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TITLE: ESTIMATING THE COST OF SUPERCONDUCTING MAGNETS AND THE REFRIGERATORS NEEDED TO KEEP THEM COLD

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TITLE: CAPACITY REQUIREMENTS FOR PRESSURE RELIEF DEVICES ON CRYOGENIC CONTAINERS

AUTHORS: A.P. Varghese and B.X. Zhang

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ABSTRACT

The cost of superconducting magnets and the refrigerators needed to keep them cold can be estimated if one knows the magnet stored energy and the amount of refrigeration needed. This report updates the cost data collected over 20 years ago by Strobridge and others. Early cost data has been inflated into 1991 dollars and data on newer superconducting magnets has been added to the old data. The cost of superconducting magnets has been correlated with stored energy and field-magnetic volume product. The cost of the helium refrigerator cold box and the compressors needed to keep the magnet cold can be correlated with the refrigeration generated at 4.5K. The annual cost of 4.5K refrigeration can be correlated with 4.5K refrigeration and electrical energy cost.

INTRODUCTION

It is often difficult to get a budgetary estimate of the cost of a superconducting magnet system and the helium refrigeration system needed to keep it cold. This report presents one method for making a budgetary cost estimate of both components based on knowing what these components have cost in the past. One of the difficulties with this kind of estimate is the choice of the appropriate scaling parameter.

As an example for superconducting magnets, the appropriate scaling parameters may be stored energy, average induction multiplied by the field volume, or magnet and cryostat mass. The choice of scaling parameter depends on the type of magnet being estimated.

For helium refrigeration systems, the choice of scaling parameter is easier to determine. This report uses the refrigeration capacity at 4.5K. Refrigeration at other temperatures is scaled appropriately. Liquefaction is converted to 4.5K refrigeration by the use of the refrigeration liquefaction coefficient for the machine.

THE COST OF SUPERCONDUCTING MAGNETS

As superconducting magnet systems increase in size and complexity, it is appropriate to analyze the corresponding trends in the costs of the major constituent components: the magnets themselves and the refrigeration required to maintain them in operation. Every decade or so, such an analysis appears in print, usually directed at specific applications. In the early seventies, when advances in plasma physics made prototype fusion reactors feasible, a number of interesting economic assessments of such devices were published.[1,2] Ten years later, superconducting energy storage reached respectability and so its economics were scrutinized.[3,4] The purpose of this paper is to take a representative cross-section of superconducting magnet systems of all types and using known costs, to put the constants of the well-known cost equations for superconducting magnets onto the 1991 basis.

The composition of our sample includes six accelerator magnets, nine dipole like MHD magnets, thirty solenoid type magnets and fourteen toroidal magnets. In size, the magnets varied from a small dipole magnet, with a stored magnetic energy of about 27 kJ to systems with stored energies in excess of 1000 MJ. Only completed systems were considered: studies, planned projects and the like were excluded from the survey.

Methodology

The system characteristics were obtained from a systematic perusal of the published literature, which included technical reports circulated among interested institutions, and confirmed by direct inquiry. For the costs, the "Technical Proposal" or its equivalent was the usual starting point, followed by an actual tracking of the project costs through information obtained from the funding agency or its representative organ. In the US, this is often simply a matter of identifying the appropriate government publication; abroad, it requires a network of helpful correspondents and friendly reciprocity. In spite of the disparity of the sources, the raw data were usually reliable to about 15%.

A magnet system was assumed to be completed on the date of its first successful acceptance test. The purpose of this artificial cut-off is to better isolate the construction costs from subsequent tuning improvements which tend to have a life and hence associated costs of their own. The actual project cost was then converted to 1991 dollars using the composite escalation index for large construction projects. Foreign project costs were converted to US currency using the exchange rate at the time of construction and then treated in the same manner as domestic projects.

Two parameters were used to characterize each system: the energy stored in the magnetic field, in MJ, without corrections for field containment, and the field-magnetic volume product, in Tm^3. This latter parameter is in certain instances a better measure of the system performance than the energy, because it attempts to define the actual portion of the magnetic field exploited by the process or device.

Results

Figures 1 and 2 are the scatter diagrams of the cost-magnet parameter relationships for the entire sample, regardless of magnet type. The lines in each figure are least square fits to the data points. The overall cost of the magnets given in Fig. 1 can be represented by the following equation:

$$C(M\$) = 0.844\ [E(MJ)]^{0.459} \qquad (1)$$

and

$$C(M\$) = 0.770\ [\Omega(Tm^3]^{0.631} \qquad (2)$$

where C is the magnet cost; E is the stored energy, and Ω is the field-magnetic volume product.

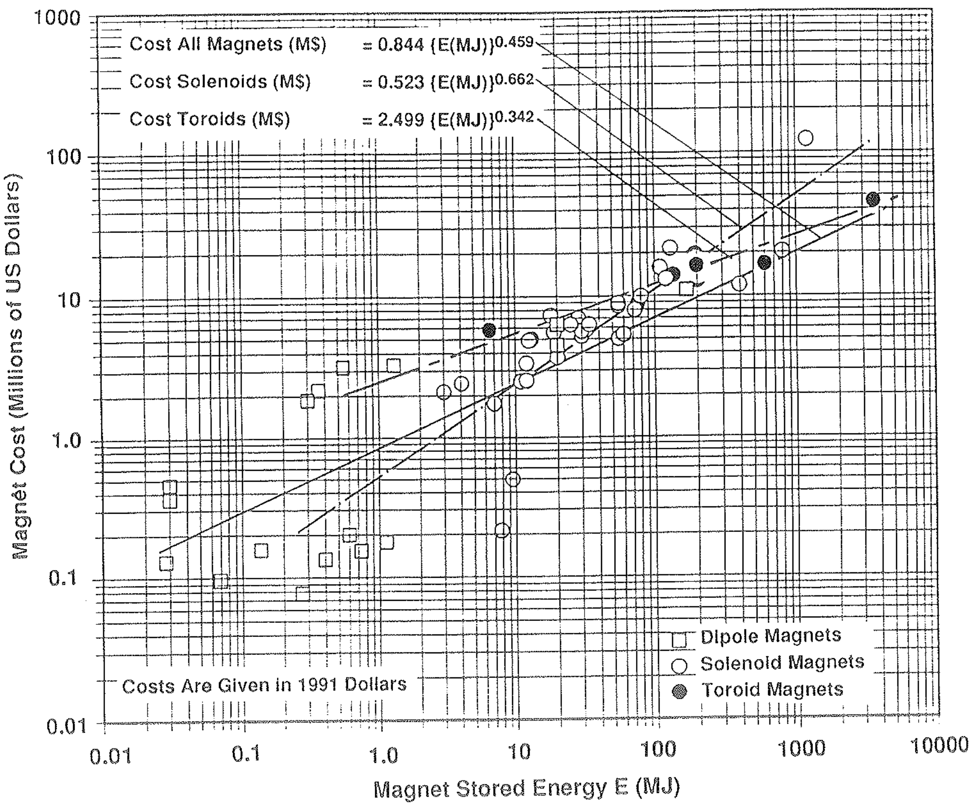

Figure 1. Superconducting Magnet Costs Versus Magnet Stored Energy.

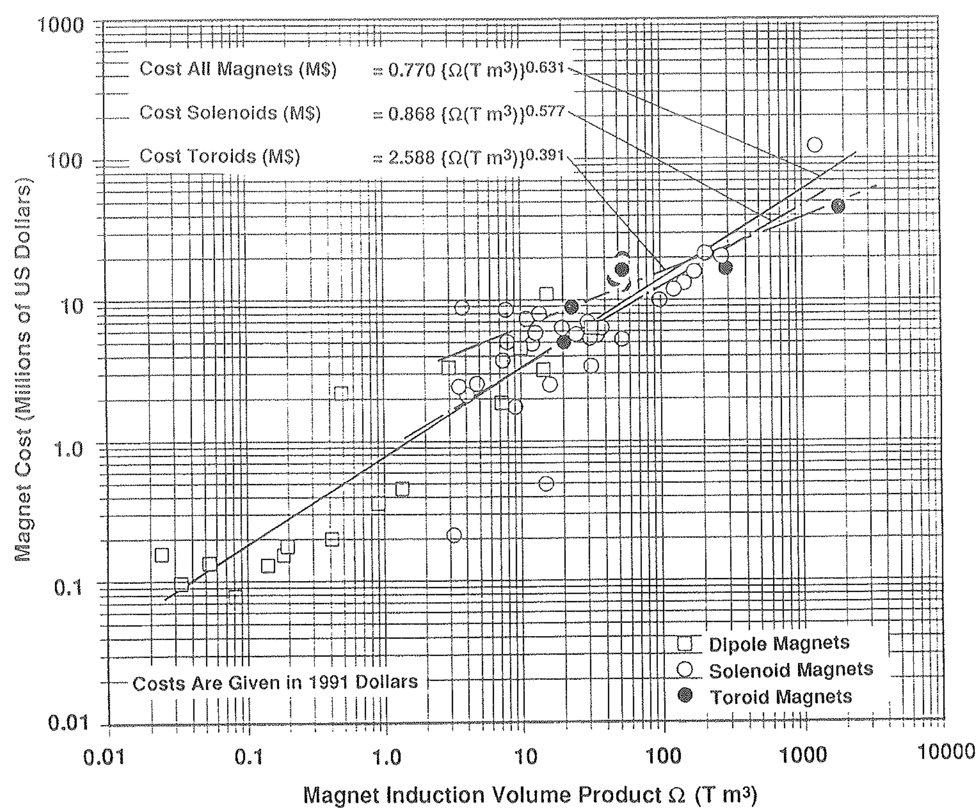

Figure 2. Superconducting Magnet Cost Versus Field-Magnetic Volume Product.

It is interesting to note that in neither case is the index even remotely close to that usually quoted in analyses of this kind.

When we separated solenoidal magnet systems, which include split coil magnets, and toroidal magnets from the data, one gets lines with different slopes (see Fig. 1 and 2). For the solenoid type of magnets, the cost equations take the following form:

$$C(M\$) = 0.523\ [E(MJ)]^{0.662} \tag{3}$$

and

$$C(M\$) = 0.868\ [\Omega(Tm^3]^{0.577} \tag{4}$$

where C, E, and Ω are defined as before. For the toroid type of magnets, the cost equations take the following form;

$$C(M\$) = 2.499\ [E(MJ)]^{0.342} \tag{5}$$

and

$$C(M\$) = 2.588\ [\Omega(Tm^3]^{0.391} \tag{6}$$

where C, E, and Ω are defined previously.

Discussion

We cannot treat accelerator and beam transport magnets in the same manner as they are invariably manufactured in considerable quantities starting from one or more prototypes. Our analysis thus provides a poor estimate of the unit cost: the prototype(s) will be wildly underestimated, while the production models will appear to be considerably more expensive. However, the total installation (accelerator, beam line) cost will follow a power law, whose constants can be determined from previously built systems.

THE COST OF HELIUM REFRIGERATION

In 1966, Strobridge, Mann and Chelton[5] developed a technique for estimating the cost of helium refrigerators based on a limited number of cost data points available at the time. In 1969, Strobridge[6] updated his study to include cryogenic refrigerators of all types. The cost of refrigeration was estimated based on the input power to the compressor. The 1969 Strobridge study was expanded in 1974[7] to include a number of newer refrigerators being built at that time. During the period between 1966 and 1974, the cost of helium refrigeration did not change. From 1974 to the present, the capital cost of refrigeration appears to have escalated at the nominal rate of inflation.

This report presents one method for making a budgetary cost estimate of superconducting magnet refrigerators based on knowing what these components have cost in the past. One of the difficulties with this kind of estimate is the choice of the appropriate scaling parameter. For helium refrigeration systems, the refrigeration capacity at 4.5K is used as a scaling factor.. Helium refrigeration at other temperatures is scaled to 4.5 K using the Carnot ratio. (One can divide 4.5 K by the refrigeration temperature to obtain the Carnot ratio.) Liquefaction is converted to 4.5K refrigeration by the use of the refrigeration liquefaction coefficient (typically 75 to 125 J g^{-1}).

The Thermodynamic Efficiency of Helium Refrigerators

Strobridge in his 1966[5], 1969[6] and 1974[7] papers discussed the efficiency of various kinds of refrigerators. Efficiency was defined as the input power of a perfect Carnot cycle refrigerator over the real compressor power which goes into the refrigerator. An efficiency plot which contains the Strobridge helium refrigerator data as well as newer data is shown in

Figure 3. The efficiency data shown in Figure 3 shows a great deal of scatter as the original Strobridge data did. Most of the points shown in Figure 3 have liquid nitrogen precooling. The has the effect of enhancing the apparent efficiency of the machine. The addition of a liquid nitrogen precooler increases the apparent efficiency by a factor of 1.5 to 1.8.

The newer data points shown in Figure 3 show that on average the overall efficiency of helium refrigerators has not increased. There are a number of reasons for this: 1) There are number of the newer points in Figure 3, (particularly those clustered between 80 and 400 W) are for machines without liquid nitrogen precooling. 2) Many of the new machines use rotary compressors (Screw compressors are the most common.). These compressors are more reliable than the older piston compressors but they are less efficient (particularly if they are small single stage machines). 3) There are more turbine expanders in smaller machines. Some of the machines built in the early 1980's are not as efficient as machines which were built later. The modern plants which are more efficient than average have two or more stages of compressors and have expanders which are staged as well.[8] Small piston expanders are more efficient than small turbine expanders. As the size of the plant grows, the turbines have efficiencies which are competitive with piston expanders.

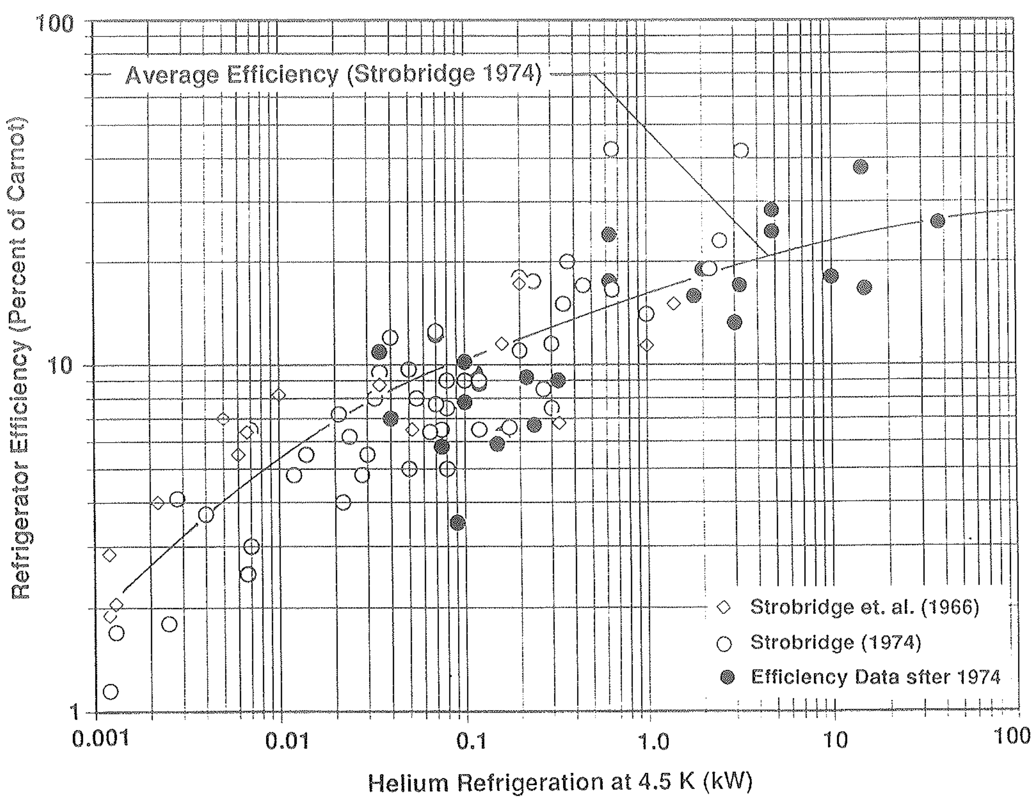

Figure 3. The Efficiency of Helium Refrigerators as a Function of 4.5 K Refrigeration

Refrigerator Cost

Figure 4 shows the cost of various 4.5K refrigerators escalated into 1991 dollars as a function of the 4.5K refrigeration. Liquefaction was converted to refrigeration using the refrigeration-liquefaction coefficient for the machine (typically 75 to 125 W per gs^{-1} depending on how the cycle has been optimized). Refrigeration at temperatures different from 4.5K have been converted to 4.5K refrigeration by using the Carnot ratio. The cost of foreign made machines was converted to dollars at the exchange rate of the year of manufacture. The dollars were escalated from the year of manufacture to 1991 dollars.

From Fig. 4, one can see that refrigerators made before 1966 are more expensive in 1991 dollars than refrigerators made after 1974. In Fig. 4 there is a line plotted with the cost points. This line represents the average cost in 1991 dollars of modern helium refrigerators which produce refrigeration from 0.040 to 15 kW. The equation for this line is:

$$C(M\$) = 1.51[\, R(kW)]^{0.7} \qquad (7)$$

where the cost is given in 1991 dollars and R, the 4.5K refrigeration.

The small refrigerators (less than 30 W) in general cost more than the curve shown in Fig. 4. The largest plants shown in Fig. 4 are quite complex. Some have several cold boxes tied together and others may include helium pump systems to circulate subcooled

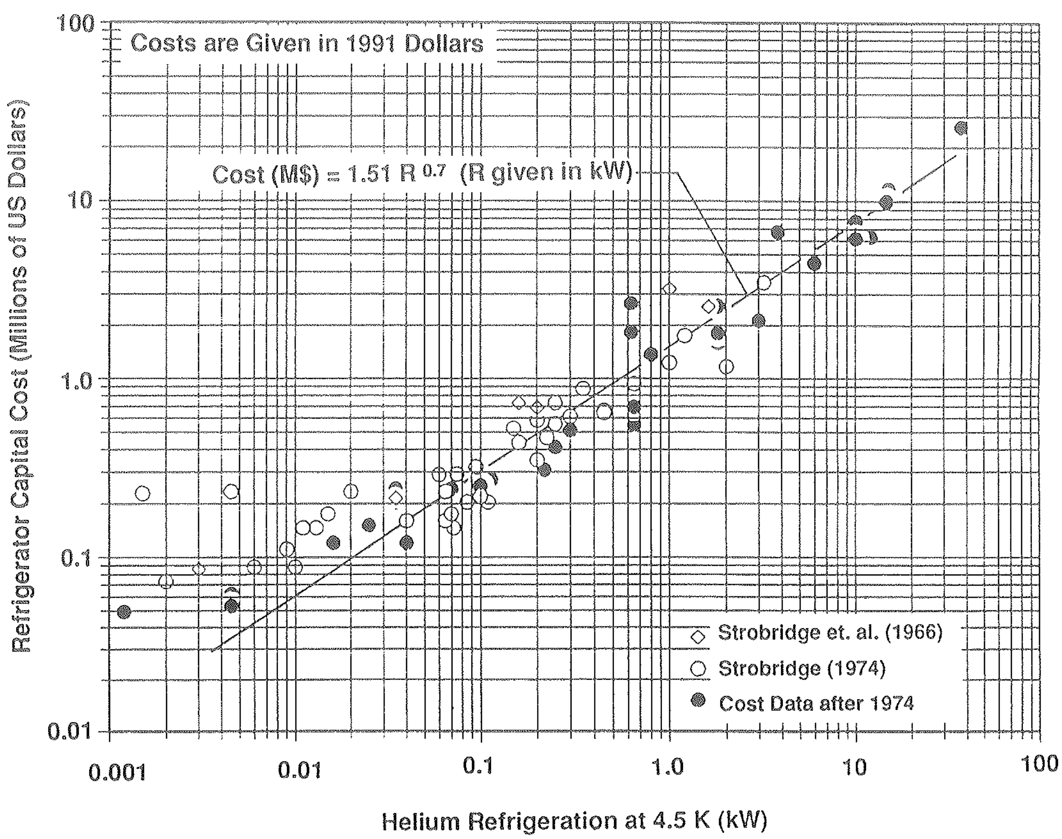

Figure 4. The Cost of Helium Refrigerators versus Refrigeration at 4.5 K

helium. As a result, these plants can be more expensive. Other factors which increase the cost of refrigeration include: computer control systems (In small machines they are usually not needed.), extra purification in the cold box (sometimes a blessing, sometimes a curse), and the extra documentation required by military type specifications.

The annual cost of refrigeration can also be estimated as a function of the amount of refrigeration delivered at 4.5 R and the cost of electric power P. The annual cost of refrigeration includes; amortization of the refrigerator, depreciation, operation and maintenance labor, electric power, liquid nitrogen cooling and compressor cooling. If one assumes that 22 percent of the capital cost goes for the annual cost amortization, depreciation, operation and maintenance, the following equation can be used to estimate the annual cost of providing 4.5 K helium refrigeration for a superconducting magnet system:

$$\text{Annual Cost(M\$/yr)} = 2.72[\,R(\text{kW})]^{0.78}\,[P(\$/\text{kWh})]^{0.56} \qquad (8)$$

Equation 8 is applicable over a range of refrigerations from 0.03 to 30 kW and a range of electrical energy costs from 0.04 to 0.18 dollars per kilowatt hour. About half of the annual cost of refrigeration is related to the cost of energy and cooling. Organizations which do not amortize or depreciate their equipment can expect an annual refrigeration cost about two thirds of that given by Equation 8. The annual cost given in Equation 8 can be expected to escalate at a rate about 60 percent of the rate of inflation.

Discussion

The cost of liquid helium refrigeration can be characterized by a simple equation such as Equation 8. Multiple units change the cost picture somewhat. Unlike superconducting accelerator magnets, there are not hundred or thousands of 4.5 K helium refrigerators being made all at once. As a result, there is not much in the way of economy of scale.

ACKNOWLEDGMENTS

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E.E - 2

Presented at CEC-91

CAPACITY REQUIREMENTS FOR PRESSURE RELIEF DEVICES ON CRYOGENIC CONTAINERS

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ABSTRACT

An equation is derived for the evaluation of the capacity requirements for pressure relief devices on cryogenic containers. It comprises terms such as the heat transfer rate into the container and the specific heat input corresponding to a given situation. This equation can be used to calculate the capacity requirements of pressure relief devices on cryogenic containers irrespective of the thermodynamic state of the cryogen. This equation is developed on the premise that all the vapor generated by the heat input need not be released to maintain a constant pressure inside the container.

The equation is then adapted for the prevalent cases such as liquid full–liquid discharging, saturated liquid and vapor–vapor discharging, saturated liquid and vapor-liquid discharging and super critical fluid discharging.

In adverse situations, such as an overturned cryogenic container on fire, connections to the pressure relief devices may be covered by liquid. For this reason, discharge rates for vapor and liquid, as well as super critical cryogens are discussed and design guidelines are provided.

INTRODUCTION

Pressure relief devices are required in all process, transfer, transportation, and storage systems involving fluids subject to unplanned pressure rises. While planned pressure increases are almost always contained by the system boundaries, those exceeding system test pressure and all unplanned pressure rises above the system design pressure have to be relieved through pressure relief devices to prevent catastrophic failures of the system. The pressure relief devices limit the pressure increases by discharging the fluids from the system.

The majority of the equations in use for determining the capacity requirements of pressure relief devices are based on the basic requirement of discharging all the vapor generated by the heat input from a fire or other emergency situations. While this approach is conservative, it imposes a heavy financial burden when the set pressure of a relief device is close to the critical pressure of the fluid.

A single equation is developed in this paper for calculating true capacity requirements of pressure relief devices for a fluid irrespective of its thermodynamic state. This equation is developed on the premise that all the vapor generated by the heat input need not be released to maintain a constant pressure inside the container.

Methods for Calculating the mass discharge requirements for fluid systems with design pressures far lower than the critical pressures of the fluids concerned are covered in publications of American Petroleum Institute (API) and Compressed Gas Association (CGA). In such literature, the mass discharge rate requirements are dependent on the rate of heat transfer into the fluid or rate of energy generation within the fluid, and the properties of the fluid. To keep the discussion simple and general, mathematical expressions for the discharge requirements will be derived for a case of heat transfer into the fluid. In the CGA and API expressions for mass discharge rate requirements, the heat transfer into the fluid system causes a certain mass of fluid to change phase and this mass of fluid has to be discharged to keep the pressure from rising. In this formulation, the density of vapor is considered negligible compared to the density of liquid. This is true only when the pressure of the fluid is far lower than the critical pressure of the fluid which is generally the case for most fluid systems. However, fluids like helium and hydrogen have critical pressures low enough to fall in the working pressure ranges of their cryogenic systems. The expressions developed here are applicable to fluids involved in both single phase or change of phase processes.

THEORY

API-CGA Formula

The API-CGA mass discharge rate formula essentially has its basis in the relation

$$\dot{m} = \frac{\dot{Q}}{L} \tag{1}$$

It is evident from thermal property tables of fluids that the latent heat of vaporization decreases as the pressure approaches its critical value, and goes to zero at pressures equal to or greater than the critical pressure. This will give an infinitely large mass discharge rate near, at and above the critical pressure if the above basic relation is used. It will also be convenient to have a general expression that could be simplified to suit the conditions at hand.

Specific Heat Input

It is easy to see that the API-CGA basic formula requires all the vapor generated by the heat flow into the fluid be discharged. The primary function of a pressure relief device is to keep the pressure at or below a safe limit for the fluid system. When heat is added to a fluid system at constant pressure, the fluid needs additional space to expand to maintain a constant pressure. The mass of fluid in this additional space has to be vented to maintain the pressure constant.

Consider a mass of fluid m at pressure p and temperature T in a cylinder with a piston as illustrated in Figure 1, The first law of thermodynamics for the system yields,

$$\begin{aligned} \delta Q &= dU + \delta W && (2) \\ \delta W &= pdV && (3) \\ H &= U + pV && (4) \\ dH &= dU + pdV + Vdp. && (5) \end{aligned}$$

For a constant pressure process,

$$dp = 0 \tag{6}$$

and therefore

$$dH = \delta Q \tag{7}$$

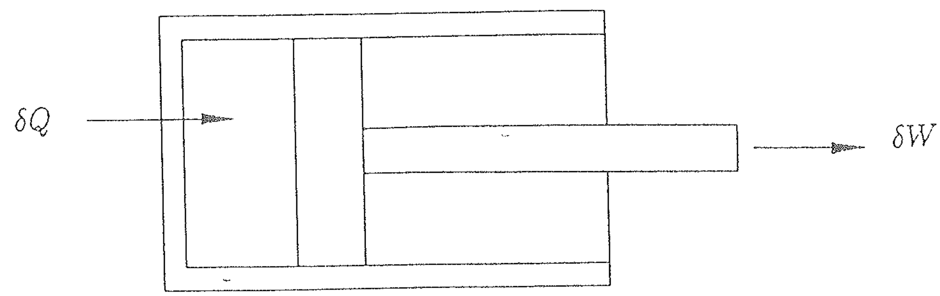

Figure 1: Illustration of First Law of Thermodynamics.

For a heat input of δQ to a fluid system at constant pressure, let the increase in volume of the fluid be dV. A fluid system that has to be protected by a pressure relief device almost always has a fixed volume. In a constant volume fluid system, the relief devices should relieve the mass

$$\begin{aligned} m_l &= \frac{dV}{v} \\ &= m\frac{dv}{v} \end{aligned} \tag{8}$$

to keep the pressure constant for a heat input of

$$\delta Q = mdh \tag{9}$$

Thus the mass expulsion needed per unit of heat transfer to maintain constant pressure is

$$\begin{aligned} \frac{m}{\delta Q} &= m\frac{dv}{vmdh} \\ &= \frac{1}{v\left(\frac{dh}{dv}\right)} \\ &= \frac{1}{v\left(\frac{\partial h}{\partial v}\right)_p} \end{aligned} \tag{10}$$

as a constant pressure process was assumed. Values of the specific heat input

$$\theta = v\left(\frac{\partial h}{\partial v}\right)_p \tag{11}$$

are given in tables of thermophysical properties of various fluids from National Institute of Standards and Technologies (NIST) They can also be generated by using computer programs[5].

Therefore, for a heat transfer rate of $\dot{Q}$, the mass expulsion rate of fluid required to maintain a constant pressure is given by

$$\dot{m} = \frac{\dot{Q}}{\theta} \tag{12}$$

APPLICATION TO FLUID SYSTEMS

The expression for mass expulsion rate derived above in terms of the specific heat input θ could now be adapted for fluid systems involving different phases.

Liquid Full System–Liquid Discharging

If values of the specific heat input θ are calculated at different temperatures for a liquid under constant pressure p, the minimum value of the specific heat input θ_{min} will be found at the saturation temperature. For this θ_{min}, a mass expulsion rate calculated using Equation 12 will be considered as the minimum mass rate of liquid flow requirement. This system will eventually reach a condition when it will contain both liquid and vapor phases.

Fluid System Containing Saturated Liquid and Vapor

In this case, any heat transfer into the fluid system at constant pressure will result in evaporation of the saturated liquid. The minimum mass discharge rate required for this case will depend on whether the pressure relief devices are connected to the vapor or liquid side. Generally, pressure relief devices are connected to the vapor side. However, under accident conditions like overturned fluid transport tanks, connections to the pressure relief devices could end up covered by liquid. These two cases have to be considered separately. For an evaporation process, the differential quantity in the specific heat input term $v(\partial h/\partial v)_p$ can be replaced by difference quantity to give

$$v\frac{\Delta h}{\Delta v} = v\frac{L}{v_{fg.}} \tag{13}$$

a. Vapor Discharge: If pressure relief devices are connected to the vapor side, the mass discharge will be in vapor form and the minimum mass discharge rate requirement expression will become

$$\dot{m} = \dot{Q}\left(v_g\frac{L}{v_{fg}}\right)^{-1} \tag{14}$$

b. Liquid Discharge: In cases where relief device connections are covered by liquid resulting in liquid discharge, the minimum mass discharge rate requirement equation for constant pressure becomes

$$\dot{m} = \dot{Q}\left(v_f\frac{L}{v_{fg}}\right) \tag{15}$$

Obviously, this can yield a very high relief device capacity requirement compared to other cases if the pressure in the system is far lower than the critical pressure. It may be less expensive to eliminate the possibility of occurrence of this condition by altering the design of the system rather than providing high capacity pressure relief devices.

Vapor Full or Super Critical System

For single phase, constant pressure systems, the specific heat input θ is temperature dependent. Variation of θ with respect to temperature for hydrogen at $1,379\ kPa$

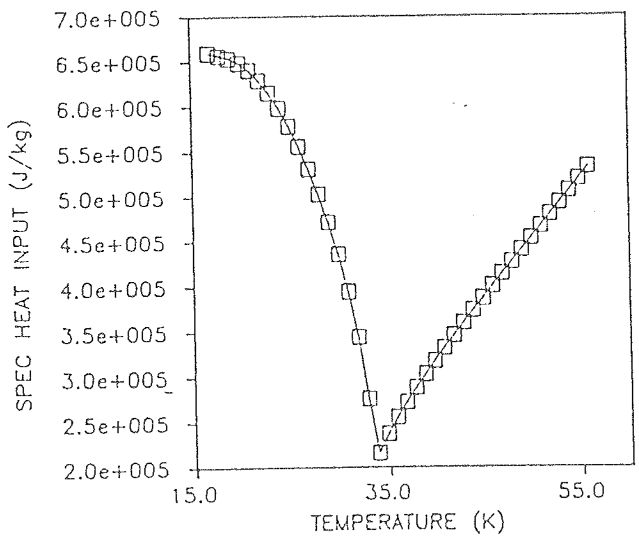

Figure 2: Specific heat input as a function of temperature for hydrogen at 1,379 kPa.

is shown in Figure 2. It is evident from the above curve that the specific heat input for a fluid has its minimum value at a particular temperature for a specified pressure. Thus, the mass discharge rate required to maintain pressure below a certain value is given by

$$\dot{m} = \frac{\dot{Q}}{\theta_{min}} \tag{16}$$

DESIGN OF RELIEF DEVICES

Capacities of relief devices are generally stated by the manufacturers in Standard Cubic Feet per Minute (Standard Cubic Meter per Hour) of air at the set pressure of the device. The formula for the required flow capacity in Standard Cubic Feet per Minute (Standard Cubic Meter per Hour) of air is obtained by combining the equations developed in this paper with the formula given in ASME Code Section VIII, Division 1. The formula for required capacity becomes

$$\dot{F}_a = \frac{D}{C}\left(\frac{p}{RM}\right)^{1/2} \dot{m}\sqrt{v} \tag{17}$$

where v is the specific volume of the fluid at the inlet of the relief device and D is a combination of the density and time conversion multipliers.

$\dot{F}_a$ will be a maximum if $\dot{m}\sqrt{v}$ is a maximum. For single phase fluids, if $\dot{Q}$ is constant, $\sqrt{v}/\theta$ has to be a maximum to yield the maximum required capacity. At constant pressure, as v and θ varies with temperature, the ratio $\sqrt{v}/\theta$ shall be maximized with respect to temperature.

Locating the relief device away from the cryogenic container can raise the fluid temperature at the inlet of the device. In such a case, the ratio of the square root of the specific volume of the fluid at the temperature at the inlet of the relief device to the specific heat input at the temperature of the fluid inside the container shall be maximized.

CONCLUSION

Realistic mass flow rate requirements are provided by the expressions developed in this work for fluid systems frequently encountered in practice. In designing pressure relief devices for fluid system, evaluation should be made of the probabilities of occurrences of each of the cases discussed here or any special cases that may develop. The minimum mass discharge rate requirements for each and every case and any probable combinations of cases shall be computed and pressure relief devices capable of meeting the highest flow requirement shall be provided.

NOMENCLATURE

C – constant for gas or vapor related to ratio of specific heats, given in ASME Code
$\dot{F}$ – volume flow rate
H – enthalpy
h – enthalpy of unit mass
L – latent heat of vaporization
M – molecular weight of fluid
$\dot{m}$ – mass discharge rate required
m – mass
p – pressure
Q– quantity of heat absorbed by system
$\dot{Q}$ – heat transfer rate
R – universal gas constant
T – temperature
U – internal energy
u – internal energy of unit mass
V – volume
v – specific volume
W – work
θ – specific heat input

Subscripts

a – air equivalent
f – liquid phase, saturated
g – vapor phase, saturated
fg – difference in saturation property between liquid and vapor
l – expulsion required
p – under constant pressure

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