NASH
A Gardner Denver Product

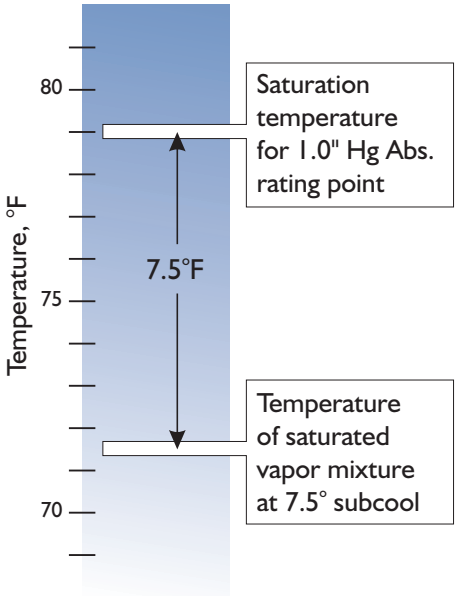
The NASH condenser exhauster saves energy when air leakage increases

These dollar figures are computed on the pages that follow for a typical utility power generating unit. The same calculations can be applied to actual generating units now in service or projected. The analysis reveals the impressive energy savings of the NASH condenser exhauster.

SEE HOW MUCH MONEY IS INVOLVED			
Steam cost per million Btu	You lose this much with steam jet ejectors	Or this much with a NASH condenser exhauster	So you save this much with NASH
	Dollars per 24-hour day		
$1	$550.80	$194.40	$356.40
$2	$1,101.60	$338.80	$712.80
$4	$2,203.20	$777.60	$1,425.60
$8	$4,406.40	$1,555.20	$2,851.20

System design assumes air leakage

A NASH AT-2006E condenser exhauster in a New England power plant.



How much water vapor will be carried by the saturated air-vapor mixture depends on its temperature, so that subcooling affects the removal capacity requirement.

Venting equipment is installed on a steam condenser to prevent non-condensable gases from accumulating in the vapor space. Small amounts of non-condensables inhibit heat transfer. Large amounts can virtually block the condensation process.

Some of these gases are released from solution in the liquid condensate. Some arrive with the exhaust steam, having been dissolved in the boiler water. The major non-condensable component in a fossil fueled system, though, is air in leakage that finds its way into the sub-atmospheric condenser system.

Although extensive precautions are taken to make the system vacuum-tight, the practical impossibility of achieving this ideal is accepted. Venting equipment, accordingly, is sized to handle the anticipated amount of leakage into a well-installed system of good commercial-quality components. Capacities specified by the HEI are based on system size, expressed as effective steam flow, and on the number of major exhaust openings into the condenser.

Capacities are expressed in terms of SCFM or pounds of dry air per hour and pounds of water vapor to saturate. The air is saturated with water vapor, and this component adds a significant increase to the load of the removal equipment.

Vapor load is calculated to saturate the air at a temperature of 7.5°F below the saturation temperature that corresponds to condenser pressure. (HEI standard) One inch Hg absolute is widely accepted as the condenser pressure for rating and comparison purposes. It corresponds to 79°F in the steam tables, so that the 7.5° subcooling assumption yields air and water vapor at 71.5°F for rating the venting equipment. Under these conditions, the equipment must remove 2.2 pounds of water vapor along with every pound of air.

VENTING EQUIPMENT CAPACITIES
A. One Condenser Shell

Effective Steam Flow *Each* Main Exhaust Opening lbs/hr		Total Number of Exhaust Openings 1	2	3	4	5	6	7	8	9
Up to 25,000	*SCFM	3.0	4.0	5.0	5.0	7.5	7.5	7.5	10.0	10.0
	Dry Air lbs/hr	13.5	18.0	22.5	22.5	33.8	33.8	33.8	45.0	45.0
	Water Vapor lbs/hr	29.7	39.6	49.5	49.5	74.4	74.4	74.4	99.0	99.0
	Total Mixture lbs/hr	43.2	57.6	72.0	72.0	108.2	108.2	108.2	144.0	144.0
25,001 to 50,000	*SCFM	4.0	5.0	7.5	7.5	10.0	10.0	10.0	12.5	12.5
	Dry Air lbs/hr	18.0	22.5	33.8	33.8	45.0	45.0	45.0	56.2	56.2
	Water Vapor lbs/hr	39.6	49.5	74.4	74.4	99.0	99.0	99.0	123.6	123.6
	Total Mixture lbs/hr	57.6	72.0	108.2	108.2	144.0	144.0	144.0	179.8	179.8
50,001 to 100,000	*SCFM	5.0	7.5	10.0	10.0	12.5	12.5	15.0	15.0	15.0
	Dry Air lbs/hr	22.5	33.8	45.0	56.2	56.2	56.2	67.5	67.5	67.5
	Water Vapor lbs/hr	49.5	74.4	99.0	99.0	123.6	123.6	148.5	148.5	148.5
	Total Mixture lbs/hr	72.0	108.2	144.0	144.0	179.8	179.8	216.0	216.0	216.0
100,001 to 250,000	*SCFM	7.5	12.5	12.5	15.0	17.5	20.0	20.0	25.0	25.0
	Dry Air lbs/hr	33.8	56.2	56.2	67.5	78.7	90.0	90.0	112.5	112.5
	Water Vapor lbs/hr	74.4	123.6	123.6	148.5	173.1	198.0	198.0	247.5	247.5
	Total Mixture lbs/hr	108.2	179.8	179.8	216.0	251.8	288.0	288.0	360.0	360.0
250,001 to 500,000	*SCFM	10.0	15.0	17.5	20.0	25.0	25.0	30.0	30.0	35.0
	Dry Air lbs/hr	45.0	67.5	78.7	90.0	112.5	112.5	135.0	135.0	157.5
	Water Vapor lbs/hr	99.0	148.5	173.1	198.0	247.5	247.5	297.0	297.0	346.5
	Total Mixture lbs/hr	144.0	216.0	251.8	288.0	360.0	360.0	432.0	432.0	504.0
500,001 to 1,000,000	*SCFM	12.5	20.0	20.0	25.0	30.0	30.0	35.0	40.0	40.0
	Dry Air lbs/hr	56.2	90.0	90.0	112.5	135.0	135.0	157.5	180.0	180.0
	Water Vapor lbs/hr	123.6	198.0	198.0	247.5	297.0	297.0	346.5	396.0	396.0
	Total Mixture lbs/hr	179.8	288.0	288.0	360.0	432.0	432.0	504.0	576.0	576.0
1,000,001 to 2,000,000	*SCFM	15.0	25.0	25.0	30.0	35.0	40.0	40.0	45.0	50.0
	Dry Air lbs/hr	67.5	112.5	112.5	135.0	157.5	180.0	180.0	202.5	225.0
	Water Vapor lbs/hr	148.5	247.5	247.5	297.0	346.5	396.0	396.0	445.5	495.0
	Total Mixture lbs/hr	216.0	360.0	360.0	432.0	504.0	576.0	576.0	648.0	720.0
2,000,001 to 3,000,000	*SCFM	17.5	25.0	30.0	35.0	40.0	45.0	50.0	55.0	60.0
	Dry Air lbs/hr	78.7	112.5	135.0	157.5	180.0	202.5	225.0	247.5	270.0
	Water Vapor lbs/hr	173.1	247.5	297.0	346.5	396.0	445.5	495.0	544.5	594.0
	Total Mixture lbs/hr	251.8	360.0	432.0	504.0	576.0	648.0	720.0	792.0	864.0
3,000,001 to 4,000,000	*SCFM	20.0	30.0	35.0	40.0	45.0	50.0	55.0	60.0	65.0
	Dry Air lbs/hr	90.0	135.0	157.5	180.0	202.5	225.0	247.5	270.0	292.5
	Water Vapor lbs/hr	198.0	297.0	346.5	396.0	444.5	495.0	544.5	594.0	613.5
	Total Mixture lbs/hr	288.0	432.0	504.0	576.0	648.0	720.0	79992.0	864.0	936.0

*14.7 psia at 70°F
Note: These tables are based on air leakage only and the air vapor mixture at 1 inch HgA and 71.5°F.

Table 9

HEI venting equipment capacities

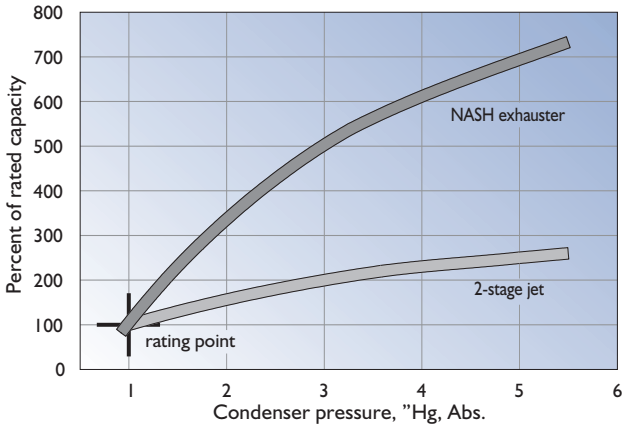
During equipment installation and initial start-up, enough manpower and facilities must be brought in to assure compliance with all performance standards. This broad generalization applies, of course, to condenser air leakage.

The difficulty of leak prevention is revealed by the fact that accepted standards do not require a vacuum-tight condenser system-only that inward air leakage be limited to reasonably low values. Then, over the years, it remains the task of the station operators to prevent those leaks from opening up any more. Such phenomena as foundation settling, vibration, thermal cycling and creep provide mechanisms for leak propagation. Yet the resources for controlling leaks at an operating station are often quite limited.

Inward air leaks can be located approximately by probing with a tracer gas outside the system. Indications of that gas are then detected in the exhaust of the condenser venting system. This process requires special equipment. A certain amount of skill and experience must be developed before plant personnel can locate leaks with accuracy and dependability.

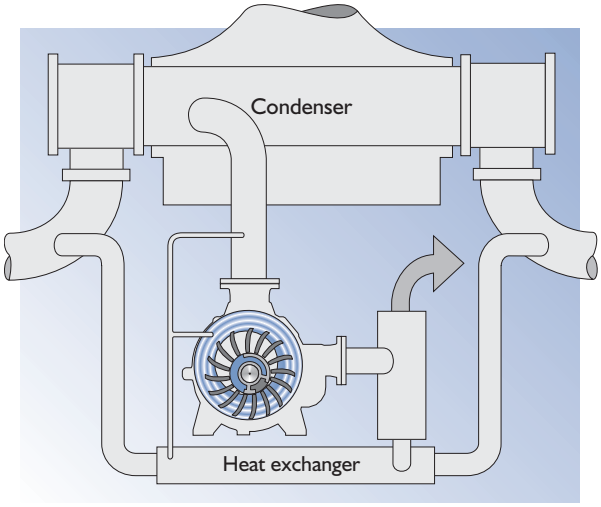
Then, the task of sealing or reducing the size of each leak remains. Some localized sources of inward air leakage can be corrected without much difficulty. When the sources are minuscule and widespread, however, corrective action becomes a continuous campaign.

Controlling leakage is difficult and costly

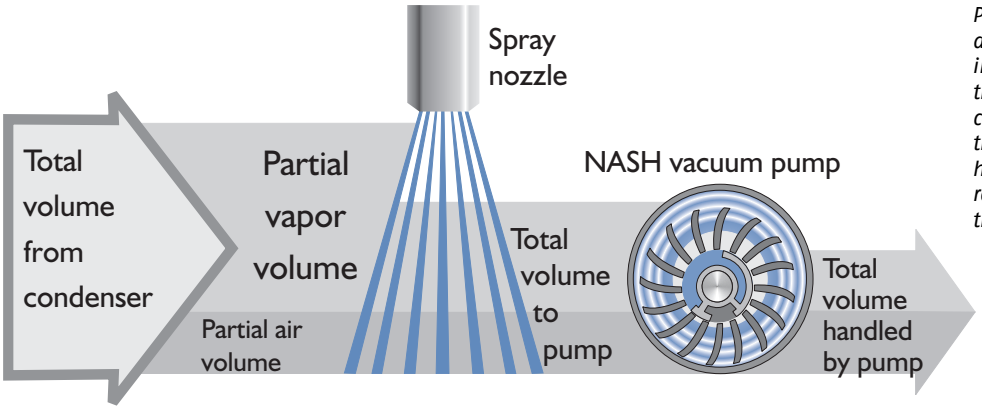


When leakage increases, condenser pressure no longer depends on circulating water temperature. It rises to a higher suction pressure that enables air-removal equipment to handle all the incoming air.

Keeping the condenser system acceptably vacuum-tight becomes more and more difficult as plants age and maintenance staffs dwindle. A condenser exhausting system capable of handling the expected additional air leakage with a minimum backpressure penalty fills this need.



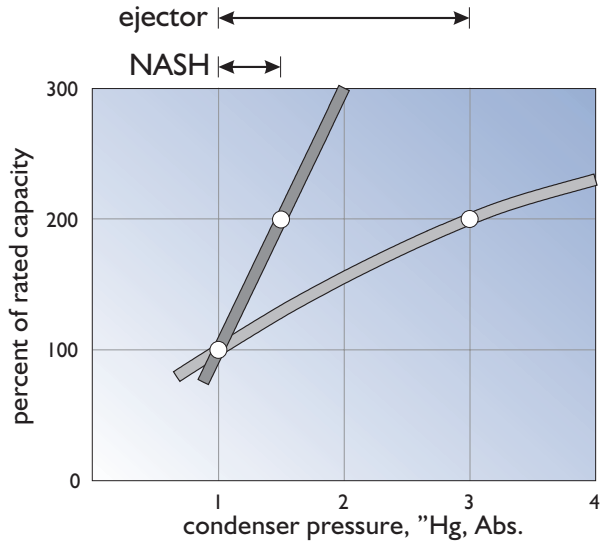
With its heat exchanger cooled by condenser circulating water, the NASH condenser exhauster's operating characteristics match



Part of the water that is used as "liquid pistons" is sprayed into the inlet line ahead of the NASH vacuum pump. It condenses some vapor, and this increases its air handling capacity by reducing volumetric flow into the pump.

A NASH condenser exhauster and a steam jet ejector system sized for the same condenser air-removal duty will both meet new-system leakage requirements at the conventional 1.0"Hg absolute rating point. When air leakage increases, condenser pressure will rise above 1.0"Hg. Although the smaller differential between condenser pressure and atmospheric pressure enables either system to handle the additional leakage, their responses to pressure changes the slopes of their rising capacity curves are not the same.

Although the jet operates with essentially a constant-mass or constant-SCFM characteristic, it does handle more air when it works against a reduced pressure differential. A NASH conical type vacuum pump operates with essentially a constant-displacement characteristic. Because air density increases substantially with any rise in condenser pressure, there are more SCFM available to it for every inlet CFM, and the NASH condenser exhauster's capacity rises steeply.



At double the new-plant air leakage, condenser pressure rises a full two inches with steam-jet ejectors, but only a half-inch with a NASH condenser exhauster.

Another feature of the NASH conical design condenser exhauster also enhances its ability to hold backpressure at low levels. The NASH conical design vacuum pump employs water as a liquid compressant, and a portion of that water is sprayed into the inlet line ahead of the pump. The conical design type of pump accepts large quantities liquid without difficulty. The spray subcools the air/vapor mixture, condensing part of the incoming vapor, which then occupies significantly lower volume when it reaches the inlet of the vacuum pump. Reducing the volume to be handled increases the pump's effective capacity. As temperature of the incoming air-vapor mixture rises with rising backpressure and spray water temperature is unchanged, more vapor can be condensed, and the capacity bonus gets larger.

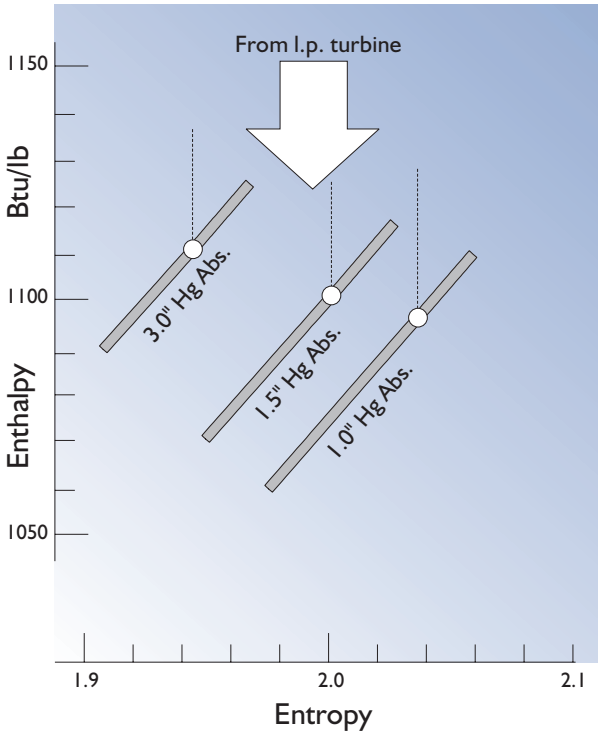
The positive-displacement characteristic and the pump's ability to condense vapor ahead of the vacuum pump give it a performance curve ideally suited to maintaining minimum condenser backpressure.

As an example, when air leakage rises to double the design rating, a NASH condenser exhauster operates at 200% rated capacity with only an approximately 0.3" pressure rise.

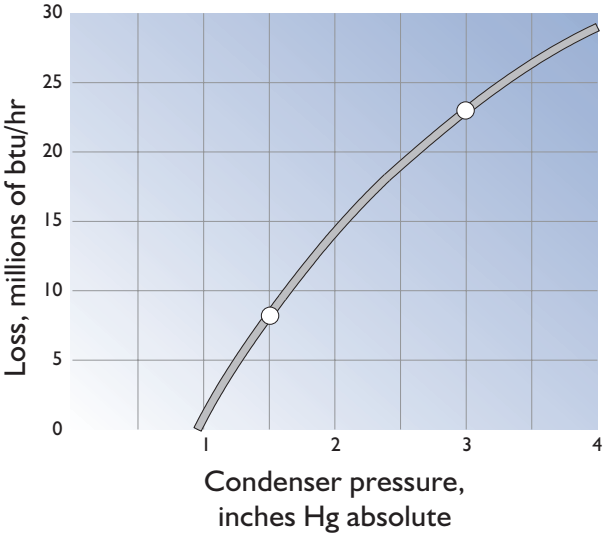
Under the same conditions, a steam jet ejector air-removal system will allow a full two-inch rise to 3.0"Hg absolute before attaining 200% of its rated capacity. This 1.5" Hg difference in warm climates and summer cooling water temperatures can sometimes cause power reductions which can be very expensive to the power plant.

Lost energy represented by a rise in condenser pressure is quite easy to calculate. Although energy transfers at the high end of the enthalpy scale in a modern steam plant may involve complexities such as several reheat cycles, LP turbine exhaust presents an unambiguous measure of energy utilization at the low end. When turbine backpressure goes up, a measurable quantity of work becomes unavailable.

Rising condenser pressure imposes an energy loss



When increased air leakage raises turbine backpressure, less energy can be extracted from the thermal cycle.



You can plot a curve like this for any steam generating unit that concerns you. Just substitute an appropriate LP turbine steam flow in the calculations above.

If the pressure rise occurs because of a rise in circulating water temperature, the resulting energy loss-at that site and under those weather conditions-is unavoidable. It becomes part of the cost of using steam to generate electric power at that time and place.

But if the pressure rise occurs because air in leakage into the condenser system cannot be removed at full vacuum, this energy loss justifies some concern.

Here are the enthalpy figures for a rise from 1.0"Hg to 3.0"Hg absolute:

Enthalpy at 3.0" Hg abs.	*1111.6 btu/lb*
Enthalpy at 1.0" Hg abs.	*1096.3 btu/lb*
Energy loss (jets)	*15.3 btu/lb*

To express this loss in terms of the entire condenser, we assume a typical 300-megawatt unit with a low-pressure turbine steam flow of 1.5 million pounds per hour.

(15.3 Btu/lb)(1,500,000 lb/hr)
= 22,950,000 btu/hr

That energy loss, which occurs when a steam jet ejector allows condenser pressure to rise from 1.0" to 3.0"Hg with a doubling of air leakage, can be compared with the smaller loss for a NASH condenser exhauster under the same conditions. Which only allows the pressure to rise only to 1.5"Hg absolute, the enthalpy figures are:

Enthalpy at 1.5" Hg. abs.	*1101.7 btu/lb*
Enthalpy at 1.0" Hg. abs.	*1096.3 btu/lb*
Energy loss (NASH)	*5.4 btu/lb*

At 1.5 million pounds of steam per hour through the low-pressure turbine, this is:

(5.4 btu/lb)(1,500,000 lb/hr)
= 8,100,000 btu/hr

Costs on which to base your evaluation

Because management decisions are usually based on dollars, an evaluation of condenser venting performance will be most meaningful in those terms. Enthalpy penalties at the exhaust end of the low-pressure turbine can, of course, be translated into a variety of reference units. We show typical dollar figures with respect to costs per million Btu at the steam segment of the power generation cycle. They afford a quick look at the magnitudes involved. More exact numbers can be calculated using the efficiencies and fuel costs that apply in individual cases.

Thus, for a steam cost of $4 per million Btu, the energy loss due to an increase of turbine backpressure to 3"Hg absolute with steam jet ejectors would be:

($4)(22.95 million Btu/hr)(24 hr/day) = $2,203.20 per day

With a NASH conical design condenser exhauster that limits turbine backpressure to 1.5"Hg under the same conditions, the loss would be reduced to:

The difference represents a saving of $1,425.60 per day.

($4)(8.1 million Btu/hr)(24 hr/day) = $777.60 per day

Dollar figures tabulated on the front cover of this bulletin are computed the same way.

Losses on this scale warrant investigation and appropriate action. One obvious first step in an existing plant is to attack the condenser leakage problem directly. Any reduction of inward air leaks that can be accomplished at reasonable costs should have a high priority.

Next, the NASH conical design condenser exhauster, with its steep-rising capacity curve, must be considered in planning for new units and at existing stations as well.

Some other benefits that support the choice

Only one aspect of condenser exhauster performance has been emphasized in the foregoing discussion. But a NASH condenser exhauster's ability to cope with excess air leakage is not the only reason for its preference by consulting engineers and by the owners and operators of steam power plants.

Its durability has been thoroughly demonstrated. Any initial concern about a motor-driven vacuum pump with one moving part in place of ejectors without any major moving parts has been dispelled by experience. The NASH pump is widely recognized as a rugged machine, quite capable of running continuously year-in and year-out. It is designed and built to run for the life of the power station without trouble and with minimum maintenance.

The rotor on a direct-driven shaft constitutes the moving part. The NASH two-stage pump runs at a relatively low speed without pulsation. It creates little noise or vibration.

Antifriction bearings are external. Shaft packing and lantern glands are accessible. Those are the maintenance items.

NASH builds the condenser exhauster package by assembling the two-stage vacuum pump, its motor and all accessory equipment on a structural steel base. Every vacuum pump is factory-tested before shipment. NASH runs dry-air capacity tests that correlate accurately with exhauster performance under saturated-air condenser conditions. Certified reports of the factory tests are made available to customers without charge.

The NASH condenser exhauster package is sold with a two-year warranty.

Qualified NASH people check the details of each installation and witness exhauster start-up to assure that station personnel will encounter no operating problems.

Operation of the NASH condenser exhauster is automatic. A single control-room switch starts and stops it.

The steep-rising capacity curve of the NASH pump, which operates in a single-stage mode during hogging, makes separate hogging equipment unnecessary. Transitions between the hogging and holding modes are automatic.

For cycling plants, when the turbine is idle, the gland seal steam can be supplied from an auxiliary steam boiler and the condenser exhauster can be left running. This greatly reduces startup time and getting quality into specifications on restart.

For a time-saving start-up, the NASH condenser exhauster can begin hogging as soon as the turbine glands have been sealed. Its operation depends only on station power, not on availability of steam pressure.

The cool water spray introduced at the inlet ahead of the NASH vacuum pump was mentioned as an important contributor to excellent performance at high air leakage rates. This same feature enables the NASH condenser exhauster to track the vacuum changes that occur normally when circulating water temperature changes. With its heat exchanger cooled either by circulating water or by an equally cool stream from another source, the NASH vacuum pump's operating characteristics stay right in step with the operating requirements of the condenser.

Fouling of boiler water is avoided by discarding carryover and condensate from the vented air. Contaminant gases, such as ammonia, are thus purged continuously instead of being allowed to redissolve into the feedwater cycle.

Some of the points made briefly here deserve more explanation than space in this presentation allows. Your NASH representative will be glad to go through the fundamentals on which they are based.

Other NASH Products

2BV

Compact liquid ring vacuum pumps built for serious cost savings

Use up to 50 percent less water than other liquid ring pumps

Monoblock and pedestal designs available

Capacity of 4 to 350 CFM with vacuum to 29+″ HgV

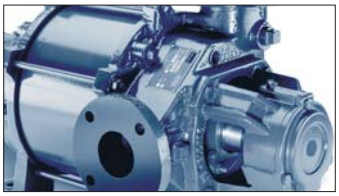
Vectra

Liquid ring vacuum pumps and compressors

Available in feature rich budget designs (XL or GL)

Designed to handle high back pressure requirements

Capacity of 115 to 2,860 CFM with vacuum to 29″ HgV



2BE3

Large liquid ring vacuum pumps with superior corrosion resistance

Top discharge capability which eliminates need for trench

Self-recirculating seal water, reducing need for external seal water source

Capacity of 4,000 to 22,000 CFM with vacuum to 29+″ HgV

TC-TCM

Integral 2 stage liquid ring pumps with improved performance at vacuum levels down to 0.8" HgA.

Designed to handle large amounts of liquid carryover without difficulty

Capacity of 100 CFM to 2200 CFM with vacuum to 0.8" HgA

NASH

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