (This pump is driven by a steam turbine, so power consumption cannot be measured).

The same calculations as those presented in the previous section are followed to find the time to an overhaul that minimizes total cost.

Using shut-off head test results

The shut-off head test information can also be used to estimate the power used in the worn state, and to perform the optimization calculations as explained in the above section.

Head-power-flow characteristics in the "new" state are needed as before, and the operating point must be known. Note the power required at the operating point, as before.

Make an overlay trace of the head-flow curve in the new condition. Place it over the "new" curve and move to the left horizontally until the curve cuts the head axis at the value of the shut-off head obtained on the test. The trace is now in the position of the "worn" head-flow curve, which is being experienced. Exactly the same process as that explained above can be followed. ■

Edited by Scott Jenkins

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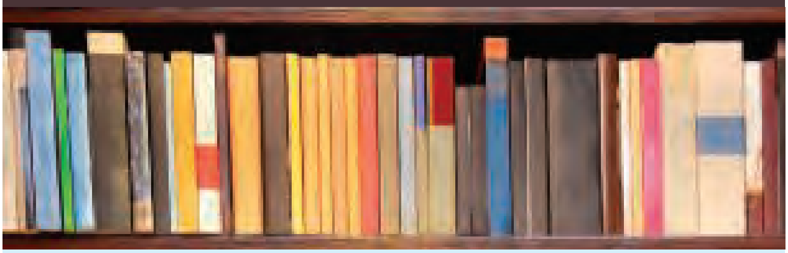

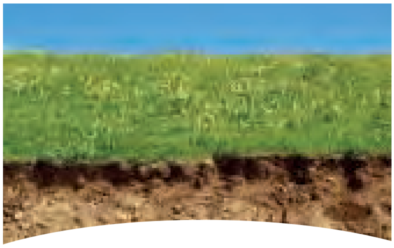

Thin-Film Drying Offers Deep Benefits

This continuous drying technology is especially safe and flexible, and offers many benefits, including low operating costs, low space requirements and high specific-evaporation capacity

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Contact agitated dryers are used for drying wet powders, slurries, filter cakes, and applications where off-gases during the drying cycle must be contained. If the applications are also heat-sensitive, such as specialty chemicals, polymers, food products and pharmaceuticals, contact drying must take place at low heating temperatures, at residence times in tune with the tolerance of the product, and occasionally in conjunction with a heated sweep gas (stripping medium). These operating conditions plus the ever-present wall fouling by the product, considerably reduce the effectiveness of commonly used agitated dryers (for instance, paddle dryers and disc dryers) to the extent that they become inefficient.

Therefore, there is a strong demand by the chemical process industries (CPI) for a high-performance, continuous agitated dryer. Thin-film drying technology (TFDT) meets these challenges. Instead of a heated sweep gas, thin-film dryers (TFDs) operate at a reduced pressure (0.01 bar or even lower). Retention time, which is normally measured in hours for paddle and disc dryers, is measured in minutes or even in seconds, and the fouling problem — a common deficiency of conventional agitated dryers — is eliminated or controlled.

TFDT is characterized by a low consumption of utilities and consequently of operating costs (1.15 kg saturated steam per 1 kg of evaporated water), compact construction (and hence low space requirements) and high specific evaporation capacity. These major features make TFDT economical. It offers operational flexibility, feedstock adjustability and simple handling. Meanwhile, low product holdup and product containment has earned TFDT the recognition of safe processing technology.

The thin-film principle is widely used for evaporation of temperature-sensitive specialty chemicals, polymers, food products and pharmaceuticals, plus a growing list of other materials (for more, see box entitled Typical applications). Applying this principle to the drying of solids involves only a modification of the well known thin-film evaporation principle to now handle salt solutions, polymer solutions, slurries, semi-solids, press cakes and wet powders from the CPI.

Whether a vertical or a horizontal thin-film dryer (TFD) type is appropriate depends on a variety of parameters, which are explained below. Applications for both TFD types are provided as illustrative examples.

TABLE 1: CLASSIFICATION, MODE AND RANGE OF OPERATION

Classification	Mode of operation		Range of operation		
	Continuous	Batch	Heating temperature, °C	Operating pressure, bars	Residence time, min
Vertical	✓	✗	0 ~ 700	0.01 – 30	0.15 – 2
Horizontal	✓	✗	0 ~ 700	0.01 – 30	0.15 – 30

TABLE 2: SELECTION CRITERIA ACCORDING TO THE CONSISTENCY OF THE FEEDSTOCK

Type of dryer	Vertical	Horizontal
Consistency of feedstock	**Suitability**	
Solution, suspension	✓	✱
Slurry	✓	✱
Thixotropic paste (pumpable)	✱	✓
Pasty (hardly pumpable)	✗	✗
Filter cake	✗	✓
Crumbly to free flowing	✗	✓
Granular (mm range)	✗	✱
Very coarse (cm range)	✗	✗
Fibrous	✗	✱

✓ = Suitable; ✱ = Conditional; ✗ = Not suitable

TYPICAL APPLICATIONS

Generally, thin-film dryers are used for drying crystalline type powders or filter cakes, which tend to agglomerate or adhere to heating surfaces during drying. More specifically, applications for thin-film dryers exist in such areas where the material to be dried is temperature-sensitive or where an auxiliary sweep gas is not satisfactory. A very small sample of applications is shown in the table below, classified according to the type of dryer.

SHORT LIST OF EXAMPLE APPLICATIONS

	Type of dryer	
Application	**v-TFD**	**h-TFD**
Glycerine separation and recovery	✓	
Saline effluents and residues	✓	
Sodium lauryl sulphate in isopropanol		✓
Pigments water wet		✓

Thin-film dryers are recommended for applications where low heating temperatures and reduced pressures plus high performance are needed. These include pharmaceuticals and polymers.

Drying intermediate pharmaceuticals

Drying intermediate pharmaceutical products is a quality-determining process step. In many occasions the production process of such products follows a batch mode. In the reaction stage of the process it is rather common to use solvents, occasionally also water, as the carrier phase. After crystallization of the intermediate product and mechanical separation of the liquid carrier phase from the solid intermediate, pharmaceutical drying is applied.

Batch drying is typical for the pharmaceutical industry. It is often justified with arguments such as, better control of the temperature and better identification of the batches; the latter being imposed by good manufacturing practice (GMP) directive. The batch drying process is characterized by the following steps:

- Charge the wet filter or centrifuge cake into the dyer
- Evacuate the process chamber of the dryer to the desired operating pressure; if possible heat up the dryer at the operating temperature simultaneously
- Dry the charge to its final volatiles content
- Cool the dryer
- Discharge the final dry material from the dryer
- Clean the dryer from the residual material
- Condition the dryer for the next charge

Beside the fact that batch drying demands long residence time (the actual process time), the total time required for the completion of one charge can be as long as 24 hours or even longer. Generally, the drawbacks of batch drying are:

1. Long residence time
2. High energy demand (heating and cooling cycles for each batch cycle)
3. Large drying equipment or a series of small equipment operating in parallel
4. High manpower demand (intensive handling)
5. Intermediate storage of batches is necessary

A viable alternative for these applications is a horizontal thin-film dryer that is customized for pharmaceutical applications (h-PTFD), which offers the following advantages:

- High heat-transfer coefficient leading to high specific drying rates, even at low heating temperatures
- Extremely short product residence time in the process chamber of the dryer (< 5 min) under mild conditions

These features also translate into small economic units.

Comparing the product capacity resulting from a conventional batch paddle dryer with the one achieved in an h-PTFD, one can immediately appreciate and acknowledge the fundamental differences (see Figures 1 and 2).

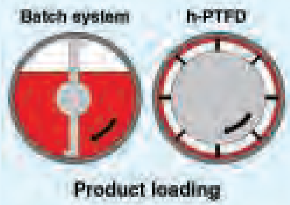


FIGURE 1. Product hold-up in a batch paddle dryer (left) is much larger than that achieved in an h-PTFD

FIGURE 2. In this test, product was tested in both an h-PTFD and a batch paddle dryer. The difference was almost a factor of 500

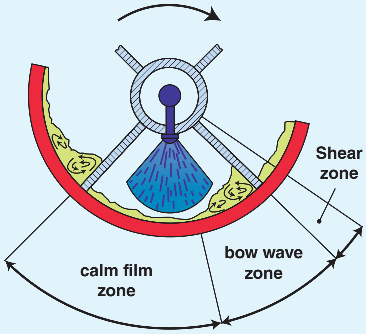


FIGURE 3. Since h-PTFDs offer perfect mixing, new possibilities for all-in-one processing, such as mixing and reacting open up with the positioning of rotating spray nozzles positioned along the axis of the rotor

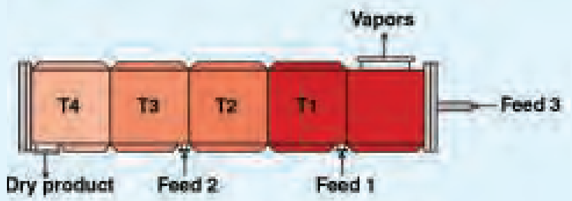


FIGURE 4. By splitting the heating jacket into different heating zones and placing multiple feed nozzles along the axis of the dryer, an h-PTFD can maximize process control flexibility

Compared to the paddle dryer, an h-PTFD has an extremely short product residence time and a high heat-transfer coefficient. The product residence times of a typical application that was processed in both dryers is shown in Figure 2.

Shortening the product residence time up to a factor of 500 has the following effects:

- Reduction of side reactions
- Reduction of color changes

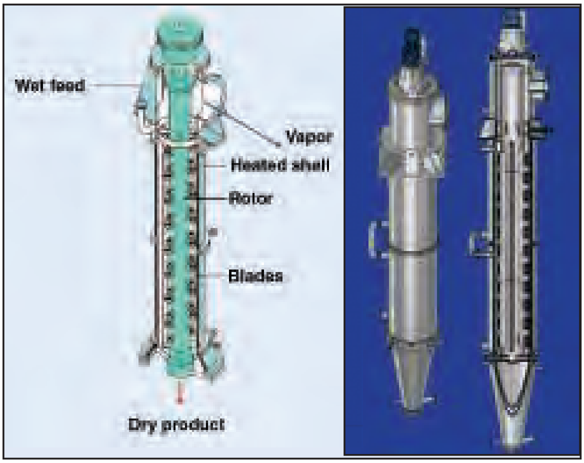
Furthermore, with rapid and non-destructive drying h-PTFDs open up new possibilities for all-in-one processing. This can be accomplished, for instance, through the provision of rotating spray nozzles positioned along the axis of the rotor (Figure 3). Through such nozzles additional substances can be sprayed on the thin-film. Since this thin-film is extremely turbulent, a perfect mixing of the substances takes place. Therefore, an h-PTFD can allow simultaneous mixing and reaction processes in one single processor.

Meanwhile, the maximization of the process control flexibility is achieved through splitting the heating jacket into different heating zones. Last but not least, additional feed nozzles can be placed at different places along the axis of the dryer (Figure 4). ❑

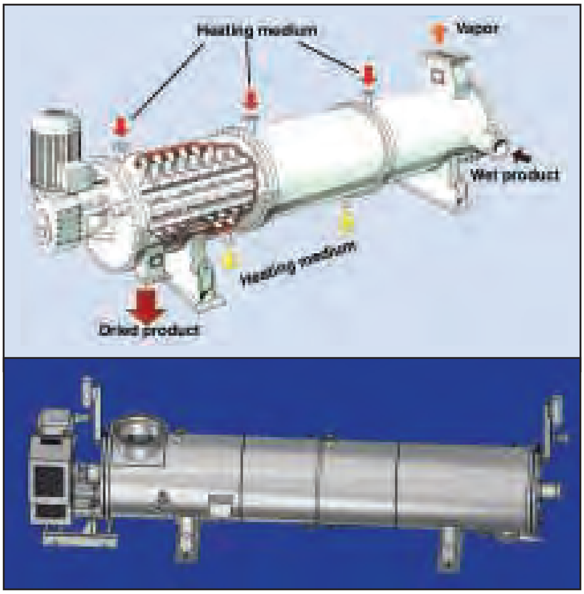
TABLE 3: SELECTION CRITERIA ACCORDING TO THE VOLATILE COMPONENT CONTAINED IN THE FEED STREAM		
Type of dryer	**Vertical**	**Horizontal**
Volatile component	**Suitability**	
Water	✓	✓
Solvent (low boiling point)	✓	✓
Solvent (high boiling point)	✓	✓

TABLE 4: SELECTION CRITERIA ACCORDING TO THE PRODUCT BEHAVIOR DURING DRYING		
Type of dryer	**Vertical**	**Horizontal**
Product behavior	**Suitability**	
Crystallizes during drying	✓	✓
Breaks up in particles	✓	✓
Pasty phase during drying	✱	✓
Sticky / viscous phase	✗	✗
Crust forming on surfaces	✱	✱
Temperature sensitive	✓	✓
Softening at low temperature	✗	✗
Melting at low temperature	✗	✗
Combustible	✓	✓
Dust explosion risk	✓	✓
Toxic	✓	✓
Abrasive	✱	✱
✓ = Suitable; ✱ = Conditional; ✗ = Not suitable		

TABLE 5: SELECTION CRITERIA ACCORDING TO THE DESIRED FORM OF THE FINAL PRODUCT		
Type of dryer	**Vertical**	**Horizontal**
Desired form of final product	**Suitability**	
Powder	✓	✓
Fine crystals	✓	✓
Coarse crystals	✱	✱
Granular	✗	✗
Paste	✓	✓
Liquid or slurry	✓	✓
✓ = Suitable; ✱ = Conditional; ✗ = Not suitable		



FIGURES 5 AND 6. Whether of vertical (above) or of horizontal design (below), a TFD consists of a cylindrical body with a heating jacket and a coaxial close-clearance rotor fitted in the shell. The rotor is equipped with special elements (blades) and revolves at high speed



Background

Drying is a thermal-separation unit operation whereby, through heating, volatile substances are removed from a single non-volatile material or a mixture of several of them. The final product is relatively or completely free of volatiles. Its consistency can be that of a powder, a granular solid, a paste or even a concentrate of high viscosity. In the majority of industrial applications, the volatile substance is an organic solvent or water and occasionally a mixture of both.

Concisely, dryers are classified according to how heat is transferred to the moist material in convective (direct contact), contact (indirect), radiation or infrared, dielectric and freeze dryers.

Thin-film drying technology falls into the category of contact agitated dryers and features the following:
- Low energy consumption
- Low space requirement
- High operational flexibility
- Safe processing of toxic and hazardous feedstocks
- Once-through process, without the need for recycling
- Ability to handle feedstocks of nearly any consistency

Thin-film dryers are characterized by the following attributes:
- Gentle treatment of temperature-sensitive products
- High heat-transfer rate
- Short residence time
- High number of stirred tanks
- Plug-flow
- Low product hold-up
- Continuous operation
- High turbulence in the thin product layer
- Environmentally friendly processing of toxic and hazardous substances due to completely closed design

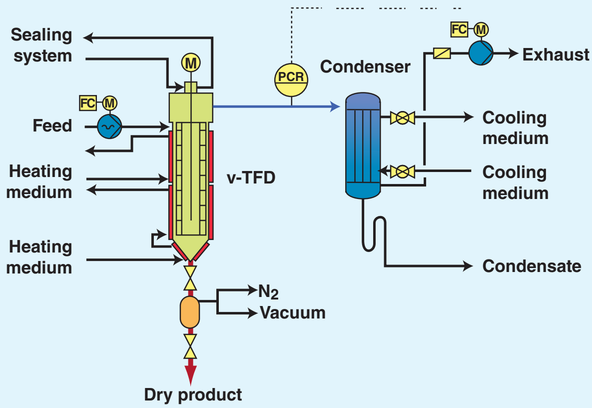
Classification and selection

Tables 1 through 5 provide guidance toward the following:
- Classification of thin-film dryers, the mode and the range of their operation (Table 1)

VERTICAL TFD APPLICATION

The v-TFD is recommended for applications where a liquid feedstock under the effect of vaporization would spontaneously crystallize. Spontaneous crystallization means that no intermediate pasty phase would take place. The case study described below demonstrates the advantages of the v-TFD.

In a specialty chemicals plant, product losses occurred with centrifugation method formerly used as part of purification procedure in the manufacture of dimethyl sulfoxide. The process specifications required removal of a relatively small amount of inorganic salts from the product. The solution was to install a v-TFD for the processing step (see figure, right).



Normally, a solvent is evaporated and passes out the top of the unit while powdered product is collected in the bottom. In the redesigned case, however, the product, dimethyl sulfoxide, is recovered from the top of the dryer and the powder in the bottom is the material removed. In operation, the feed material enters the top as slurry. A feed-specific distribution system spreads the feed material evenly over the circumference of the thermal section. As the rotor turns and the blades wipe the shell of the unit, the drying process occurs in seconds.

The result is that the improvement over the centrifugation method has been dramatic. Product losses have been almost eliminated. Bottoms are in excess of 96% dry. Capital recovery from improved product yield is enabling the equipment to pay for itself in a relatively short period of time. Maintenance has also been greatly improved, with at least several thousand dollars having been saved. The system operates under vacuum, which along with the short residence time minimizes overheating and helps to maintain the high quality of the dimethyl-sulfoxide solvent being recovered. ❏

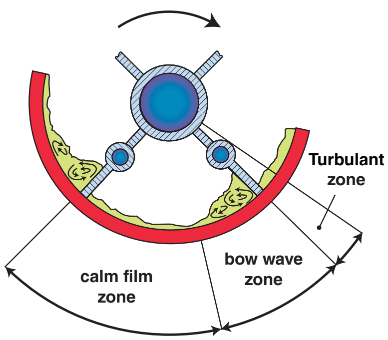


FIGURE 3. In the thin film principle, a highly agitated bow wave is formed in front of the rotor blades. The turbulence increases as the product passes through the clearance before entering a calming zone situated behind the blades

- Selection according to the consistency of the feedstock (Table 2)
- Selection criteria according to the volatile components contained in the feedstream (Table 3)
- Selection criteria according to the product behavior during drying (Table 4)
- Selection criteria according to the desired form of the final product (Table 5)

Thin-film principle

Whether of vertical (Figure 5) or of horizontal design (Figure 6), a TFD consists of a cylindrical body with a heating jacket and a coaxial close-clearance rotor fitted in the shell. The rotor is equipped with special elements (blades) and revolves at high speed. The blades are designed to evenly spread the material over the entire heated wall. This action forms a thin layer (film) over the heated wall, irrespective of the rheological consistency of the feed material, which is continually renewed and accompanied by intense turbulent flow conditions. The thickness of the layer is defined by the clearance between the rotor blades and the wall.

A highly agitated bow wave is formed in front of the rotor blades (Figure 7). The turbulence increases as the product passes through the clearance before entering a calming zone situated behind the blades. As the product conveys along the heated wall, the volatile component evaporates continuously. The minimum clearance between the blades and the dryer's wall (blades do not contact the heated wall) prevents fouling of the heating surface, minimizing the resistance to heat transfer, yielding rapid vaporization, promoting plug flow and granting the possibility to precisely control the product temperature.

Operating principle of TFDs

Both horizontal and vertical types of dryers can be heated electrically, inductively or with steam, hot water or thermal oil.

Vertical TFDs. Typically, the drying process in a vertical thin-film dryer (v-TFD) comprises three distinct zones: the preheating, the slurry and the powder zone. In the preheating zone the product reaches its boiling temperature. Evaporation starts, supersaturation is reached and then solid particles begin to form. In the slurry zone, the concentration of solids increases. In this phase many products tend to form agglomerates, which are crushed under the action of the blades. In the powder zone the solid particles still contain volatile components, but they further vaporize until the drying process completes.

For a given v-TFD, the length of each one of the three zones depends on the feedrate, the initial volatiles content and the operating conditions (predominantly pressure and heating temperature). Under steady operating conditions, if the feedrate to the dryer were to increase, the length of the preheating and the slurry zone would also increase. The same effect would happen, if at a constant feedrate the amount of contained volatiles would significantly increase. In either case, the result is that the powder zone decreases, which in turn leads to incompleted drying (higher final volatiles content). Nevertheless, for a well designed v-TFD system, fluctuations of the preheating and slurry zone can be compensated through the adjustment of the operating conditions.

The vapors of the volatiles move through the dryer counter-current

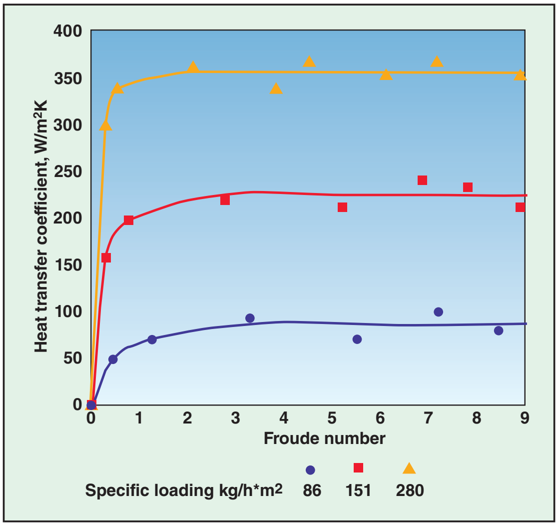


FIGURE 8. In a TFD, once the Froude number is higher than 1, the heat transfer coefficient stays practically constant. Similar measurements on a paddle dryer showed that constant heat-transfer coefficients are not reached until the Froude number is at least higher than 4. In terms of energy, this means the paddle dryer must turn at a higher RPM — with a larger electric motor, higher purchase expenditure and permanently higher operating cost — to reach the same heat transfer coefficient as in the TFD

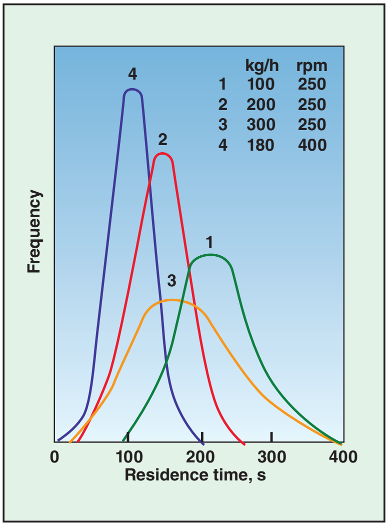


FIGURE 9. In this graph of residence time distribution (for a reference material at various operating conditions), of particular importance is Run 4, which shows the mean residence time of 60 s at a specific load of 195 kg/h-m². This is a particularly narrow span for a solids-handling piece of equipment

to the product flow and condense in an external condenser. As they flow through the dryer toward the vapor exit nozzle, any entrained fine particles re-agglomerate due to the contact with the wet feed. The final product from a v-TFD is usually a free flowing powder. However, other final product forms are possible, such as paste, slurry, melt and so on. The discharge of the final product from a v-TFD depends on its consistency and on the operating conditions. Specialized discharge systems are available.

Horizontal TFDs. The horizontal TFD (h-TFD) is usually applied for drying pastes, such as filter or centrifuge cake feedstocks, slurries or crumbly materials. Pumps, screws or other feed devices supply the wet material to the dryer. The rotor is an endless screw. It is fitted with shovel-type blades to move the material, mixing blades to reduce the size of the lumps, and fixed clearance blades to maintain an even distribution while exposing every particle of product to the heating surface. The necessary residence time — a parameter defined by the product characteristics or the process specifics or from both — is controlled by adjusting the blades configuration of the rotor.

Comparing performance

Froude number. The performance of TFD can be compared with other agitated dryers by using heat transfer and the Froude number (Fr). The Froude number is the ratio of the centrifugal to the gravitational forces inside the unit. It is defined as follows:

$$Fr = \frac{(2\pi * n)^2 * r}{g} \quad (1)$$

Where n = The number of revolutions of the rotor, 1/s

r = radius of the rotor, m

g = gravitational acceleration, m/s²

In TFD a proper Froude number is important because it ensures that the material covers the whole heating surface. This is accomplished when the centrifugal force is at least equal to the gravitational force. Figure 8 shows the dependence between U and Fr. The results were obtained with a reference free-flowing, dry material.

From the figure, it becomes evident that once the Froude number is higher than 1, the heat transfer coefficient stays practically constant. Furthermore, it has been shown that the value of the heat transfer coefficient depends only on the specific loading of the TFD. Similar measurements on a paddle dryer showed that constant heat-transfer coefficients are not reached until the Froude number is at least higher than 4. In terms of energy, this means the paddle dryer must turn at a higher rotational speed — with a larger electric motor, higher purchase expenditure and permanently higher operating cost — to reach the same heat transfer coefficient as in the TFD.

Normally, heat transfer coefficients for free-flowing, dry granular materials in horizontal thin-film dryers are one order of magnitude higher, compared with conventional disc or paddle dryers. For the drying of filter cakes, slurries and very wet powders, TFD is particularly efficient, since Froude numbers greater than 1 can be used effectively. This is because the wet product in the dryer now behaves as a liquid, and accordingly reacts to turbulent conditions. As a result, there is more exposure to the heating wall, plus a greater break-up of the agglomerates as they are made.

U-value. The overall heat transfer coefficient is defined as follows:

$$U = \frac{Q}{A\,\Delta T} \quad (2)$$

Where

U = Overall heat transfer coefficient, W/m²K

Q = Total heat input into the unit, W

A = dryer surface area, m²

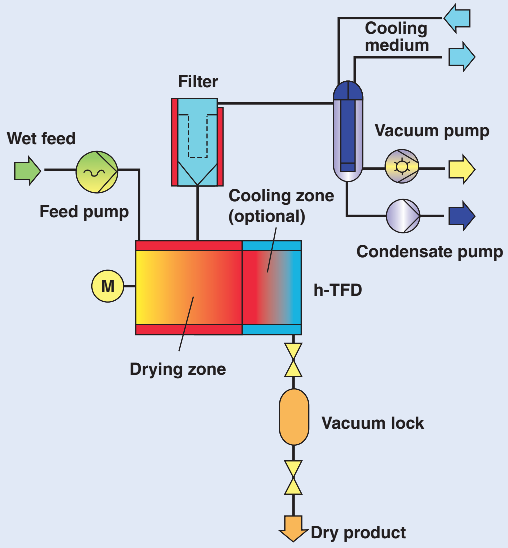
ΔT = Temperature difference between the heating medium and the boiling point of the volatiles, K

Typical U-values in a TFD range from 350 to 750 W/m²K. The relatively high U-values indicate that high heat flux

HORIZONTAL TFD APPLICATION

A complete h-TFD offers a closed, high-performance system. The figure, right, shows a h-TFD set up for the continuous vacuum drying of a pumpable press cake, which is temperature-sensitive. Briefly, the feed is metered into the unit, and drying occurs as the material is conveyed along the dryer length. The dried product is then discharged through an air lock at timed intervals. The volatiles are condensed and pumped back to the process. Once the dryer temperature and vacuum are established by standard control methods, the remaining operating variable becomes the feedrate, which is dependent only upon the material characteristics and solvent content.

With the new drying-system, solvent worth several hundred thousand euros is being recovered on an annual basis. Short residence time of only a few minutes makes it possible to dry the heat-sensitive product at relatively high temperatures, while maintaining high quality. Since the system is completely closed, with no sweeping gas required, no loss of solvent occurs and there is no airborne dust. Hence, a potential air pollution problem has been eliminated. Product hold-up in dryer is less than 10 kg. Energy requirements are low, about 1.1 kg of steam per kg of water and solvent evaporated. Operating costs have been reduced due to the efficient use of energy and labor. Some care is needed in the operation. Close tolerance between rotor blades and wall requires occasional adjustment of rotor blades. The clearance is important for maintaining heat transfer and high throughput. ❑



accompanies thin-film drying. The heat flux can be as high as 45 kW/m^2, which is similar to a flash dryer. Due to high U-values and high heat fluxes, the residence time in TFD is short. For free-flowing solids the typical drying time is 45 s to 1 min. For viscous products, which can stick to the wall and agglomerate, it ranges from 3 to 5 min. While rotor speed does influence retention time to a degree, this method is not recommended for the regulation of the residence time. Instead, regulation is done by modifying the inner geometry of the rotor without sacrificing either heat transfer performance or materials handling capability.

It is evident that product holdup in TFD is low, since just enough material is present to completely cover the heating wall in a thin layer. The product holdup depends on the consistency of the feedstock and its behavior during drying and is determined experimentally. This amount varies, depending on whether the feedstock is a free-flowing or pasty material or a filter cake.

Residence time distribution

In trying to prevent product degradation, it is of no use to secure a short mean residence time for the product in the heated zone if at the same time the residence time distribution is bad. A bad distribution of residence time means that although some of the feed goes through in less than the mean time, a certain portion will stay much longer than the mean time. Even a small portion of the total feed, which stays long enough to become degraded, will contaminate the whole product.

Owing to the rotor configuration of TFD, the feed pursues a helical path along the heated wall. Every particle of the feed follows in a strict sequence, thereby insuring that the treatment time is about the same for all portions of the feed. Figure 9 shows the residence time distribution of a reference material at various operating conditions. Of particular importance is Run 4, which shows the mean residence time of 60 s at a specific load of 195 kg/h-m^2. This is a particularly narrow span for a solids-handling piece of equipment.

Because of the three previously-mentioned factors (low holdup, short residence time and minimum back mixing), a TFD is scaled up with a high degree of assurance.

Particle integrity

Another important advantage of the TFD is retaining particle integrity. It would appear that high shear forces set up by the rotational speed and the narrow clearance would damage the basic structure of free-flowing solids. However, this is negligible as demonstrated from numerous experimental results. It has been proven that crystalline or even amorphous materials exhibit a marginal decrease of their particle size mainly due to abrasion of the edges rather than due to disintegration. For extremely shear-sensitive particles, such as pigments, it is recommended to investigate the particle integrity on a case-by-case basis.

Press cakes and slurries containing micro-size particles (1 to 5 microns) show a strong tendency to agglomerate and case-harden while drying. This is a natural phenomenon that fouls the heating surface of standard dryers. The elements of a TFD are designed to minimize the agglomerate build-up and to break up any agglomerates that have already been formed.

Concluding remarks

Adapting the thin-film principle to a rotary contact dryer offers the specialty chemicals, polymers, food and pharmaceutical industries a new dryer. Wet feedstocks — irrespective of their consistency and hazardous behavior — can now be continuously dried under product specific pressure and temperature at high drying rates and without a sweep gas. ■

Edited by Rebekkah Marshall

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Cooling Water Outlet Temperature:

EVALUATING THE BEST MAXIMUM VALUE

This article discusses the evaluation process — and tradeoffs — that must be considered to optimize recirculating cooling-water systems

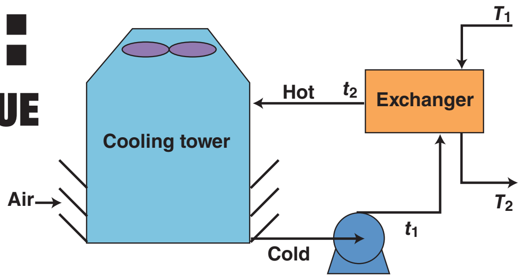

FIGURE 1. The components of a typical recirculating-cooling-water system are shown here with the various inlet and outlet streams labeled

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Recirculating cooling-water systems are essential parts of many facilities throughout the chemical process industries (CPI), including petroleum refineries, petrochemical, chemical and nuclear plants, and others. The cooling system shown in Figure 1 is widely used as a safe method for removing heat from various process operations.

Typically, the process engineer uses rules of thumb to define the cooling water flowrate, the temperature of the cooling-water return stream to the tower (t_2) and the inlet temperature of the cooling water stream to the heat exchanger (t_1) [1]. These criteria can be used to obtain quick and reliable estimates, and to optimize both system design and operation.

The temperature of the cooling-water return stream to the tower has a direct impact on the water flowrate and the required surface area for the heat exchange equipment. On one hand, for a determined inlet temperature of the cooling water stream to the heat exchanger (t_1), the use of a higher temperature of the cooling-water return stream to the tower (t_2) can help to decrease the quantity of cooling water that is required. Consequently, the ΔT_{LMTD} value calculated by the heat exchanger design equation [shown below as Equation (2)] decreases and the heat-transfer surface area thus increases (thereby increasing a fixed cost of the system). But this tradeoff is often worthwhile, since the cooling-water pumping costs (an operational cost) also decrease as a result of the reduced water requirements.

Efforts to achieve an economic balance between the temperature, the cooling water flowrate, and the required heat exchanger area indicate that the optimal temperature of the cooling-water return stream to the tower (t_2) occurs at the point of minimum total cost for cooling water and equipment fixed charges.

The purpose of this article is to show that selecting a higher temperature for the return stream to the cooling tower can often be justified, when one considers the impact of higher operational costs (in terms of energy and cooling water) in comparison to the fixed costs related to increased heat exchanger size.

Meanwhile, since scale formation is a common issue in cooling tower systems, this article also discusses how to use the Ryznar Stability Index (RSI) to provide another evaluation of the selected water temperature. The RSI gives a qualitative value of the water's tendency to form scale. Thus, an RSI evaluation guarantees the correct operability of the system with the derived optimum-return temperature. By using the calculated RSI value, the designer or process engineer can decide whether the calculated optimum return temperature should be used, based on the actual scale-forming tendencies of the water.

The basic equations

The procedure described below can help users to find the optimum temperature for the return stream to the cooling tower (t_2). It is based on the method described by Peters [2]. The energy balance in any heat exchanger system is given by Equation (1):

$$Q = wCp(T_2 - T_1) \quad (1)$$

Considering the heat exchanger design equation:

$$Q = UAF_G LMTD \quad (2)$$

Where the $LMTD$ is defined using Equation (3):

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left(\frac{T_1 - t_2}{T_2 - t_1}\right)} \quad (3)$$

Working out the value of w from Equation (1):

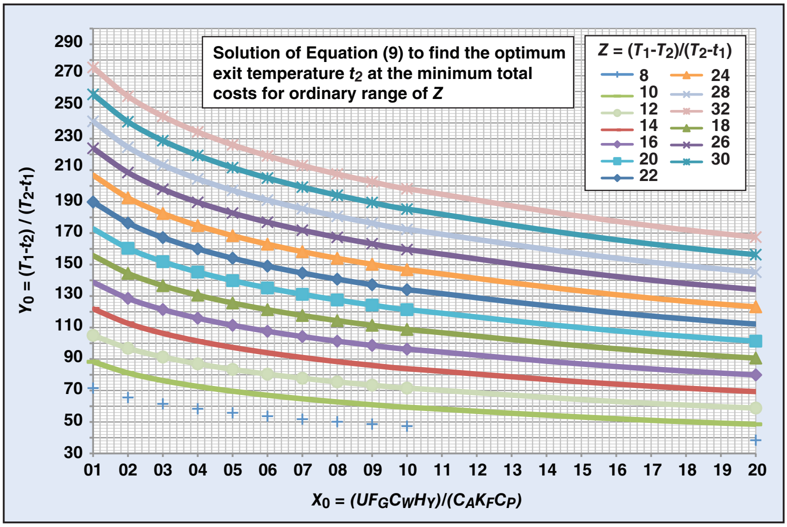

FIGURE 2. This figure shows the solution of Equation (9) for an ordinary range of *Z*. With the given known data, X_0 and *Z* are calculated using Equations (11) and (12), respectively. Then, the value of Y_0 can be read from this chart, and with this value, the desired value (t_2) can be obtained, using Equation (10)

TABLE 1. KEY PARAMETERS USED FOR THE EXAMPLE

Economic information	Year		
	1990[1]	2006[2]	2011
Cooling water costs, \$/lb	1.81×10^{-5}	Not calculated	3.63×10^{-5}
Cost per square foot of heat transfer surface, \$/ft^2	32.2	45.0	53.0
Chemical Engineering Plant Cost Index (CEPCI)[3]	357.6	499.6	588.9

1. The year 1990 was chosen to provide a representative range of time against which the current data could be compared.
2. The year 2006 was used as a reference to obtain the 1990 and 2011 values of the cost per square foot of heat transfer surface material. The price for year 2006 is an average price for a shell-and-tube heat exchanger type based on all-stainless-steel construction and may vary depending on the manufacturer [3].
3. Evaluated monthly by the staff of *Chemical Engineering* and printed along with historical values of this and other indexes on the last page of each issue.

$$w = \frac{Q}{Cp(T_2 - T_1)} \tag{4}$$

the annual cost for cooling water is QH_yC_w. From Equation (4), the operational cost (including the costs of the cooling water and pumping) is expressed using Equation (5):

$$CO = \frac{QH_yC_w}{Cp(t_2 - t_1)} \tag{5}$$

The fixed costs, which depend on the heat exchanger size, are calculated using Equation (6):

$$CF = AK_FC_A \tag{6}$$

Then, by combining the foregoing relations, the total annual cost is expressed according to Equation (7):

$$CT = \frac{QH_yC_w}{Cp(t_2 - t_1)} + AK_FC_A \tag{7}$$

Substituting the area (*A*) from Equation (2):

$$CT = \frac{QH_yC_w}{Cp(t_2 - t_1)} + \frac{K_FC_AQ \cdot \ln(\frac{T_1 - t_2}{T_2 - t_1})}{UF_G\left[(T_1 - t_2) - (T_2 - t_1)\right]} \tag{8}$$

Equation (8) contains the desired variable, which is the temperature of cooling water leaving the heat exchanger (t_2). Then, the corresponding optimum exit temperature can be found by differentiating Equation (8) — which reflects the annual sum of water costs and fixed charges — with respect to t_2, or more simply, with respect to the temperature difference (T_1–t_2), and setting the result equal to zero to find the minimum total annual costs. When this is done, Equation (9) is obtained:

$$X_0 = (\frac{Z + 1 - Y_0}{Y_0 - 1})^2 (\frac{1 - Y_0}{Y_0} + \ln Y_0) \tag{9}$$

Wherein:

$$Y_0 = \frac{(T_1 - t_2)}{(T_2 - t_1)} \tag{10}$$

$$X_0 = \frac{UF_GC_WH_Y}{C_AK_FC_P} \tag{11}$$

$$Z = \frac{(T_1 - T_2)}{(T_2 - t_1)} \tag{12}$$

Calculation of the desired (T_1–t_2) values is achieved using a numeric method. But the numeric method can be avoided by using the chart shown in Figure 2.

An example

The goal of this example is to design a counterflow cooling tower to cool 10,000 lb/h of gas, having a C_p of 0.24 Btu/lb, from 200 to 90°F, using water that enters the cooling tower at 85°F. To do this, we follow the steps described below, to calculate the optimum temperature for the return stream to the cooling tower (t_2) for the year 1990 and 2011.

For this example, it is assumed that the overall heat-transfer coefficient at the optimum conditions may be taken as 7.8 Btu/(h)(ft^2)(°F). The heat exchanger is to operate 8,400 h/yr, and the heat exchanger has a single shell pass and a single tube pass (1-1). Thus, it has a geometrical factor of 1 (Note: This measure depends on the geometrical arrangement of the shell and tube passes in the exchanger and serves as a correction factor that must be used with the log mean temperature difference for a countercurrent heat exchanger to accommodate the fact that the flow of two streams is more complicated than simple countercurrent or concurrent flow. The value of this factor can be obtained from graphs or equations typically reported in the literature). The rest of the required parameters are provided in Table 1.

Before carrying out the direct calculation of the optimum temperature for the return stream to the cooling tower (t_2) using Equation (9), some values that are required as inputs to this equation must be calculated. The procedure for obtaining these values, which are the parameters from Table 1, is briefly explained next.

Cost per square foot of heat-transfer surface material. The costs per square foot of heat transfer surface for 1990 and 2011 were determined, using the *Chemical Engineering* Plant Cost Index (CEPCI), as shown in Table 1 from the known cost for the year 2006 [3]. The results are shown in Table 1. The calculation is carried out using Equation (13):

$$Cost\ in\ year\ A = \frac{\left(\begin{array}{c} CE\ PCI\ Value \\ in\ year\ A \end{array}\right)}{\left(\begin{array}{c} CE\ PCI\ Value \\ in\ year\ B \end{array}\right)} \cdot \left(\begin{array}{c} cost\ in \\ year\ B \end{array}\right) \quad (13)$$

(CEPCI data is available at www.che.com/pci)

For the year 1990:

$$Cost\ in\ 1990 = \frac{(357.6)}{(499.6)}(45) = 32.2$$

For the year 2011:

$$Cost\ in\ 2011 = \frac{(588.9)}{(499.6)}(45) = 53.0$$

Cooling-water costs. The cooling water cost was calculated using Equation (14), which was obtained from Ref. *4*:

$$C_{S,u} = a(CEPCI) + b(C_{S,f}) \quad (14)$$

Table 2 shows the values of the parameters required for Equation (14).

The cooling water costs for year 1990 and 2011 were obtained by substituting the values from Table 2 into Equation (14).

For year 1990:

$$C_{S,u} = 0.0000725(357.6) + 0.003(5.07)$$

$$C_{S,u} = 0.04\ \$/m^3 = 1.81 \times 10^{-5}\ \$/lb$$

For year 2011:

$$C_{S,u} = 0.0000725(556.8) + 0.003(13.19)$$

$$C_{S,u} = 0.08\ \$/m^3 = 3.63 \times 10^{-5}\ \$/lb$$

Optimum outlet temperature of the water. Finally, once the cost per square foot of heat- transfer surface and the cooling water costs are calculated, these values are substituted in the Equation (9) for both years:

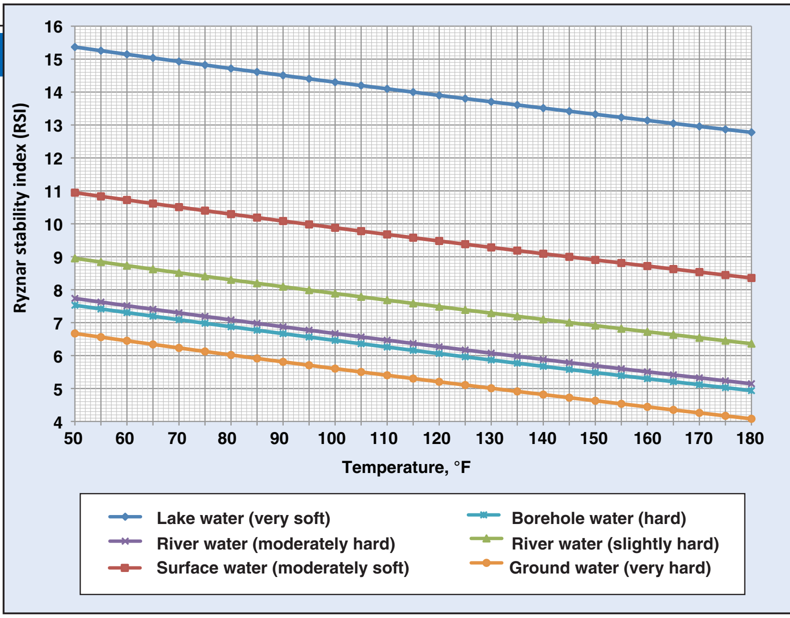

FIGURE 3. This plot shows the Ryznar Stability Index (RSI) data versus temperature for different types of water

TABLE 2. PARAMETER VALUES FOR WATER COST CALCULATION		
Price of energy for industry in U.S. $/GJ[4]	1990	5.07
	2011	13.19
Chemical Engineering Plant Cost Index (CEPCI)	1990	357.6
	2011	556.8
Cooling water cost coefficient, a (Grassroots plant with q = 10 m³/s, $/m³)	0.0000725	
Cooling water cost coefficient, b (Grassroots plant), $/m³	0.003	

4Calculated with a weighted average of the most common fuels used in U.S. Data obtained from the May 2011 Monthly Energy Review published by the U.S. Energy Information Administration

For year 1990:

$$X_0 = \frac{UF_G C_W H_Y}{C_A K_F C_P} =$$

$$\frac{(7.8)(1)(1.81 \times 10^{-5})(8400)}{(32.2)(0.1)(1)} = 0.37$$

$$Z = \frac{(T_1 - T_2)}{(T_2 - t_1)} = \frac{(200-90)}{(90-85)} = 22$$

It is obtained from Figure 2 that:

$$Y_0 = 16.2 = \frac{(200 - t_2)}{5}$$

Thus, t_2=119°F.

For the updated values, year 2011:

$$X_0 = \frac{UF_G C_W H_Y}{C_A K_F C_P} =$$

$$\frac{(7.8)(1)(3.62 \times 10^{-5})(8400)}{(53.04)(0.1)(1)} = 0.45$$

$$Z = \frac{(T_1 - T_2)}{(T_2 - t_1)} = \frac{(200-90)}{(90-85)} = 22$$

Once again, from Figure 2, it is obtained that:

$$Y_0 = 15.7 = \frac{(200 - t_2)}{5}$$

Thus,

$$t_2 = 121.5°F$$

From the calculations, it is observed that in 11 years the optimum temperature has increased by 2.5°F. This increase is justified for the rise in energy cost over that time period.

Ryznar Stability Index

Once the optimum temperature has been calculated, users should critically evaluate the use of this particular temperature using the RSI scale index, as mentioned earlier. This re-evaluation allows the user to know if — at the calculated optimum temperature — the likelihood for scale problems in the cooling water system rises (thereby creating potential problems and additional costs).

As noted, for a given inlet water, the tendency for a given system to form

NOMENCLATURE

- Q Exchanged heat, Btu
- w Flowrate of the cooling water, lb/h
- Cp Heat capacity, Btu/(lb)(°F)
- U Constant overall coefficient of heat-transfer determined at optimum conditions, Btu/(h) (ft^2)(°F)
- A Area of heat transfer, ft^2
- F_G Geometrical factor (depends on the geometrical arrangement of the shell-and-tube passes in the exchanger), dimensionless
- $LMTD$ Log mean temperature difference driving force over heat exchanger, °F
- T_1 Temperature of the hot process stream entering the heat exchanger, °F
- T_2 Temperature of the hot process stream leaving the heat exchanger, °F
- t_1 Temperature of cooling water entering the heat exchanger, °F
- t_2 Temperature of cooling water leaving the heat exchanger or temperature of the cooling water return stream to the tower, °F
- CO Operational cost, $/yr
- H_y Hours the exchanger is operated per year, h/yr
- C_w Cooling water cost assumed as directly proportional to amount of water supplied, $/lb
- CF Annual fixed costs, $/yr
- K_F Annual fixed charges including maintenance, expressed as a fraction of initial cost for completely installed equipment, dimensionless
- C_A Installed cost of heat exchanger per square foot of heat-transfer area, $/ft^2
- CT Total annual costs, $/yr
- $C_{S,u}$ Cooling water price for use in Equation (14), $/m^3
- $C_{S,f}$ Fuel price for use in equation (14), $/GJ
- CEPCI *Chemical Engineering* Plant Cost Index, which is an inflation parameter for projects in the U.S.[5]
- a Cooling water cost coefficient, $/m^3
- b Cooling water cost coefficient, $/m^3
- q Total water capacity, m^3/s
- RSI Ryznar Stability Index, dimensionless
- pH pH value of studied water
- T Water temperature, °F
- pH_S pH value of water saturated in $CaCO_3$
- S Coefficient of the equation to calculate pH_S, dimensionless
- t Coefficient of the equation to calculate pH_S, dimensionless
- D Coefficient of the equation to calculate pH_S, dimensionless
- B Coefficient of the equation to calculate pH_S, dimensionless
- TDS Total dissolved solids, mg/L
- TH Total hardness as Ca^{2+}, mg/L
- $T_{Alkalinity}$ Total alkalinity, mg/L

5. Calculated with a weighted average of the most common fuels used in the U.S. data obtained from the May 2011 Monthly Energy Review published by the U.S. Energy Information Administration

TABLE 3. TYPICAL WATER ANALYSIS, PPM

Parameter	Lake water (VS)	Surface water (MS)	River water (SH)	River water (MH)	Borehole water (H)	Ground water (VH)
pH	6.3	6.8	7.4	7.5	7.1	7.1
Total alkalinity	2	38	90	180	250	470
Total hardness	10	53	120	230	340	559
Total dissolved solids	33	88	185	332	400	1,670

VS: Very soft MS: Moderately soft SH: Slightly hard MH: Moderately hard H: Hard VH: Very hard

calcium carbonate scale can be predicted from calculations involving calcium hardness, total alkalinity, total dissolved solids, pH and temperature. The most commonly used indicators are the Langelier Saturation Index (LSI), the Ryznar Stability Index (RSI), and the Puckorius (Practical, Predictive) Scale Index (PSI) [*5*].

For this article, the RSI was chosen to predict the scale tendencies. The RSI has several advantages over the other techniques: it always gives positive values, provides better estimates of the severity of scaling tendencies [*6*], and it is not possible for low-hardness and high-hardness waters to have the same value (which can sometimes happen with the LSI [*7*]).

RSI is defined by Equation (15) [*8*]:

$$RSI = 2pH_s - pH \quad (15)$$

Usually, pH_s is calculated by the following relation [*8*]:

$$pH_s = 9.3 + S + t - D - B \quad (16)$$

Where:

$$S = 0.1\left(\log\left(TDS\right) - 1\right) \quad (17)$$

$$t = -13.12\log\left(\frac{T + 459.67}{1.8}\right) + 34.55 \quad (18)$$

$$D = \log\left(TH\right) - .04 \quad (19)$$

$$B = \log\left(T_{Alkalinity}\right) \quad (20)$$

Some typical values corresponding to different types of water for calculating the parameters S, D and B are shown in Table 3 [*9*].

For practical purposes, the RSI was calculated considering a wide range of temperatures and the values of the parameters shown in Table 3 for all types of water. The results are presented as a graph in Figure 3. This graph helps one to easily determine the RSI, given a particular type of water and a cooling water temperature.

Using the optimum temperatures obtained from the example for year 1990 and year 2011, and considering river water (slightly hard), RSI can be determined easily from Figure 3.

For year 1990 and t_2 = 119°F, the value of RSI according to the graph is 7.5. From Table 4, it is found that with this RSI value, scale may be dissolved. It is suggested to use the value of 7 > RSI > 6 as a breaking point, instead of using the published value of RSI = 6.

Now for year 2011, where the temperature to evaluate for scale-forming tendency is higher, the following exercise analyzes also the effect of the type of water on the RSI value. The results of evaluating the RSI for all types of water, considering t_2 = 121.5°F, are shown in Table 5.

As it can be observed, the optimum temperature obtained for year 2011 is higher than that for year 1990. For both years, comparing the results of the RSI obtained for the same type of water (river water, slightly hard), this temperature increase gives a lower RSI value (7.4). In this case, given the characteristics of the water, scale also may be dissolved, but the RSI value tends to be in the limit of being in chemical balance.

For the same example, and considering the case of borehole water that is relatively hard, this increase on the temperature of the cooling water return stream to the tower (t_2) is even more convenient, because, according to Table 4, the RSI (6.0) shows that the system is in chemical balance — thus, the water is essentially neutral. Hence, operating with a higher cooling water outlet temperature is justified and should provide no additional problems in terms of added scale.

Readers should note that the RSI relates only to scaling — not to corro-

TABLE 4. CALCIUM CARBONATE SCALING TENDENCIES		
Properties	**RSI Value**	**Interpretation**
Scale may be dissolved	> 8.5 >7.5	Aggressive scale removal Scale removal expected
RSI ≈ 6 Generally in chemical balance	7.0 6.0 5.0	Essentially neutral. Mild scale formation or removal possible
Scale may form and precipitate	< 4.5	Scale formation anticipated

TABLE 5. RSI VALUES FOR DIFFERENT TYPES OF WATER USING THE 2011 OPTIMUM TEMPERATURE FROM THE EXAMPLE						
	Lake water (VS)	Surface water (MS)	River water (SH)	River water (MH)	Borehole water (H)	Ground water (VH)
RSI	13.9	9.4	7.4	6.2	6.0	5.2

VS: Very soft MS: Moderately soft SH: Slightly hard MH: Moderately Hard H: Hard VH: Very hard

sivity — even though these two troublesome conditions are often loosely correlated. The RSI and other indicators were never intended as measures of corrosive tendencies towards mild steel or other metals. They describe only the water's corrosivity towards an existing calcium carbonate scale [*5*].

From this exercise, one can see that the optimum tower-inlet temperature increase is justified for the rise in energy cost, but this increase may result in higher scaling problems. To avoid scaling, the temperature may need to be decreased, or the scale problem may need to be managed through the use of additives. One should carry out a cost-benefit analysis for the use of anti-scale additives, to compare with the costs of decreasing temperature.

Final thoughts

With rising energy costs, the prospect of increasing costs for cooling water pumping makes it more attractive to select a higher optimum temperature of the cooling-water return stream to the tower (t_2). This tradeoff may modify the current rules of thumb used for selecting the temperature.

According to the example presented in the article, the cooling-tower return temperature should be 121.5°F. This is 1.25% higher than the temperature recommended using the existing rules of thumb, which suggested using a temperature of 120°F [*1*].

However, further analysis of the optimum temperature found that, with the given conditions, the water may have a slightly scale-dissolving tendency. Not only does this depend on the water temperature, but also on its chemical characteristics. The RSI calculation for a wide range of critical parameters, such as the pH and the temperature, provides a panoramic view of the impact for different operation conditions.

If water has a scale-formation tendency at the optimum temperature, then lowering this temperature or adding an anti-scale additive should be considered for the most cost-effective tower performance. ■

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Reciprocating Compressors Startup and Capacity Control Methods

Procedures and guidelines that will help operators minimize energy consumption and maintenance requirements of compressor systems

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There are two main reasons why compressor capacity regulation is used. The most prevalent reason is to adjust the suction flow to match the process demand. The second reason is to save energy. As a rule, capacity control is determined by the compressor discharge pressure. Compressor capacity-control methods are utilized to maintain a required delivery under variable process conditions [*1*].

In this article, methods for controlling the capacity of reciprocating compressors are presented in detail. These methods are bypass recycling, unloader valves, clearance pockets, stepless valves and variable-speed drives. The advantages and disadvantages of each method, as well as their applications, are presented.

In addition, auxiliary devices and packages used under transient conditions (namely, during startup, shutdown and maintenance) and during normal operation are described. Barring devices (gear), hydro-jacking systems, frame lubrication systems and cylinder lube oil systems are explained. Finally, a recommended procedure for starting up and stopping a reciprocating compressor in single and parallel operation is presented, along with some major points that should be taken into consideration by plant operators.

Capacity control methods

Bypass control. As the name implies, this control method uses an external bypass around the compressor to recycle gas from the compressor discharge to the inlet, or to the atmosphere in the case of an air compressor. The take-off point for the bypass must be downstream of a heat exchanger so that cooled gas will be spilled back to the suction. If there is no exchanger in the discharge, the bypass must branch into the suction line upstream of an exchanger.

Alternatively, a cooler may be placed in the bypass line. In any case, the bypass should tie-in upstream of a suction knockout drum so that any condensate resulting from the expansion cannot enter the compressor. Bypass control is preferred over other control methods because of its smoothness, simplicity and low initial costs. It is, however, inefficient because excess compressor capacity is expanded across the control valve in the recycle line. For this reason, this method is commonly accompanied by the use of inlet valve unloaders or clearance pockets, which reduce compressor capacity in discrete predetermined steps [*1–5*].

In multistage compressors, a bypass around the first stage or a partial bypass can be used. Because of decreasing discharge pressure, the absolute power input would be reduced in this way, although the mass flowrate is increased as shown in Equations (1) and (2).

$$W = \frac{\dot{m}R}{M_w}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right]\left(\frac{k}{k-1}\right)T_{in} \quad (1)$$

$$Q = PD\left[1 - C\left(\frac{p_2}{p_1}\right)^{1/k}\right] \quad (2)$$

Where:
W = Compressor power
$\dot{m}$ = Rated mass flow
M_w = Gas molecular weight
p_2/p_1 = Compressor ratio
K = Gas specific-heat ratio
T_{in} = Gas inlet temperature
PD = Compressor piston displacement
Q = Compressor flow
C = Compressor clearance volume
R = Ideal gas constant

It must be remembered that a reduction in the flow to the second stage causes a drop of all inter-stage pressures, and consequently it can lead to excessively high-pressure ratios and discharge temperatures in subsequent stages. Moreover, this pressure shifting may cause an overload in the last stage. Thus, there should be a limitation on the maximum by-passed flow in multistage equipment. The minimum capacity that can be obtained depends on the number of compression stages.

The more stages used for a given overall compression ratio, the wider the achievable control range.

Inlet valve unloaders. Valve unloaders are mechanisms that are held open or bypass one or more cylinders' inlet valves at each end of double-acting cylinders. This provides complete unloading of one or both ends of the cylinder. For a single-cylinder compressor stage, valve unloaders can achieve three-step unloading that provides nominal cylinder capacities of 100, 50 and 0%; and two-step unloading that provides cylinder capacities of 100 and 0%. Thus, considering two cylinders per compressor, the capacity can be controlled in five stages 100, 75, 50, 25 and 0%.

Three-step (100, 50 and 0%) capacity control is used on some general purpose machines, such as air compressors. This method of control requires more cyclic actuation of the unloaders than five-step control. Therefore, three-step control is more detrimental to machine components — especially to valves.

Standard API 618 [*6*] introduces three unloader types: finger, plug and port. The finger type unloader is not recommended when other types are viable. Plug type unloaders should be installed on extra suction ports. A minimum number of valve unloader steps should be considered in order to maximize compressor reliability.

Unloaders should be pneumatically operated by instrument air and equipped with positioner indication. The unloader actuator should be sized to operate on minimum air pressure as well. Nitrogen purge ports should be provided for all types. The unloader stem packing should be provided with a lantern ring and a nitrogen connection for the purging of leak gas. All purge and vent connections should be piped up to a single purge and vent connection. All lines to and from unloaders are to be AISI 316 L stainless steel [*6, 7*].

It should be noted that there are some restrictions on actuator sizing at suction pressures higher than 70 barg.

Clearance pockets. Cylinder delivered capacity can be reduced by increasing the cylinder clearance volume; this is done by a clearance pocket. There are two types of pockets: fixed and variable. Opening the pocket reduces the cylinder's inlet volumetric flow by trapping additional gas in the larger clearance volume at the end of the piston stroke. Consequently, clearance-pocket flow adjustment is frequently applied to the head end only, leading to a maximum of 50–60% reduction in the head end capacity and a 25–30% overall capacity reduction in a double acting cylinder [*8*].

Valve unloaders combined with clearance pockets can provide five-step unloading leading to nominal cylinder capacities of 100, 75, 50, 25 and 0% in a compressor with only one cylinder. This means that 75% capacity is achieved by opening the head-end clearance pocket, 50% load makes use of the crank-end valve unloader, 25% load makes use of the crank-end valve unloader and head-end clearance pocket simultaneously and 0% load is achieved by activating the valve unloaders on both ends of the cylinder.

At compression ratios below 1.7, the pocket volume becomes very large relative to cylinder size, thus, it should not be used in this case [*2*].

Clearance pockets can be arranged for local manual operation, manual/pneumatic operation, or automatic/pneumatic operation. However, the automatic/pneumatic type is preferred.

Using clearance pockets is usually restricted to services with cylinder inlet pressures less than 70 barg.

Stepless capacity control. A stepless capacity control system, known as reverse flow control, can provide a capacity range of approximately 100 to 40% of rated capacity; however, more turndown ratios have become feasible recently. In this system, an unloading device is fitted to each suction valve. At partial loads, the unloading device does not allow the inlet valve to close when the piston is at its bottom dead center position, but rather delays the closing in a controlled way. Consequently, an amount of gas, which can be adjusted, is allowed to return to the compressor inlet manifold before the compression starts. As the piston velocity increases in the compression stroke, it pushes gas in reverse flow back out the inlet valve faster and faster. This creates a larger and larger gas drag force on the inlet valve plate that eventually overcomes the unloader force and closes the inlet valve. Applying this type of control requires economic justification. In general it is not recommended because of limitations in performance accuracy and fatigue consequences on the valve plate and seat [*7*].

Inlet throttling. This method of capacity control is not recommended because pressures lower than normal inlet pressures cause higher piston-rod loads and discharge temperatures. Consequently, the mass flow range that can be safely achieved in this regard is normally too low to suit process needs.

Variable speed drive. Reciprocating compressors should normally be specified for constant-speed operation in order to avoid excitation of torsional, acoustic and mechanical resonances. When variable-speed drivers are used, all equipment should be designed to run safely throughout the operating speed range, up to and including the trip speed [*6*]. Gas engine drives, which could operate in a 75 to 100% speed range, make use of variable speed method for flow control. Operation below 75% is normally done by bypass control and inlet valve unloaders or clearance pockets [*9–12*].

Startup unloading. Some means of unloading for startup is usually required because drivers lack sufficient torque to accelerate the train rotating parts under positive displacement compression loads. Inlet valve unloaders and external bypasses are the two most commonly used methods for startup unloading. The machine should be capable of starting with operating suction pressure on each stage. An unloading system may be designed to operate automatically on a voltage dip for service continuity where automatic reacceleration of motor driven compressors is required.

Compressor operation modes

Auto mode. In the automatic (auto) mode of operation, the compressor meets the demand of all the main processes, including all cases of operation requested by the equipment datasheet. In some cases, operation with nitrogen is also requested for pre-commissioning, cleaning purposes, testing of the automatic functions and leak testing of other equipment. In this mode, the machine is controlled automatically.

When the compression gas is changed, set point values are only adjusted for the new condition with no change in control procedure.

Maintenance mode. In this mode, during which the compressor is at standstill, all auxiliary drivers (such as auxilliary oil pump, water pumps, and bypass valves) can be manually started by operators.

Off mode. In this mode, all drivers are out of operation, and standby functions are disabled. This mode will be activated only when the compressor is not running; otherwise, selecting the "off" mode will have no effect on the operation of the compressor.

Mode selection between auto, off and maintenance is done within the distributed control system (DCS). A compressor is provided with several local instrument boards (LIBs), with gages for indicating pressure, temperature, flowrate and level of process gas, lubrication oil and other utilities (such as water, air and nitrogen), during compressor operation in all modes. Additionally, the LIB shows whether local motor-control stations are also available for local starting and stopping of auxiliary equipment in maintenance mode. Auxiliary motors also have local motor-control stations located within a few meters from the associated auxiliary-drive motor. Visualization and control of the compressor operation is normally done from a central control room (CCR) station.

Compressor auxiliary systems

Hydro jacking device. For large reciprocating compressors and motors, manufacturers consider adding an additional system to the equipment in order to lessen the breakaway torque in the bearings in transient conditions (startup, shutdown or maintenance). For this purpose, one oil pump is usually added in series after the lubrication pump. A jacking oil pump assures lasting oil film existence between bearing and journal (shaft) surfaces at the instant a rotation commences.

Barring device. The compressor should be fitted with a barring device (barring gear), which can be manual, electrical or pneumatic. This device is used only for maintenance, while the compressor is stopped. For compressors with a rated power equal to or greater than 750 kW (1,000 hp), and torque requirement equal to or greater than 1,600 Nm (1,200 ft-lb), either the electrical or pneumatic type should be considered. Manual barring may be achieved by turning the flywheel. The barring device should be designed so that gas forces acting on the pistons cannot cause the compressor either to accelerate or to rotate in the reverse direction. With the barring device engaged, it should not be possible to start the compressor [*6*].

Manual barring devices should have a locking feature, whereas pneumatic and electrical devices should incorporate appropriate safety switches or measures for interlocking with the main driver. Moreover, a pneumatic system should be equipped with an air-supply inlet filter and oiler, and a four-way control valve for operation in forward or reverse directions.

The barring device should be designed to enable barring over the pressurized machine at suction pressure. Before the barring device is enabled to start, the hydro jacking of compressor and main motor have to be started, and the minimum pressure must be obtained.

The barring device operates with a local push button (forward and reverse) on the local panel near the motor. The barring device motor works only while one of the push buttons is pressed; release of the button stops the barring device and locks the crank gear. When maintenance is finished, the stopped barring gear must be moved back in the "turn wheel free" position, which is displayed on the DCS system.

Compressor frame lubrication system. A frame lubrication system is pressurized through two electrically driven oil pumps. One is used for normal operation (main pump) and the second is in standby mode as an auxiliary pump for necessary conditions. A crankshaft-driven main pump is allowed per API 618, but for large machines, each oil pump should be individually equipped with an electrical motor.

In auto mode, the main oil pump has to be started before starting the compressor main motor. Generally, pre-lubrication time is set for approximately one minute. When pre-lubrication time has passed, the startup condition "Pre-lubrication time passed" is activated and displayed on the DCS.

When neither the low-lube-oil pressure alarm nor the low low-lube-oil pressure alarm is activated, the hydro jacking pump and subsequently the main motor are permitted to start. During normal operation of the compressor, the standby oil pump should be in auto mode and start automatically when a low-oil-pressure alarm is triggered on the DCS.

If the low low-oil-pressure signal lasts more than a specified time (usually 2 s), the compressor main motor should be stopped immediately by a shutdown (trip).

Since the compressor is stopped normally, post lubrication is started; this means that the lubrication oil pump will continue to run for a minimum of approximately 2 min. When post lubrication time has passed and the normal stop sequence is accomplished, the oil pump can either be stopped by the operator or continues running. Note that in maintenance mode, the frame oil pump can be started individually.

Moreover, the operator can enable the oil heater by pushing a button in the DCS. If the heater is enabled, the temperature of the frame lubrication oil is controlled by the switch controller inside the heater. Note that the crankcase oil temperature should not exceed 70°C (160°F).

For safety reasons, the heater is interlocked and must be switched off when the low "low oil level" is activated. It should be noted that during compressor operation, the heater shall always be stopped.

Compressor cylinder lubrication system. The cylinder lubrication system is pressurized by electrically driven multiple-plunger-type oil pumps (plunger per point) or by the divider block method. However, using a divider block system is not recommended because of complexity and low reliability.

The cylinder lubrication system will be started locally if the cylinder oil-tank level is fulfilled (above low level).

When the pre-lubrication time (approximately 2 min) has elapsed, the compressor start condition "cylinder pre lubrication time passed" is fulfilled and displayed in the DCS, and the over lubrication time starts. The total time of the cylinder lubrication pumps running without the main motor running is cumulative via cylinder lubrication time.

The compressor main motor must be started within the over lubrication time (approximately 5 min) to prevent accumulation of oil in the cylinder.

If the total over-lubrication time has elapsed and the main motor is still not running, a compressor main-motor-start inhibit will be activated and an alarm "remove oil collected in cylinders before re-start" appears on the DCS.

Resetting the "over lubrication time" alarm will be done by slowly rotating the compressor for a number of revolutions (usually two or three) by a barring device to distribute the accumulated lube oil among the compressor cylinders. At this time, the alarm can be reset in the DCS manually.

During cylinder barring-over or maintenance operations, the cylinder lubrication should not be activated.

At compressor normal stop, post lubrication time starts for approximately 2 min, and the cylinder oil pump switches off automatically afterwards. By pressing the emergency stop, the cylinder oil pump stops immediately and no post lubrication is carried out. The cylinder lubrication-oil pumps should run while the compressor main motor is running. In the event of low flow, an alarm will be triggered in the DCS and in the case of low low-flow, compressor trip action will be activated.

Moreover, the operator can enable the oil heater by pushing a button in the DCS. If the heater is enabled, the temperature of the cylinder lubrication oil is controlled by a switch controller inside the heater body. The heat density of the heater should be limited to 2.3 W/cm^2 (15 W/in.^{2}). A temperature switch prevents overheating. An "over temperature" switch activates an alarm and switches off the heater. For safety reasons, the heater is interlocked and must be switched off when the low "low-oil level" is activated.

Compressor cooling water system. Minimum cooling water flow to the cylinders and packings is one of the allowed conditions for starting the main motor. As long as low flow is detected during normal running, an alarm will be triggered in the DCS. In a closed cooling cycle, the system is pressurized through two electrically driven water pumps. One pump is used for normal operation (main pump) and the second is in standby mode as an auxiliary for necessary conditions. For the compressor to get a permissive start, it is necessary to run the main water pump in order to deliver a minimum required flow of cooling water. Auxiliary water pumps can be started, as long as the water-pressure or low-flow alarm is activated in the DCS.

Main motor purging system. The main-motor purge system is an independent working system. In EEx"p" drivers, the motor becomes explosion-proof by pressurizing its cage. The main drive motor should be started only if it has been purged with air for a period of time specified by the motor manufacturer. (Purge time is approximately between 30 to 60 min).

Preparation before starting

Before the compressor is allowed to be started, the following checks and conditions have to be fulfilled in the field:

Check if the cooling-water supply valves are open (for inter-stage gas- and oil-system coolers). Moreover, the cooling water specifications, such as flowrate, temperature and pressure, should be verified with design conditions by local instrument devices. Check the oil level in the tanks; it has to be above the minimum level on the oil sight glass.

Check if the lube oil systems are ready for operation and the manual valves are in the correct position.

Drain valves of all process pipes and vessels (including pulsation dampeners, inter-stage coolers and separator drums) have to be checked for the presence of liquid. If liquid is present, it should be drained. Otherwise, liquid would be carried over to the cylinder and damage the compressor when starting the main motor.

Check if all utilities including instrument air, hydraulic oil (for high-pressure valves actuation) and nitrogen for purging of pressure packings are available.

The main drive motor should be started only if it has been purged and the pre-lubrication time of the crank gear and cylinder have passed. This should be considered at the time of starting the compressor.

Check the operability of all voltage circuits and the shut down system.

The oil pumps can be started at minimum ambient temperature, whereas the compressor only when the lube oil temperature is above the minimum temperature recommended by the manufacturer (about 15°C). Hence, frame and cylinder lubrication heater and tracing system should be on (temperature start permissive of motors).

The compressor should be barred over by a barring device (for two to three revolutions) to ensure that liquid is not present in cylinders.

Start inter-lock system. Before compressor startup, several parameters (such as level, pressure, temperature and flowrates) are checked and compared with set point values, specified by the compressor manufacturer in the set point list (set point lists include all instrument devices' set values specifying alarm or trip signals). Afterwards, compressor start permission can be passed by the DCS and the next step will be executed. Note that interlocks will only prevent the start sequence from continuing, or will interrupt the start sequence unless all specified conditions are fulfilled; interlocks are not trip signals.

Depending on the manufacturer and user concerns, an interlock system could be varied, but interlocks (alarm conditions) below are considered in most applications. An alarm is a signal generated automatically from an irregular state, which does not lead to a compressor shutdown.

The start sequence will be aborted if an alarm occurs. In other words, an alarm will inhibit the start of the machine as long as it is active. Alarms and warnings do not stop the compressor. Causes of alarms must be investigated and rectified immediately by operators, or else they could cause the machine to trip (shutdowns). The compressor is ready to start if none of the following alarms are active:

- Low lube-oil tank level
- Low cylinder oil-tank level

- Low suction pressure
- High discharge pressure
- Low packing-purge pressure
- Low cylinder lube-oil flow
- Low crank-gear oil pressure
- Low crank-gear oil temperature
- High oil-filter differential pressure
- Bypass valves over stages are opened
- Barring gear is in safe position (barring gear disengaged)
- High level in separators
- Low compressor cooling-system flow
- Compressor has been barred over for a minimum of 1 min (mandatory if cylinder over-lubrication time has elapsed before)
- Crank-gear lube-oil pump pre-lubrication time has passed
- Cylinder pre-lubrication time has passed

A trip is an irregular state that requires an immediate and automatic shutdown of the compressor in order to avoid damage to the equipment and personnel. Note that prior to shutdown an alarm will be indicated in the DCS to warn the operators. The following trips commence the normal stop procedure at any time of operation.

The compressor is ready to start if none of the following shutdown conditions are active. The start sequence will be aborted if a trip occurs.

- High high-suction-gas temperature
- High high-discharge-gas temperature
- Low low-gas pressure on suction side
- High high-level in separators
- High high-main-bearing temperature
- High high-vibration on compressor casing and rod drop low low-upstream-bearings oil pressure
- Low low-cylinder lube-oil flow

Starting sequence of the compressor in auto mode. Regarding compressor-package control philosophy, it could be started from the DCS or LCS (local control station). After startup, all functions will be carried out through the DCS automatically. Bypass valve controllers should be activated by the operator when the compressor start sequence has been completed successfully. For the compressor starting procedure to be deemed successful, the position of the manual valves must be as follows:

- Gas suction isolating valves are open
- Gas discharge isolating valves are open
- Blow-off valve and vent valve to safe location, flare, and atmosphere are closed
- Control valves and isolation valve on separator drain are open
- Shut-off valves for pressure instruments are open
- Bypass of separator drain is closed
- Separator drain valves have been put in auto mode, and their relevant interlocks are activated

The preconditions for startup sequence are as follows:

1. All heaters, including oil heaters and the motor space heater, must be switched on.
2. Purge air must be available. If the purge time of the main motor has elapsed, the message "purge end" is displayed in the DCS.
3. The compressor has been barred over. To do this, first the compressor's operating mode should be changed to maintenance mode in the DCS, and the operator in the field should engage the barring device manually. Then, the frame oil pump is started and compressor/motor jacking oil pumps will be started (with delay) afterwards because barring device operation is only possible as long as the hydro-jacking inlet-oil pressure alarm is not activated. At this time, barring over should be enabled from the local panel by pressing two buttons for operation of barring gear in clockwise and reverse direction. It should be noted that, none of pre-lubrication time, low-oil-temperature, or the lube-oil-pressure alarms should be activated. When barring over time (approximately 5 min) has elapsed, the operator should de-engage the barring device and remove any oil.
4. No low cooling-water-flow alarm.
5. No low-level alarm on interstate separators.
6. No low-level alarm on oil tanks.
7. No high-temperature alarm on the main motor.

Permission to start. At this stage, the frame oil pump is started, and the jacking pumps of the main motor and compressor will be started if pre-lube time, low oil temperature and low-pressure alarms are not active. Additionally, the following conditions should be satisfied:

- No low-suction-pressure alarm
- No high-discharge-pressure alarm
- Barring device in safe position (disengaged)

Next, the cylinder lubrication pump will be started if all preconditions are fulfilled. This will also start the cylinder pre-lubrication timer. If the cylinder pre-lubrication time has elapsed, the compressor start condition "cylinder pre-lubrication time passed" is fulfilled and will be displayed in the DCS and over-lubrication time starts afterwards. The cylinder lubrication timer should be programmed as a cumulative timer. If it is not reset, it will accumulate the total time of the cylinder lubrication pump running while the main motor is not running up to the over-lubrication time limit. During this time range, it allows multiple startups as long as they are within the over-lubrication time.

If the over-lubrication time has elapsed and the main motor is not running, the start-up sequence will be cancelled and an alarm to remove collected oil appears in DCS. Accumulated oil in the cylinders has to be removed by turning the barring device before another starting sequence commencement.

Since cylinder lube-pump start is the last step in the start sequence prior to motor starting, impact on plant availability is minimized.

If the main motor starts while over-lubrication time has not elapsed, motor space heaters and lube oil heaters will be stopped at the same time. Moreover, for approximately 2 min, the low low-pressure trip on the compressor suction will be overridden to allow a stable condition without tripping the machine. After that, the overridden function will be deactivated.

The jacking oil pump will be stopped 1 min after a successful start of the compressor. In this position, the compressor is now running at no load condition. In order to obtain the desired discharge pressure for each stage, the bypass valve setting must be performed manually. For this purpose, the first stage bypass valve shall be enabled to control suction pressure automatically. Note that automatic control capability of all bypass valves will have been deactivated during startup. At this time, close the second-stage discharge to first stage's discharge bypass valve gradually until the desired second-stage discharge pressure is obtained. For the last stage, the desired pressure is obtained by closing the discharge isolating valve. In this way, the stroke position of the previous bypass valves are slowly adjusted to control pressure between each stage. Thus, the compressor runs in 0% capacity continuously and bypass valve positions are set for 0% capacity (or 100% turndown). It is of high importance to close the bypass valves slowly and gradually to stabilize the conditions and prevent overshooting of discharge pressure.

Note that in compressors, which are only started by bypass method, the starting procedure is performed with bypass valves fully open. If an unloader valve also exists, the bypass valves are fully closed and unloader devices open the inlet valves, until compressor reaches the rated speed at 0% capacity. Due to increasing suction-valve temperature in this condition, the compressor is usually run between 5 to 10 min as maximum.

After that, the performance step is increased by loading each cylinder chamber in the sequence shown in Table 1 for a compressor with two double-acting cylinders (HE = cylinder head end, CE = crank case cylinder end).

After a period of time (adjustable between 5 to 15 min), in each performance step, solenoid valve activation on unloaders will be automatically switched over. This switching prevents increasing the temperature of the suction valve. Moreover, if purchase requisition states that the compressor shall run at 0% capacity for a long time, the compressor should run at 25% step capacity, and total flow is recycled by bypass valve thereafter. Due to unbalance matters, minimum load on each stage could be changed to 50%. Note that compressor mechanical stability must have priority over energy consumption.

Bypass valves

In this section, we investigate the capacity control procedure of a reciprocating compressor with bypass valves in two stages. As a principle rule, each stage is controlled with one bypass valve, which is independent of the other bypass valves, but valve setpoints are adjusted together and finalized during the compressor commissioning period. Per API recommendation, the maximum predicted discharge temperature on each stage should not exceed 150°C (300°F). This limit applies to all specified operating and load conditions. However, in actual designation, this limitation is decreased to 135°C (275°F) by compressor manufacturers.

To control discharge temperature, pressure ratio or differential pressure is a parameter that is considered as a controlling value. Designers mention relevant settings in the instrument set-point-list document.

Figure 1 shows an example of a typical capacity-control diagram. In this figure, PCV is pressure control valve and PT is pressure transmitter. The main objective of the capacity control is to maintain constant suction pressure (PT1). The capacity of the compressor is controlled by bypass over the first stage (PCV1) and bypass over the second stage (PCV2). If compressor suction pressure PT1 is decreased, the first stage bypass valve (PCV1) could be opened continuously up to a percent of stroke (between 50 to 60%) until suction pressure (PT2) is above the setpoint of the minimum suction pressure in the second stage.

If suction pressure (PT1) still falls, controller will open the second to first stage bypass valve PCV2. This bypass valve will open continuously to 100%. Now the first stage bypass (PCV1) will be able to control the main suction pressure (PT1) with PCV2 in parallel. Thus, the desired pressure is obtained by high-pressure gas, which is recycled through first- and second-stage bypass valves.

When both valves are fully open, the compressor will operate in full recycle mode, until suction pressure is increasing again. When suction pressure is increasing, the bypasses will close in reverse.

Rapid valve movement causes rapid load change on the compressor. To prevent this from happening, the bypass characteristic is linear and the stroke travel rate is approximately 2 min; however, this rate can be adjusted during compressor commissioning.

Please note that, in case of higher capacity requirements, increasing the suction pressure (PT1) could increase the compressor capacity.

Bypass and unloading valves

In most process applications, compressor capacity control is established by a suction pressure controller acting on the bypass valve and step capacity control (0, 25, 50, 75 and 100%). The

TABLE 1. LOADING SEQUENCE FOR A COMPRESSOR WITH TWO DOUBLE-ACTING CYLINDERS

For 0% load (X = Loaded; 0 = not loaded)				
	Cylinder A		Cylinder B	
Time (min)	HE	CE	CE	HE
Maximum (5-10 min)	0	0	0	0
For 25% load (X = Loaded):				
	Cylinder A		Cylinder B	
	HE	CE	CE	HE
0	X	0	0	0
5	0	X	0	0
10	0	0	X	0
15	0	0	0	X
20	Return to 0 Min.			
For 50% load (X = Loaded):				
	Cylinder A		Cylinder B	
Time (min)	HE	CE	CE	HE
0	X	0	X	0
5	0	X	0	X
10	Return to 0 Min.			
For 75% load (X = Loaded):				
	Cylinder A		Cylinder B	
Time (min)	HE	CE	CE	HE
0	X	X	X	0
5	0	X	X	X
10	X	X	0	X
15	X	0	X	X
20	Return to 0 Min.			
For 0% load (X = Loaded):				
	Cylinder A		Cylinder B	
	HE	CE	CE	HE
No Limitation	X	X	X	X

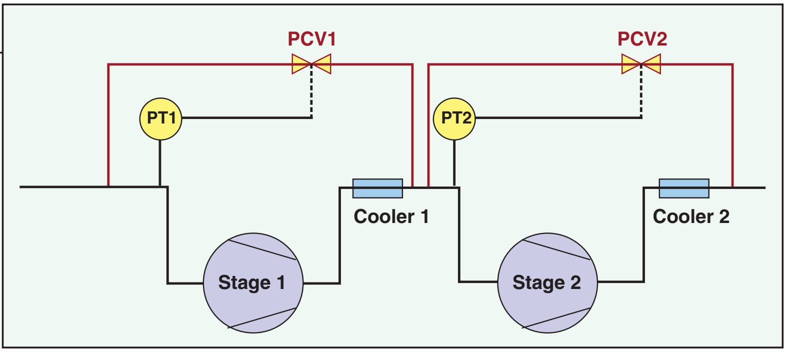

FIGURE 1. Shown here is a typical setup for capacity control in a two-stage reciprocating compressor using bypass valves (PCV = pressure control valve; PT = pressure transmitter)

TABLE 2. CONDITIONS FOR UP- AND DOWN-STEPPING PARALLEL COMPRESSORS				
Up-stepping:				
Compressor load B (Slave)	**Compressor load A (Master)**			
	25%	50%	75%	100%
25%	Upstep A	Upstep B	Upstep B	Upstep B
50%	Upstep A	Upstep A	Upstep B	Upstep B
75%	Upstep A	Upstep A	Upstep A	Upstep B
100%	Upstep A	Upstep A	Upstep A	No Action
Down-stepping:				
Compress or load B (Slave)	**Compressor load A (Master)**			
	25%	50%	75%	100%
25%	No Action	Downstep A	Downstep A	Downstep A
50%	Downstep B	Downstep B	Downstep A	Downstep A
75%	Downstep B	Downstep B	Downstep B	Downstep A
100%	Downstep B	Downstep B	Downstep B	Downstep A

choice of the step depends on the set-points of the bypass valves.

Up-stepping is one-step increasing in capacity, for example, 25 to 50% and down-stepping is one step decreasing, for instance, 75 to 50%.

The suction pressure controller reaches a balance in opening or closing the bypass valve depending on actually adjusted step load (0–25–50–75–100%) and actual flow through the compressor. If the actual bypass flow is smaller than 30% and higher than 5%, no up- or down-stepping action is required in the step capacity control. This means that suction pressure is only controlled by the stage bypass valve in this case.

If bypass flow is lower than 1% or for a time duration (approximately 3 min) lower than 5%, bypass flow is very small, and therefore, the suction pressure controller is not able to maintain suction pressure with fully closed bypass. The step capacity control has to make the up-step, increasing the compressor performance over 25% by adjusting the next higher step.

Increasing compressor performance will cause a momentary suction pressure drop. Hence, time delay (1 min) is considered to prevent activation of low or low low-suction-pressure alarms or trips. After a certain time, suction pressure will reach stability again and restore the suction pressure to its normal value.

If bypass flow is higher than 35% or for a time duration (approximately 3 min) higher than 30%, bypass flow is very high and the compressor is wasting energy. To avoid unnecessary bypass flow, the step capacity control should perform down-stepping, decreasing the compressor performance by 25%. Decreasing compressor performance will cause a pressure peak in suction. Consequently, a time delay (approximately 1 min) should be considered to prevent activation of high or high high-suction-pressure alarms or trips. After passing a certain time duration, the suction pressure will reach stability again and restore the suction pressure to its normal value.

The lowest automatic down-step will be to 25%. The 0% performance step is only used in compressor startup and shutdown sequences.

Compressor parallel working. Generally, 10% increasing flowrate is performed by increasing suction pressure, but in many applications, two compressors are needed to send out enough flowrate downstream of the system. For this purpose, one of the compressor controllers will be master and the other the slave. The master takes the control decisions about suction pressure control and up- and down-stepping of the capacity. A slave compressor always follows the master decisions.

To start two compressors, when the master machine is stable at 100% capacity, the slave machine should be started in 0% load. Then, during several steps, performance of both are equaled as much as possible. In this regard, in each stage, 25% of master compressor capacity (one step) will be decreased and 25% capacity of slave machine will be increased subsequently. Meanwhile, enough time delay (approximately 1 min) between each step should be considered so that flow stability is obtained at suction of compressors.

The conditions for up- and down-step criteria are the same as for single compressor operation and are shown in Table 2.

If the operator decides to take out one of the two running compressors, the selected compressor will be stopped with the following automatic steps with a time delay (approximately 1 min):

The load of the selected compressor will be decreased by 25%. At the same moment, the load of the compressor that is intended to remain in operation will be increased by 25%. This method will be repeated until the selected compressor is completely unloaded (at 0% capacity).

Stopping the compressor

Generally, three compressor-stop methods are considered regarding the permitted overhaul time of compression units and the safety level of the machine for plant protection.

Normal compressor stop. A normal shutdown is manually initiated by the operator from the DCS or LCS, if the compressor needs to be stopped for reasons such as overhauling the machine or unit maintenance. The following actions will be taken automatically afterwards:

- All bypass valves will be opened by a slow ramp up to fully open position. Note that ramp shall be set slowly enough to avoid over pressurization of lower stages downstream of the bypass valves
- The hydro jacking pump of the main motor and the compressor will be started
- The main motor will be stopped, and all control valves will be de-energized and return to their fail position, especially bypass valves, which will be closed
- The isolating suction valve will be forced closed, and the frame lube-oil pump and cylinder-oil pump will continue to run after post-lubrication time
- Oil tank heaters and the motor space heater will be enabled
- At last, the discharge isolating valve will be closed and the compressor will be manually depressurized by relief valves (vent valves) installed in each stage

Automatic stop based on trip. The automatic shutdown of the compressor is used to avoid damage of equipment and to ensure personnel safety. This stop sequence is exactly the same as a normal stop, except that the bypass valves are not opened at the first step.

Emergency stop. In case of danger, manual actuation of the emergency push buttons, located around the machine or on the emergency shut down (ESD) panel, shall switch off all the electrical consumers (main motor, oil pumps, heaters, solenoid valves on bypass valves and so on). This stop sequence is similar to automatic trip stop except that no post lubrication is needed by frame and cylinder oil pumps.

Final remarks

Most reciprocating compressors are specified for constant speed operation to avoid excitation of torsional critical speeds. For all constant speed applications, it is recommended that an automatic bypass control be provided. For more flexibility of the system, an unloader valve or pocket may be furnished to decrease power loss during turndown capacity. Moreover, if the stepless method is employed, it should be supplemented with a bypass control arrangement. ■

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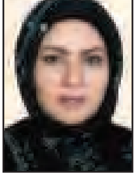

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Vacuum Systems: Recommendations For Safe Operation

Follow this guidance to ensure that steam ejector systems, mechanical vacuum pumps and integrated vacuum systems are designed, operated and maintained to ensure process safety

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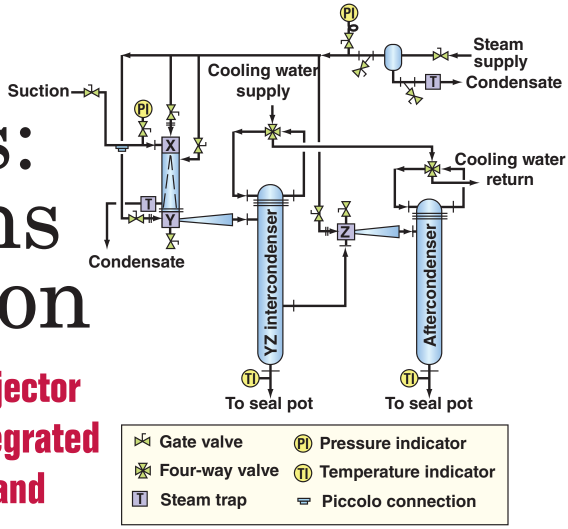


FIGURE 1. Shown here is a schematic of a three-stage steam ejector system with inter-condensers (Reprinted with permission from Ref. 1)

This article is the first in a two-part series, in which the types of process vacuum equipment (steam ejector systems and mechanical vacuum pumps) are discussed.[1] Emphasis is given on how they are constructed and operate, and how to design associated parts of the systems (such as foundations, suction piping, discharge piping, and so on), to ensure safe operation.

Many unit operations — including distillation, evaporation, drying, crystallization, filtration, and others — are often conducted under vacuum conditions. To achieve the desired vacuum, the following three process- vacuum systems can be used:

- Steam ejector systems
- Mechanical vacuum pumps
- Integrated vacuum systems (which combine steam ejectors and mechanical vacuum pumps)

Available capacities and operating ranges for vacuum pumps and vacuum pumping systems that are most often used in process operations throughout the chemical process industries (CPI) are listed in the Table on p. 60 [*1*].

This article reviews the hazards associated with steam ejector systems and mechanical vacuum pumps, and recommends design and operating practices that can be taken to prevent or mitigate these dangers.

1. Editor's note: This is the first half of a two-part article. Part 2 is scheduled to appear in the October 2012 issue.

Steam ejector systems

Steam ejector systems are generally categorized into one of four basic types: single-stage, multi-stage non-condensing, multi-stage condensing, and multi-stage with both condensing and non-condensing stages.

For many years, steam ejector systems dominated process applications thanks to their simplicity, low purchase cost, good reliability and their ability to be designed for very large capacities (in excess of 1,000,000 acfm) and be operated at very low pressures in the micron range (for instance, six-stage units are routinely designed for suction pressures in the range of 3–10 microns). They also are available in a variety of materials of construction.

Steam ejector systems are ideal for wet-vacuum and highly corrosive applications. As a result, they are still being used instead of mechanical vacuum pumps in applications that require a reliable vacuum system that can tolerate corrosive chemicals, liquid slugs and solids carryover.

Safety considerations. In general, there are very few safety problems associated with the operation of steam ejector vacuum systems. However, one safety issue associated with their design is the potential for back-streaming of steam into the process equipment.

For instance, steam ejectors normally use a "steam bleed" to control the suction pressure. If the process contains a water-reactive chemical (such as acetic anhydride), a steam bleed is not acceptable. If the ejector should "backfire," this will push steam back into the process and could result in a possibly violent reaction, which could overpressure the system.

To avoid this problem, a nitrogen bleed can be used to control the suction pressure. If the jet should back fire, nitrogen will prevent the steam from reaching the process. However, using nitrogen to control an existing multi-stage ejector may not be technically feasible. If nitrogen is to be used to control a multi-stage jet, the user must indicate this when writing the performance and equipment specifications for the steam ejector.

Using nitrogen to control the suction pressure of the first stage — that which is closest to the process — will affect the design of subsequent stages, and using nitrogen to control suction pressure can significantly affect the design of the inter-condensers. If the back flow of steam or water into the process equipment from the ejector system can cause a serious safety or quality problem, then a liquid separator (a so-called "knockout pot") may be installed between the ejector system and the process equipment.

Another possible safety problem may result from the loss of the inter-condenser cooling medium (usually water), which could lead to overpressure of the system.

In instances where suction gases, or materials entrained in the suction gases, might freeze inside the ejector, ejectors with external heating jackets can be specified.

Instrumentation. It is good engineering practice to provide adequate instrumentation to monitor and control the operation of the steam ejector system. The following instrumentation items are recommended for monitoring and controlling the vacuum system, and to support troubleshooting efforts:

- Control system for controlling the steam bleed to maintain the desired vacuum level
- Pressure gage on the main steam-supply line
- Steam pressure gage on each ejector
- Suction pressure gage on the inlet to each ejector
- Vacuum gage to monitor absolute pressure at the process vessel
- A flowmeter to monitor the water flow to each inter-condenser
- Temperature gages on the inlet and outlet water lines to and from each shell-and-tube inter-condenser
- Temperature indicator in the barometric leg from each barometric condenser

TABLE 1. CAPACITY AND OPERATING RANGE FOR VACUUM PUMPS AND VACUUM PUMPING SYSTEMS COMMONLY USED IN PROCESS APPLICATIONS

Type	Blind or base pressure	Lower limit for process applications	Single unit capacity range, ft^3/min
Single ejectors			
One stage	50 torr	75 torr	10–1,000,000
Two stages	4 torr	10 torr	
Three stages	800 micron*	1.5 torr	
Four stages	100 micron	250 micron	
Five stages	10 micron	50 micron	
Six stages	1 micron	3 micron	
Liquid-ring pumps			
60°F water-sealed:			
One stage	50 torr	75 torr	3–18,000
Two stages	20 torr	40 torr	
Oil-sealed	4 torr	10 torr	
Air ejector first stage	2 torr	10 torr	
Rotary-piston pumps			
One stage	5 micron	100 micron	3–800
Two stages	0.001 micron	10 micron	
Rotary-vane pumps			
Operated as a dry compressor	20 torr	50 torr	20–6,000
Oil-sealed, rough-vacuum pump	0.5 torr	20 torr	50–800
Oil-sealed, high-vacuum pump:			
One stage	5 micron	100 micron	3–150
Two stages	0.001 micron	10 micron	
Rotary-lobe blowers:			
One stage	100 torr**	300 torr	30–30,000
Two stages	10 torr**	60 torr	
Integrated pumping systems			
Ejector-liquid ring pump	1 micron	3 micron	100–100,000
Rotary-blower, liquid-ring pump	1.0 torr	5 torr	100–10,000
Rotary-blower, rotary-piston pump	0.1 micron	0.10 torr***	100–30,000
Rotary-blower, rotary-vane pump	20 micron	200 micron****	100–30,000

* 1.0 micron = 0.001 torr
** Based on intercooled design that uses gas admitted to a trapped discharge pocket to cool the blower
*** Based on using a two-stage, rotary-piston pump as the backing pump
**** Based on a two-stage, rough-vacuum, rotary-vane design that exhibits a base pressure of approximately 0.5 torr

Source: Ref. 1

Mechanical vacuum pumps

Design options. Mechanical vacuum pumps usually are classified as either wet pumps or dry pumps. The following lists show which mechanical vacuum pumps are wet and which are dry [2]:

Wet pumps:

- Oil-sealed rotary vane (single- and dual-stage)
- Oil-sealed rotary piston
- Liquid ring

Dry pumps:

- Rotary claw
- Rotary lobe (Roots)
- Rotary scroll
- Rotary screw
- Rotary vane
- Rotary piston
- Diaphragm

Oil-sealed vacuum pumps. Both rotary-vane and rotary-piston vacuum pumps rely on oil for the following functions:

- To seal the internal clearances between rotary components and housing to reduce gas slippage
- To transfer the heat of gas compression away from wear surfaces
- To lubricate the rotary internals
- To flush away moisture
- To inhibit corrosion of internal parts

During operation, an oil-sealed pump compresses the gas to a pressure that is slightly higher than atmospheric to expel it through the exhaust valve. At this pressure, and at an operating temperature of 70–100°C, active gases readily attack pump bearings, seal materials and the oil. The

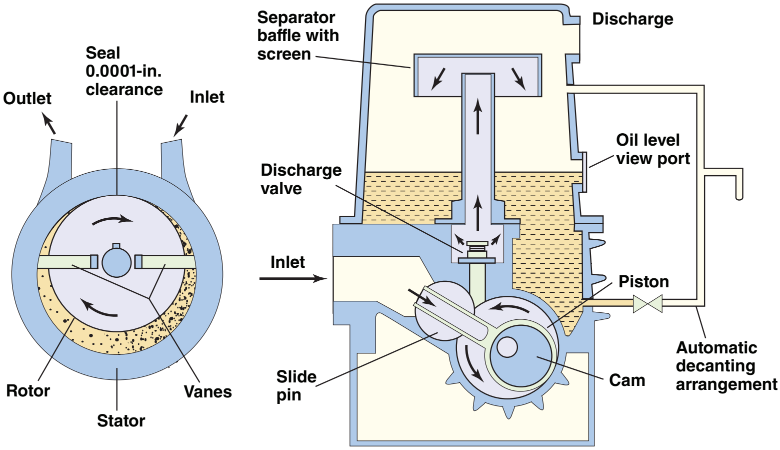


FIGURE 2. **Shown here is a schematic of the internals of a rotary-vane vacuum pump (left) and a rotary-piston vacuum pump (right) (Reprinted with permission from Ref. 1)**

selection of an appropriate oil for the application is critical for good pump maintenance.

This dependency on the oil for internal lubrication is a limiting factor in the use of these types of pumps in the CPI. The integrity of the oil must be maintained to avoid internal damage that could cause contaminant buildup. Particulates in the oil must be filtered out ahead of the vacuum pump, and water or solvent vapors must either be removed ahead of the pump by using pre-condensers, or prevented from condensing within the pump by the use of gas ballasting (air stripping) or oil distillation.

For oil-sealed, rotary-vane vacuum pumps, a variety of oils are available, each with very different properties. The sealing oil used in such pumps that are destined for chemical service has several additional requirements. For instance, it must:

- Be inert to chemical attack by acids, bases, halogens and halogen-induced aging)
- Be resistant to strong oxidants such as oxygen, chlorine and fluorine
- Possess good solvent power for problematic, mostly organic constituents of pumped media, such as oligomers, polymers, resins, crystalline decomposition or reaction products

Both oil-sealed, rotary-vane and piston vacuum pumps need periodic oil changes. The chief purpose of changing of the oil is to remove particles that either originate from aging of the oil or particles that are generated by the process (such as dust, decomposition products, and other contaminants).

When either an oil-sealed rotary-vane or rotary-piston vacuum pump is improperly installed a hazardous condition can result. The following installation practices will minimize potential problems with these types of oil-sealed vacuum pumps [*1*]:

Foundations. The foundation design will depend on the type and size of the oil-sealed pump. If the pump is inherently balanced, it can be mounted on any floor that is able to support the weight of the pump. However, pumps that are partially balanced, should be installed with vibration mounts or matting between the base and the floor. If the pump is not balanced, the base should be securely fastened to a concrete foundation with anchor bolts and grouting. The pump manufacturer should be consulted when designing the pump foundation. It is always good engineering practice to install flexible, bellows-type connectors in the suction and discharge piping; these can help to isolate the rest of the system from pump vibration.

Suction piping. Certain precautions must be taken when installing the suction piping for an oil-sealed vacuum pump. Care must be taken to prevent any backstreaming of pump oil into the process vessel and to prevent process liquids and solids from reaching the pump. Backstreaming of oil is especially significant with larger pumps. Baffles and traps are frequently installed in the suction line to prevent this from happening.

Cold traps, knockout pots, wet scrubbers, and bag filters are some of the devices used to protect the pump from liquids and particulates in the suction stream. The degree of protection necessary will depend on the process application. If the upstream knockout pot is not continuously drained, a high-level sensor should be installed to automatically stop the pump or close an isolation valve to prevent any overflow from the knockout pot from reaching the pump.

Oil-sealed vacuum pumps must be protected from flooding if the oil flow does not automatically stop when the pump is shut down. To prevent flooding, a valve can be installed in the suction line to admit air or nitrogen to break the vacuum when the pump stops. This prevents the siphoning of oil from the reservoir when the oil is not being discharged from the pump by the normal pumping action. A zero-speed switch or solenoid valve is used to actuate the air or nitrogen bleed and close the isolation valve. Closing the isolation valve allows the process to remain under vacuum and allows the vacuum in the pump to be broken more quickly.

Flooding may be avoided with some oil-sealed pumps by sealing the discharge line using a barometric leg or a discharge check valve. This allows the system to pull a vacuum on the oil reservoir so that oil siphoning stops. A check valve or knockout pot must be installed in the suction line to prevent reverse flow from carrying oil into the process vessel. A knockout pot is recommended for critical applications where carryover of oil into the process vessel might result in a fire or explosion.

Discharge piping. Condensable vapors that pass through an oil-sealed pump tend to condense in the discharge line. The discharge piping should slope away from the pump, and a drip-leg or knockout pot should be installed in the discharge line to prevent the backflow of condensate into the pump. Backpressure on the pump should be minimized.

Serious mechanical damage can result if the pump is operated against a closed discharge valve. Discharge valves are therefore not recommended unless the pump must discharge into a common exhaust header. If a discharge valve is required, an interlock (that ensures that the exhaust valve is open before the pump can be started) is recommended; in fact, this is a condition of the warranty for some vacuum-pump manufacturers.

A fine mist of oil drops is emitted with the exhaust gases that are discharged from an oil-sealed pump. Exhaust filters are normally installed to eliminate this oil mist. The most popular design has removable filter elements and can be mounted anywhere in the discharge line. Units are available that can remove up to 99.9% of the exhausted oil mist. Oil mist filters should not be allowed to become blocked, as hazardous over-pressures could occur. A differential pressure sensor and alarm can be installed on oil mist filters to monitor their operation and detect potential blockages.

If the oil mist is contaminated with organic compounds from the process gas, especially if they are toxic, then an activated carbon filter should be installed after the oil mist filter. The activated carbon filter will remove these contaminants by adsorption.

Pump cooling. Oil-sealed pumps must be cooled to remove the absorbed heat of compression. Both water- and air-cooled models are available. Air-cooled models should be installed in an area that will allow an ample flow of fresh air to reach the pump. The jackets on water-cooled pumps usually contain small clearances that can plug with solids. A strainer or filter should be installed in the cooling water supply line to remove any solids. A regulating valve should be installed in the cooling-water supply line to control the pump operating temperature. A high-temperature "cutout" switch is sometimes installed to protect the vacuum pump from overheating.

Movement of oil and flammable vapor droplets from oil-sealed vacuum pumps has been known to generate a static charge, which could lead to a fire or explosion [*3*]. To avoid or minimize this potential hazard, it is critical to inert the exhaust stream with nitrogen to eliminate oxygen or reduce it to below the limiting oxygen concentration (LOC).

Also, in many cases, it is prudent to use nitrogen to dilute the gas stream upstream of the pump inlet to move it out of the flammable range. It is critical that the nitrogen supply source be free of oxygen. There have been a few instances where the nitrogen supply was compromised, allowing in air that can lead to an explosion [*3*]. For pyrophoric gases, such as silane, special care must be taken to prevent air or oxygen from entering the system.

Liquid-ring vacuum pumps

Liquid-ring vacuum pumps are widely used in the CPI. They are available as single-stage (one or two impellers in parallel) or two-stage (two impellers in series) units. This type of pump does not require internal lubrication of the impellers, since they do not contact the housing. The sealant liquid, used for both sealing and cooling, can be virtually any liquid that is compatible with the process gas. For instance, water, low-viscosity oils, glycols and many process solvents such as toluene, xylene, methanol, ethanol, propanol, butanol and ethylbenzene have been used as sealants in liquid-ring pumps. Other liquids have also been used, such as ethylene glycol and propylene glycol, and even sulfuric acid.

The motor horsepower must be corrected for the sealant viscosity and density (see Reference 1, p. 273). These sealants can be recirculated in a full recovery system that includes a gas-liquid separator tank and a heat exchanger to cool the sealant liquid. Even higher vapor-pressure sealants can be used if a low-temperature coolant (such as glycol solution) is used in the heat exchanger to reduce the sealant liquid temperature. This recovery system allows process materials to be collected in the pump and either returned to the process or collected for disposal, thereby minimizing contamination of other liquids or the environment.

The most advantageous property of liquid-ring vacuum pumps is the fact that they are nearly isothermal. The liquid in the pump provides an ample heat sink for the removal of the heat of compression. This means that the temperature rise in the casing is very small. It is a function of the characteristics of the sealant fluid used, its temperature, flowrate, and of course, the process gas conditions. The resulting low internal temperatures provide improved safety, especially when compressing explosive gases and fluids.

Generally, internal temperatures in liquid-ring vacuum pumps are the lowest of most mechanical vacuum pumps. Hence, gases exit at almost the same temperature, or in some cases, at a lower temperature than they enter the vacuum pump. Further, since the gases handled are in direct contact with large amounts of sealant liquid, even if touching of internal parts (the rotor with the casing) occurs, the resulting sparks are not hot enough or of sufficient duration to cause explosions or ignition of even the most hazardous gases. This provides a significant safety advantage, especially when handling easily ignitable gas streams. The occurrence of an explosion within liquid-ring vacuum pumps handling flammable vapors is very unlikely as the sealant liquid in the pump casing acts as a hydraulic flame arrester, quenching any flame that may be generated and preventing it from propagating.

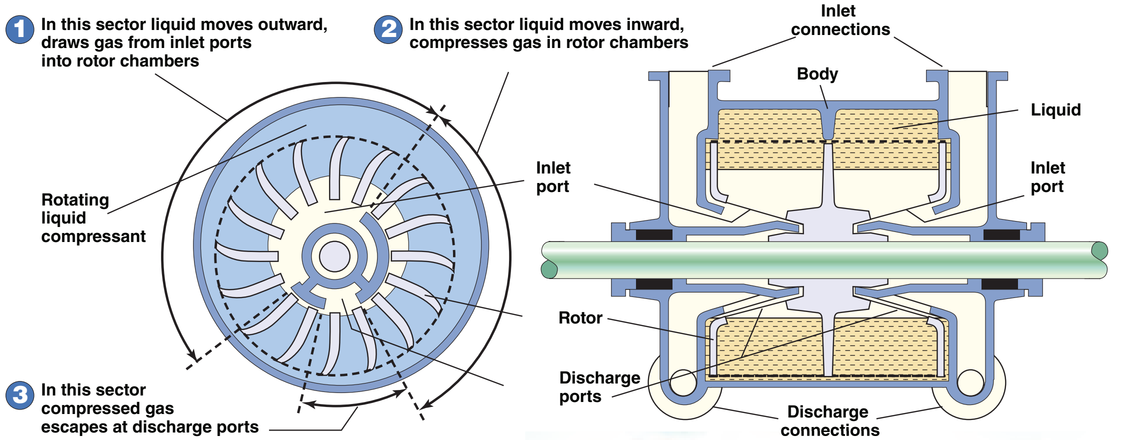
However, one of the major drawbacks of liquid-ring vacuum pumps is that they must cope with cavitation when running at low inlet pressure. If cavitation is allowed to continue over long periods of time, serious damage can be done to the pump. The amount of cavitation can be affected by the sealant liquid, sealant temperature, impeller speed, blade angle and inlet pressure. For a given pump and sealant liquid, cavitation can normally be suppressed by bleeding air (or nitrogen) into the pump inlet to raise its total pressure above the vapor pressure of the sealant at operating temperatures.

All liquid-ring pump units have auxiliary equipment such as a vapor-liquid separator, a sealant recirculation pump, a sealant cooler, and various instrumentation items, relief devices and valving.

Some advantages and disadvantages of liquid-ring pumps are summarized below:

Advantages:

- Has a simple design that uses only one rotating assembly
- Can be fabricated from any castable material
- Creates minimal noise and vibration



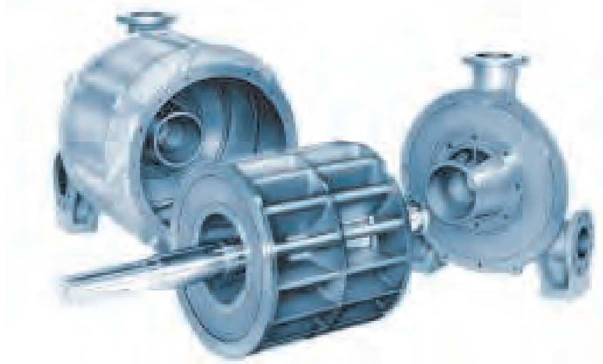


FIGURE 3. This figure shows a cross-sectional view (above) and general view of assembly (right) of a liquid-ring vacuum pump (Reprinted with permission from Ref. 1)

- Experiences little increase in the temperature of the discharged gas
- Experiences no damage from liquid or small particles in the process gas
- Enables simple maintenance and rebuilding
- Uses a relatively slow rotational speed (1,800 rpm or less), which maximizes operational life
- Can use any type of liquid for the sealant fluid in situations where co-mingling of the sealant with the process vapor is permissible
- Has no lubricating liquid in the process equipment to be contaminated
- Can accommodate both condensable and non-condensable gases while operating as both a vacuum pump and condenser

Disadvantages:

- Mixing of the condensed gas/vapor with the sealing liquid, which requires external equipment to separate them
- The risk of cavitation requires a portion of the process load to be noncondensable under operating conditions
- High power consumption requires large motors to form and maintain the liquid ring
- Achievable vacuum is limited by the vapor pressure of the sealant liquid at the operating temperature
- The power consumption of a liquid-ring pump is normally higher than that of other types of mechanical vacuum pumps

The proper installation of a liquid-ring vacuum pump is critical to its operation and maintenance. The following recommendations should be followed to ensure proper installation [*1, 4*]:

Foundations and pump checking. Since liquid-ring pumps do not usually create vibration problems, special foundations are not required. Pumps that are about 50 hp or above are best placed on a concrete pad. Smaller units may be mounted on existing floors and skids. On pumps of all sizes, the base should be leveled and the coupling alignment should be checked during installation. On belt-driven models, the sheaves should be aligned and the proper v-belt tensions set according to the manufacturer's instructions. After the motor has been wired, the pump rotation must be checked to ensure that the polarity is not reversed. Do not test the polarity without liquid in the pump.

Suction piping. A check valve or a valve that closes automatically when the motor is shut down should be installed in the suction line to prevent dumping of the liquid from the pump back into the upstream process vessel in the event of pump failure. This check valve will also serve to prevent any backflow of air into the upstream process vessel.

Where protection against backstreaming air is critical due to process requirements, the vent and drain on the separator should be barometrically sealed. This can be accomplished by discharging to a seal pot or by installing a vertical run in the suction line to serve as a barometric leg between the pump and the process vessel. If the suction line is used as a barometric leg, the vacuum pump discharge must be sealed beneath the liquid level in the separator.

A vacuum relief device should be installed in the suction line on pumps that are not automatically controlled. Otherwise, the pump may cavitate during operation at low suction pressures or when it is dead-headed by mistake. The vacuum relief valve should be connected to a supply of inert gas (usually nitrogen) if admitting air to the system is unacceptable.

If inlet spray nozzles are used to condense incoming vapors, the nozzles should be located in the suction piping as close as possible to the pump. An enlarged pipe section or a small vessel may be needed to provide contacting space between the spray liquid and incoming vapors.

Discharge piping. The pump and discharge piping must be installed to minimize backpressure on the pump.

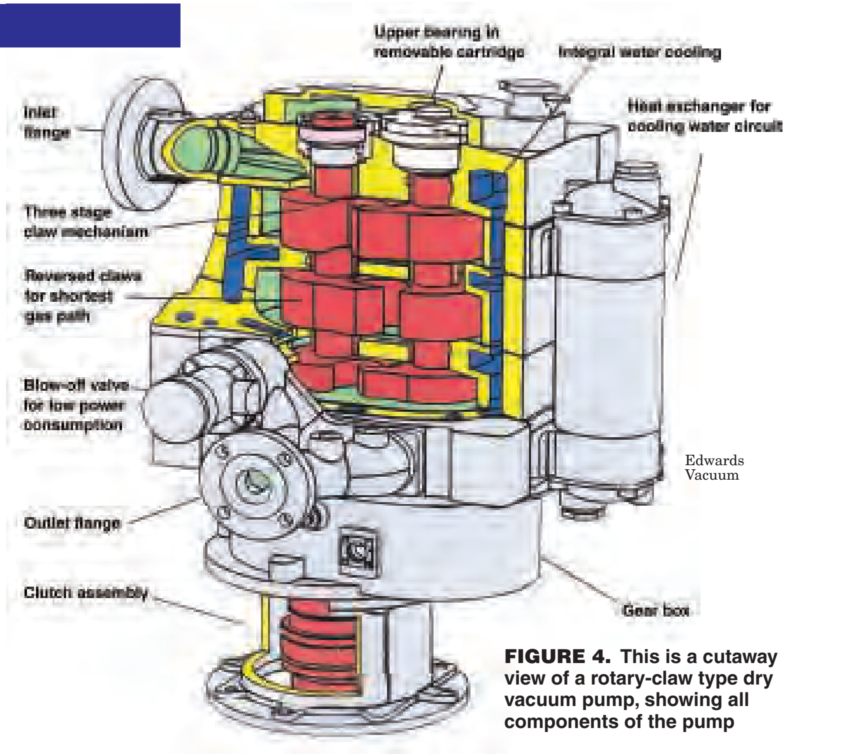


FIGURE 4. This is a cutaway view of a rotary-claw type dry vacuum pump, showing all components of the pump

Both the compressed gases and the sealing liquid are discharged through the pump discharge connection, and the discharge piping must be sized for two-phase flow. Piping of the same diameter as the discharge connection will usually be adequate, but a larger line may be required if the distance to the vapor-liquid separator is more than a few feet. The discharge piping should never rise more than 2–3 ft above the pump before the gas and liquid are separated. The discharge piping should be designed so that it is free to drain after a shutdown.

Pump draining. The pump casing should be drained to the shaft level before a liquid-ring pump is started. Starting the pump when it is completely filled with liquid will overload the motor and place undue mechanical stresses on the shaft and rotor. Shaft failure is not uncommon on large liquid-ring pumps operated in this way.

A solenoid valve installed in the sealant supply line to the pump is the best way to stop the flow of sealant when the pump is shut down. It will not be necessary to drain any residual liquid from the casing. If sealant is recirculated from the vapor-liquid separator, the sealant will automatically drain to the shaft level if the separator is designed with an overflow nozzle at the pump shaft level. The sealant may be manually drained to the shaft level through a hole drilled in the pump casing or by a drain in the sealant supply line.

If the pump is located outdoors or in unheated buildings, it should be completely drained for freeze protection during prolonged shutdowns. Similarly, the pump must never be restarted dry; it must be partially filled with sealant liquid. This may necessitate a bypass around the solenoid valve in the sealant supply line.

Sealant piping. There are three possible arrangements for sealant flow to liquid ring pumps: Once-through (no recovery), closed loop (recirculated sealant) and partial recirculation system (partial recovery).

Once-through arrangement. A once-through installation does not necessarily imply that none of the sealant is recovered; rather, the term simply refers to the absence of any return piping from the vapor-liquid separator drain to the pump-sealant supply connection. This is a common arrangement where conservation or contamination of the seal liquid is not a concern.

The line supplying sealant to the pump should contain a shut-off valve followed by a strainer with a blow-down valve to remove particulates. An automatic solenoid valve may be installed in the supply line, interlocked with the pump motor's operation, to prevent flow of sealant to the pump in case the motor stops running, (in this case, if the motor stops, the valve closes to prevent the pump casing from filling with liquid).

If a solenoid valve is used, a bypass should be installed to allow some sealant to be introduced before starting (when the pump has been completely drained). The sealant flow can be regulated manually or automatically.

The manual method regulates the flow by a globe valve (upstream of an orifice), which uses pressure drop across the orifice to set the desired pressure to the pump. The orifice is usually chosen so that the proper flow to the pump is set by maintaining a pressure drop of 5 to 10 psi across the orifice. Automatic control is achieved by using a flow controller to maintain a constant pressure drop if the sealant supply pressure tends to fluctuate.

Closed loop (total recirculated sealant) arrangement. In this arrangement the system is designed to operate as a closed loop with no addition of fresh sealant. Only the vapors that condense in the pump need to be removed from the system. In actual practice, however, some fresh sealant is often continually added to control the concentration of low boilers or other contaminants in the sealant or to make up for evaporative losses from the system.

A heat exchanger is installed in this system to remove the heat of compression and condensation from the sealant before it is reintroduced into the pump. The heat exchanger is normally water cooled, but any suitable coolant can be used. Often, the system is provided with a sealant recirculation pump. This allows more flexibility in the design and operation of the vacuum system. The sealant cooler can be designed for a higher pressure drop, which results in better heat transfer and reduced fouling. A recirculation pump must operate at suction pressures that are too high to "siphon" an adequate flow of sealant from the separator and in operations involving frequent or prolonged evacuation. A recirculation pump is often used for systems in which several vacuum pumps share a common separator.

The fresh sealant makeup line is installed in much the same way as for a once-through installation. In addition, a check valve should be installed in the makeup line to prevent reverse flow if makeup supply pressure is lost while the vacuum pump is operating.

The recirculation line should contain a strainer and one or more temperature gages. The strainer removes solids that would otherwise accumulate in the sealant. Temperature gages should be installed on both sides of sealant coolers to monitor heat exchanger performance.

Partial recirculation (partial recovery) arrangement. A partial recirculation system has no sealant cooler. Fresh solvent makeup is used as the heat sink. A portion of the total sealant flow is supplied as cold makeup and the rest is recirculated from the separator. The fresh and recirculated streams mix and come to some intermediate temperature before entering the vacuum pump. The desired temperature is controlled by varying the ratio between the two streams until a satisfactory temperature is reached. A partial recirculation arrangement can reduce fresh sealant requirements by 50% or more compared with once-through systems, depending on the temperature of the fresh stream.

The recommendations about piping design and filters given in the section on closed-loop arrangements apply here as well.

Potential hazards and options to prevent or mitigate them. Listed below are some potential hazards of liquid-ring vacuum pumps and measures that can be used to prevent or mitigate them [*5*]:

- The escape of flammable or toxic vapors from failed mechanical seals can create external hazardous conditions. Liquid-ring vacuum pumps handling non-hazardous vapors usually are provided with single mechanical seals. If these pumps handle flammable or toxic vapors, then double mechanical seals should be installed with seal-welded connections to eliminate leak paths
- Improper installation, mounting, and alignment of pumps: This can cause damaging vibration, which can lead to potential seal or other mechanical failures and leakage to the environment of internal fluids (sealant, gas or both)
- Improper installation of piping can result in external loads on the casing leading to pump failure, or again, leakage of internal fluids to the atmosphere
- Liquid-ring pumps use seal liquids that are in direct contact with the process gases, and if the gases pumped are flammable or toxic, the discharged gases and sealants will potentially also be hazardous
- Since seal liquids are in direct contact with the process gases, liquid ring pumps act as mixers in a sense, and any potential reactions between the seal liquid and the process gas must be considered
- Gases and seal liquids are discharged together and proper separation must be done if the gases are to go to subsequent processes. If gases or vapors condense or dissolve in the seal liquid, consideration must be given to selecting the proper materials of construction for the discharge system to resist attack by any of the gases present in the process gas stream to the pump, or introduced and/or formed in the seal liquid
- Seal liquids, if recirculated, will become saturated to equilibrium states with the process gases being pumped. As a result, the overflows from a liquid-ring pump system must be treated as a hazardous liquid if the gases are flammable or toxic. Since, in many cases the seal liquid is reused in the system, the concentrations of contaminants can increase with time
- Since compression in a liquid-ring vacuum pump is essentially isothermal, this means that any vapors or gases exiting will be saturated with the seal liquid and that any potentially hazardous vapors or gases may condense and be discharged as liquids to the discharge system
- Proper materials of construction must be selected if liquid-ring pumps use seal liquids that are corrosive (such as sulfuric acid) or if the process gases contain corrosive components
- Liquid-ring vacuum pumps are not generally built for high pressures and use non-live sealed gasket systems for the most part. This means that if pressure can rise in hazardous systems quickly, the liquid ring system is often the lowest pressure component, and must be protected from the rise or particular design variations made, to ensure that they are resistant to maximum pressures that may occur

Dry vacuum pumps

Dry vacuum pumps have been used since the 1980s, originally in the semiconductor industry, then in the pharmaceutical industry, and now in other chemical process applications. They have several advantages, as follows:

- No contamination of the process caused by backstreaming of sealing liquids or lubricating oils
- No contamination of evacuated gas with sealing liquid or lubricant
- Due to lack of condensation (which is assured because the pump runs hot), the pump can be fabricated of standard, inexpensive cast iron
- They have a rugged rotor design which is constructed of sturdy cast iron, or ductile iron, without any flimsy rotating components
- Noncontact design: Timing gears are oil lubricated in a sealed-off end chamber to synchronize the rotors for proper phasing and noncontacting operation
- High operational speed: Operation at high speeds reduces the ratio of gas slip to displacement, increases net pumping capacity and reduces ultimate pressure. To accomplish this, the rotors are well balanced
- They can be designed with multiple staging
- The pump can discharge to the atmosphere
- They can be used for handling corrosive gases
- They can be used for handling flammable gases as the pumps are designed for containment of an internal explosion for this service

However, they also have the following drawbacks:

- They cannot handle particulates, nor slugs of liquid
- They may require a silencer

- They may discharge gases at high temperatures, in some cases as high as 350–400°C. Newer designs allow running at lower temperatures and have precise temperature control
- Some models are difficult to repair or rebuild
- They may require a gas purge for cooling or to protect the bearings and seals from the process gas
- Due to the high operating temperature, some process materials such as monomers, may have a tendency to polymerize

An excellent overview of dry vacuum pumps is presented by Ryans and Bays [*6*]. Two types of vacuum pumps that are used in high and ultra-high vacuum applications are the turbo-molecular pump and diffusion pump. However, they are not discussed in this article. ■

Edited by Suzanne Shelley

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Fractionation Column

Try harder to stay awake

Mike Resetarits is the technical director at Fractionation Research, Inc. (FRI; Stillwater, Okla.; www.fri.org), a distillation research consortium. Each month, Mike shares his first-hand experience with *CE* readers

Nobody would have ever called Denny proactive. He wasn't even reactive. He was barely active in any sense of the word. A majority of his acquaintances would have, in fact, called him inactive. Nevertheless, Denny was on our sales team, albeit the junior member.

Dan was the division vice president. If Dan ever smiled, I don't remember it. He wasn't very inspiring. In fact, when he spoke, he usually depressed us, at least until bowling night.

Denny, Dan and I went to an oil city to talk to an angry oil company about a highest-cost bid we had submitted. It was my job to convince the oil company engineers that ours was truly the only viable option. It was not my job to talk cost or price. Our salespeople were instructed to keep the technical experts clean — uncontaminated by cost/price negotiations.

I was truly convinced that ours was the only option. I gave a one-hour presentation attempting to prove it. Thereafter, the oil company engineers put me through a ringer. I kept my cool. I had most of the answers. I explained my answers to their apparent satisfaction. Then, they said, "Now let's talk about your ridiculous price." I turned to Denny, who had been sitting patiently in a corner. Dan and the oil company engineers also turned toward Denny.

Denny was fast asleep. In fact, he was past fast asleep and nearly comatose. He was stretched way out in the conference room chair. His arms were hanging almost to the floor. His mouth was wide open and two flies were perched on his lips wondering about today's lunch. In a raised voice, Dan said, "Hey, Denny!" Denny awoke with a start. He didn't seem as embarrassed as he should have been. Being embarrassed would have been appropriately reactive, but reactive was not Denny's way, as I said previously.

Dan asked Denny to review our costing strategy. Denny's response was half-hearted and lacking in details and understanding. It was lunchtime. The oil company engineers let us off the hook. We were led unceremoniously to the main lobby and a taxi.

In the taxi, Dan did all the talking. I was talked out; Denny was talked at. Dan fired Denny, recounting all of Denny's previous occasions of non-proactivity and non-reactivity. I looked out the taxi window hoping that the airport would arrive real soon. Luckily, in the airport, we caught three totally different flights.

All of us have had occasions where we have nodded off, or nearly nodded off, at meetings or presentations. There are several remedies. Drink more morning coffee. Stand up in the back of the room. Take rest room breaks. Participate more fully in the discussions. Take detailed notes. Stick to a weekend sleep schedule that matches your weekdays. Avoid caffeine during evenings. Fly to overseas meetings one day early. There are some days where sleep is more "fun" than anything else on your action list. Get more if you can. ■

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44 Mixing, Blending Equipment
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46 Piping, Tubing, Fittings
47 Pollution Control Equipment & Systems
48 Pumps
49 Safety Equipment & Services
50 Size Reduction & Agglomeration Equipment
51 Solids Handling Equipment
52 Tanks, Vessels, Reactors
53 Valves
54 Engineering Computers/Software/Peripherals
55 Water Treatment Chemicals & Equipment
56 Hazardous Waste Management Systems
57 Chemicals & Raw Materials
58 Materials of Construction
59 Compressors

1	16	31	46	61	76	91	106	121	136	151	166	181	196	211	226	241	256	271	286	301	316	331	346	361	376	391	406	421	436	451	466	481	496	511	526	541	556	571	586
2	17	32	47	62	77	92	107	122	137	152	167	182	197	212	227	242	257	272	287	302	317	332	347	362	377	392	407	422	437	452	467	482	497	512	527	542	557	572	587
3	18	33	48	63	78	93	108	123	138	153	168	183	198	213	228	243	258	273	288	303	318	333	348	363	378	393	408	423	438	453	468	483	498	513	528	543	558	573	588
4	19	34	49	64	79	94	109	124	139	154	169	184	199	214	229	244	259	274	289	304	319	334	349	364	379	394	409	424	439	454	469	484	499	514	529	544	559	574	589
5	20	35	50	65	80	95	110	125	140	155	170	185	200	215	230	245	260	275	290	305	320	335	350	365	380	395	410	425	440	455	470	485	500	515	530	545	560	575	590
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7	22	37	52	67	82	97	112	127	142	157	172	187	202	217	232	247	262	277	292	307	322	337	352	367	382	397	412	427	442	457	472	487	502	517	532	547	562	577	592
8	23	38	53	68	83	98	113	128	143	158	173	188	203	218	233	248	263	278	293	308	323	338	353	368	383	398	413	428	443	458	473	488	503	518	533	548	563	578	593
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15	30	45	60	75	90	105	120	135	150	165	180	195	210	225	240	255	270	285	300	315	330	345	360	375	390	405	420	435	450	465	480	495	510	525	540	555	570	585	600

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Classified Index September 2012

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People

SEPTEMBER WHO'S WHO

Nathan

Cooper

Duckworth

Shana'a

Matzen

John Nathan becomes director of operations for **Magnetic Products** (Highland, Mich.).

Intellitect Water (Hants, U.K.) makes two appointments: *Mark Duckworth* becomes engineering manager in product development; *Jo Cooper* becomes product and market specialist.

CleanTech Biofuels (St. Louis, Mo.) names *James E. Russell*, an independent consultant who spent 35 years with Science Applications International Corp. (SAIC), to its board of directors.

May Shana'a becomes group vice president, technology and growth strategy, for **Ashland Specialty Ingredients** (Wilmington, Del.).

Manfred Baumann, currently the managing director of **Gerresheimer Medical Plastic Systems** (Wackersdorf, Germany), assumes international leadership of the company's new sales and engineering center, technical competence center, making, finance & controlling, IT, human resources, marketing and legal. *Oliver Burgel* joins the company to become chief production officer.

OPX Biotechnologies (Boulder, Colo.) appoints *Earl Douglas* as vice president and general counsel.

Christian Matzen becomes head of sales for packaging systems for **Beumer Maschinenfabrik GmbH & Co. KG** (Beckum, Germany). ■

Suzanne Shelley

BUSINESS NEWS

PLANT WATCH

Air Products to build world's largest energy-from-waste facility in the U.K.

August 7, 2012 — Air Products (Lehigh Valley, Pa.; www.airproducts.com) plans to build and operate a renewable energy plant in the U.K. using advanced-gasification energy-from-waste (EfW) technology. The Tees Valley plant, located at the New Energy and Technology Business Park, near Billingham, Teesside, is said to be the first of its kind in the U.K., and the largest of its kind anywhere in the world with an approximate capacity of 50 MW. Air Products has secured the necessary environmental and planning approvals, and the renewable energy facility is scheduled to enter commercial operation in 2014.

Cobalt and Rhodia to build biobutanol demonstration plant in Brazil

August 1, 2012 — Cobalt Technologies (Cobalt; Mountain View, Calif.; www.cobalttech.com) and Rhodia, a member of the Solvay Group (Brussels, Belgium; www.solvay.com), will begin joint development and operation of a biobutanol demonstration facility in Brazil. This is a step toward the construction of commercial-scale biorefineries using Cobalt's technology to convert Brazilian bagasse and other local non-food feedstock into bio-*n*-butanol. Operational testing at the demonstration plant is expected to be completed by mid-2013.

BASF and Sinopec to explore building a world-scale isononanol plant in Maoming

July 31, 2012 — BASF SE (Ludwigshafen, Germany; www.basf.com) and China Petroleum & Chemical Corp. (Sinopec; www.sinopec.com) have signed a memorandum of understanding (MoU) to jointly explore the possibility of building an isononanol (INA) plant in Maoming Hi-tech Industrial Development Zone, Maoming, China. The final scope of the investment will be determined following the outcome of the joint feasibility study, which is expected by the end of 2012. Under the terms of the MoU, the two parties will evaluate the viability of jointly owning and operating a world-scale facility for the production of INA under a 50-50 joint venture (JV) agreement.

Novasep invests €3 million to boost its HPAPI manufacturing capabilities

July 19, 2012 — Novasep (Pompey, France; www.novasep.com) has announced a €3-million ($3.7 million) investment to expand its highly potent, active-pharmaceutical ingredients (HPAPI) manufacturing capabilities at its facility in Le Mans, France. In the new facility, Novasep will be able to perform cryogenic chemistry at –60°C in Hastelloy reactors, large-scale HPLC chromatography and drying in confined areas to manufacture antibody-drug conjugates (ADC) toxins at commercial scale. The plant expansion is expected to be fully operational by the beginning of 2013.

Uhde Inventa-Fischer to construct a polyamide 6 plant in China

July 18, 2012 — Uhde Inventa-Fischer (Berlin, Germany and Domat/Ems, Switzerland; www.uhde-inventa-fischer.com) and Changle Highsun Synthetic Fiber Technology Co. have signed a contract for the supply of a polymerization plant to produce high-performance polyamide 6 for textile applications. The multi-line plant will have an overall production capacity of 180,000 metric tons per year (m.t./yr) and will be located in Binhai Industrial Zone, Changle City, Fujian Province, China. Operation of the plant will be based on Uhde Inventa-Fischer's High-Performance Polyamide 6 technology (HPPA).

Teijin to expand dimethyl terephthalate supply capability

July 17, 2012 — Teijin Ltd. (Tokyo; www.teijin.co.jp) says it will expand its supply capability for dimethyl terephthalate (DMT), a petrochemical used as a raw material in producing polyethylene terephthalate (PET) and for other specialty chemical applications. The company's existing DMT production facility at the Matsuyama Factory in Ehime Prefecture, Japan, will add a new line that processes DMT into high-purity briquettes. Capital expenditure for the expansion will be approximately one billion yen. Construction will begin in October, with operations scheduled to begin in April 2013. The expansion will enable Teijin to double its annual supply capability for DMT from the current 50,000 tons to 100,000 tons.

Linde to construct two air-separation units for Tata Steel

July 16, 2012 — The Linde Group (Munich, Germany; www.linde.com) will be investing €80 million ($98.8 million) to construct two air-separation units (ASUs) at the Kalinganagar industrial complex in Odisha, India, for Tata Steel Ltd. The two companies recently signed an agreement to this effect. The air separation units are part of a long-term, on-site-gases supply contract for a new integrated steelworks that Tata is currently building in Kalinganagar. Each of the two new ASUs will be capable of producing 1,200 m.t./d of air gases. Once they go onstream in 2014, the units will supply gaseous oxygen, nitrogen and argon. Tata's new blast-furnace-based steel plant is also set to go onstream in 2014. It will initially produce 3 million m.t./yr. The company aims to increase capacity in the medium term to 12 million m.t./yr.

MERGERS AND ACQUISITIONS

Teijin to establish polyester chemical recycling JV in China

August 8, 2012 — Teijin Ltd. plans to establish Zhejiang Jiaren New Materials Co., Ltd., a JV with Jinggong Holding Group (Shaoxing, Zhejiang Province, China; www.jgsteelgroup.com). Through the JV, Teijin will chemically recycle polyester, as well as manufacture and sell the resulting fibers. The JV — to which Teijin will contribute 49% and Jinggong Holding Group will contribute 51% — will invest around ¥6 billion (about $76 million) in the construction of facilities for DMT production, polymerization and fiber spinning. The construction will begin this November, with operations scheduled to begin by the end of March, 2014.

AkzoNobel signs agreement to sell its shares in ICI Pakistan

July 30, 2012 — AkzoNobel (Amsterdam, the Netherlands; www.akzonobel.com) has reached an agreement to sell its 75.81% shareholding in ICI Pakistan Ltd. to the Yunus Brothers Group for $152.5 million (€124.4 million). The price is subject to adjustments. The transaction is expected to be completed toward the end of this year, once regulatory approvals have been obtained and the purchaser has completed a legally required tender offer for at least 50 % of the shares in ICI Pakistan held by the other shareholders.

Fluor forms joint venture with Brazil's Construcap

July 23, 2012 — Fluor Corp. (Irving, Tex.; www.fluor.com) has formed a new JV with CFPS Engenharia e Projetos, S.A. (Construcap; Sao Paulo, Brazil; www.construcap.com.br), to pursue engineering, procurement and construction management (EPCM) projects in Brazil. Financial terms of the JV were not disclosed. ■

Dorothy Lozowski

FOR ADDITIONAL NEWS AS IT DEVELOPS, PLEASE VISIT WWW.CHE.COM

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FOR MORE ECONOMIC INDICATORS, SEE NEXT PAGE

Economic Indicators

2010 — 2011 — 2012

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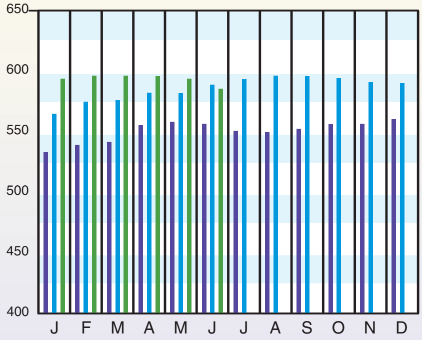
CHEMICAL ENGINEERING PLANT COST INDEX (CEPCI)

(1957-59 = 100)	June '12 Prelim.	May '12 Final	June '11 Final
***CE* Index**	585.5	593.8	588.9
Equipment	713.9	726.2	718.0
Heat exchangers & tanks	661.4	683.6	678.0
Process machinery	666.5	680.1	664.5
Pipe, valves & fittings	917.7	926.7	904.8
Process instruments	425.1	428.9	440.9
Pumps & compressors	927.0	928.1	904.7
Electrical equipment	513.7	515.2	510.8
Structural supports & misc	759.9	763.8	760.7
Construction labor	322.5	322.9	325.6
Buildings	527.2	527.7	519.1
Engineering & supervision	327.9	328.3	332.6

Annual Index:

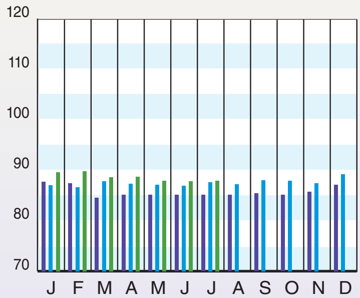
2004 = 444.2
2005 = 468.2
2006 = 499.6
2007 = 525.4
2008 = 575.4
2009 = 521.9
2010 = 550.8
2011 = 585.7



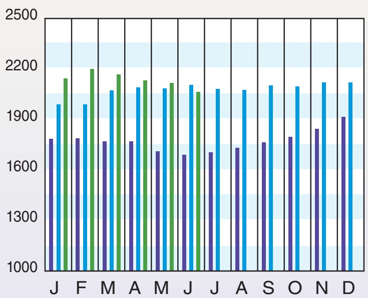
CURRENT BUSINESS INDICATORS

	LATEST			PREVIOUS						YEAR AGO		
CPI output index (2007 = 100)	Jul. '12	=	87.9	Jun. '12	=	87.8	May. '12	=	87.9	Jul. '11	=	87.6
CPI value of output, $ billions	Jun. '12	=	2,065.5	May. '12	=	2,116.1	Apr. '12	=	2,132.6	Jun. '11	=	2,104.6
CPI operating rate, %	Jul. '12	=	75.9	Jun. '12	=	75.8	May. '12	=	76.0	Jul. '11	=	75.6
Producer prices, industrial chemicals (1982 = 100)	Jul. '12	=	295.4	Jun. '12	=	312.7	May. '12	=	324.4	Jul. '11	=	338.6
Industrial Production in Manufacturing (2007=100)	Jul. '12	=	95.0	Jun. '12	=	94.5	May. '12	=	94.0	Jul. '11	=	90.4
Hourly earnings index, chemical & allied products (1992 = 100)	Jul. '12	=	158.7	Jun. '12	=	157.1	May. '12	=	157.6	Jul. '11	=	159.1
Productivity index, chemicals & allied products (1992 = 100)	Jul. '12	=	103.8	Jun. '12	=	104.6	May. '12	=	104.1	Jul. '11	=	108.0

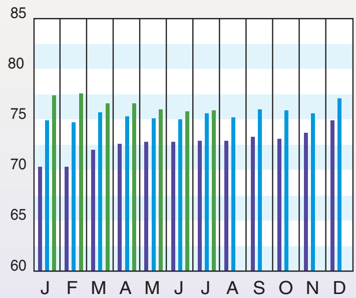
CPI OUTPUT INDEX (2007 = 100)



CPI OUTPUT VALUE ($ BILLIONS)



CPI OPERATING RATE (%)



Current Business Indicators provided by IHS Global Insight, Inc., Lexington, Mass.

CURRENT TRENDS

Capital equipment prices, as reflected in the *CE* Plant Cost Index (CEPCI; top), dropped 0.6% from May to June (the most recent data). Meanwhile, the Current Business Indicators from IHS Global Insight (middle), including the operating rate, changed very little from June to July.

Meanwhile, according to the American Chemistry Council's (ACC; Washington, D.C.; www.americanchemistry.com) latest weekly economic report at *CE* press time, the Organization for Economic Cooperation and Development (OECD) composite leading indicator (CLI) for June and other data "continue to point to an easing of economic activity" in most major OECD economies and slowdowns in most major non-OECD economies."

The June CLIs for China and India continued to point to a slowdown, and that for Russia suggests that it, too, may be headed toward a slowdown, the ACC report said. Brazil, on the other hand, appeared to be set for a moderate pickup in economic activity. The OECD CLI is designed to provide early signals of turning points between expansions and slowdowns in economic activity. The reading for the OECD countries plus Brazil, China, India, Indonesia, Russian Federation and South Africa showed a 0.1% decline in June, the ACC report said. It was the third consecutive monthly decline. ■