Purdue University
Purdue e-Pubs

International Compressor Engineering Conference | School of Mechanical Engineering

2014

A Chiller Control Algorithm for Multiple Variable-speed Centrifugal Compressors

Piero Caballero
Florida State University, United States of America, plc11@my.fsu.edu

Chiang Shih
Florida State University, United States of America, cshih@admin.fsu.edu

W. Turner Thornton
Danfoss Turbocor, United States of America, U266040@danfoss.com

Joost J. Brasz
Danfoss Turbocor, United States of America, U265950@danfoss.com

Follow this and additional works at: https://docs.lib.purdue.edu/icec

Caballero, Piero; Shih, Chiang; Thornton, W. Turner; and Brasz, Joost J., "A Chiller Control Algorithm for Multiple Variable-speed Centrifugal Compressors" (2014). *International Compressor Engineering Conference.* Paper 2340.
https://docs.lib.purdue.edu/icec/2340

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html

A Chiller Control Algorithm for Multiple Variable-speed Centrifugal Compressors

Piero CABALLERO[1]*, Juan ORDONEZ[1], Chiang SHIH[1], Turner THORNTON[2], Joost BRASZ[2]

[1]Florida State University, Department of Mechanical Engineering,
Tallahassee, Florida, USA
Contact Information (305-968-7463, Fax, plc11@my.fsu.edu)

[2]Danfoss Turbocor Compressor Inc.
Tallahassee, Florida, USA
Contact Information (U265950@danfoss.com)

* Corresponding Author

ABSTRACT

Capacity control of a chiller with multiple variable-speed centrifugal compressors running in parallel between a common evaporator and condenser under reduced load conditions can be achieved by speed variation and/or adjusting the number of compressors in operation. Quite often there is not a unique solution for a specific part-load condition in terms of the number of compressors required and the speed of each of these compressors. These different capacity control schemes can result in substantial differences in chiller performance under part-load conditions. The decision in selecting the optimum number of operational compressors to reach a desired part-load capacity depends not only on the required partial load but also on the required pressure ratio at that part-load condition. To illustrate the control dilemma, the paper will start with an example that shows the need for a different control strategy for two different chiller pressure ratios at equal part-load capacity. This difference in partial load control can be planned by observing the location of the efficiency islands on the variable-speed centrifugal compressor map. For a given refrigerant the efficiency of a compressor is a function of only two process variables: head (isentropic enthalpy rise) and volumetric flow rate. With that information a multiple compressor control algorithm has been developed that can estimate optimum chiller performance for any head/flow combination the chiller with multiple compressors might encounter. The improvement in chiller part-load performance using this algorithm can be substantial as will be illustrated in the paper for a two compressor chiller configuration.

1. INTRODUCTION

Oil management issues have limited in the past the use of multiple conventional centrifugal compressors on chillers sharing a common evaporator and condenser, despite the potential of those chillers for part-load performance improvement and cooling redundancy (limited cooling still available in case of one compressor failure). Variable-speed oil-free centrifugal compression technology has resulted in a proliferation of chillers with multiple compressors. The master controller of such chillers has to prescribe the required capacity of the individual compressors, but it also has to decide when to add or remove a second or third compressor. That decision is not obvious as will be shown for a two-compressor chiller. Figure 1 shows a simplified schematic of such a chiller with two compressors operating in parallel sharing a common evaporator and condenser. The two compressors, when both running, experience the same suction and discharge pressures dictated by the saturation temperatures of the common condenser and evaporator. When the capacity of the chiller can also be met by a single compressor it will again encounter the same suction and discharge pressures. Two temperature conditions (corresponding to high- and low-head compressor operation) are shown in this figure. Note that most of the temperature change when going from low to high head occurs in the condenser and that the evaporator temperatures are hardly affected. This is a reflection of the fact that leaving chilled water temperature always has to stay the same for building dehumidification or process cooling purposes while the condenser entering water temperature coming from the cooling tower changes with ambient conditions.

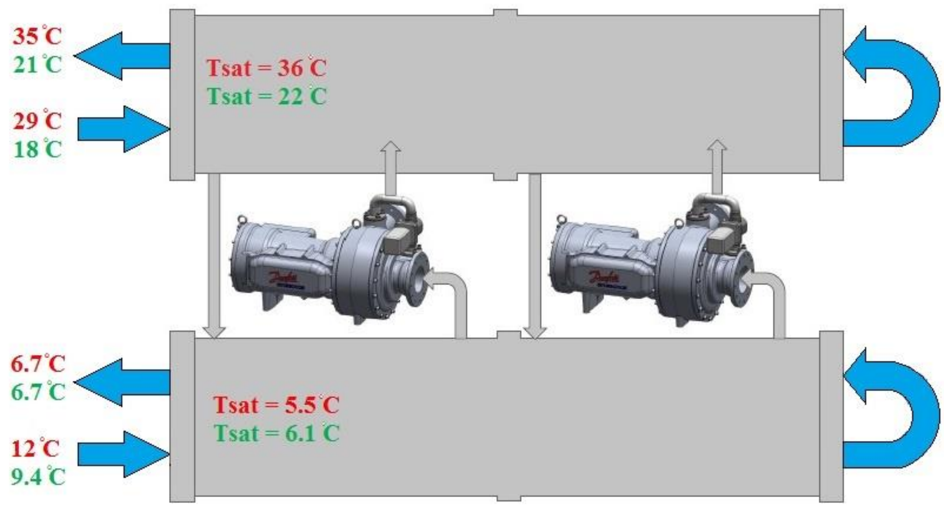

Figure 1: Two compressor chiller schematic with temperature in red showing the high lift condition and temperatures in green the low lift condition

What should be the control strategy when the chiller requires only half the capacity it can deliver with its two compressors:

1. Should both compressors run at 50% load at reduced speed?
2. Should one compressor be shut down and the other one run at full-load capacity?

In order to answer this question we have to compare the efficiency of the single compressor at 100% capacity against the compressor performance of two compressors running at 50% capacity. Centrifugal compressor efficiency is often given as a function of volumetric flow rate and head (isentropic enthalpy rise). The head H defined as the isentropic enthalpy rise can be determined from the evaporator and condenser saturation temperatures $H = h\left(P_{cond}, s_{evap}\right) - h(P_{evap,sat}, T_{in})$.

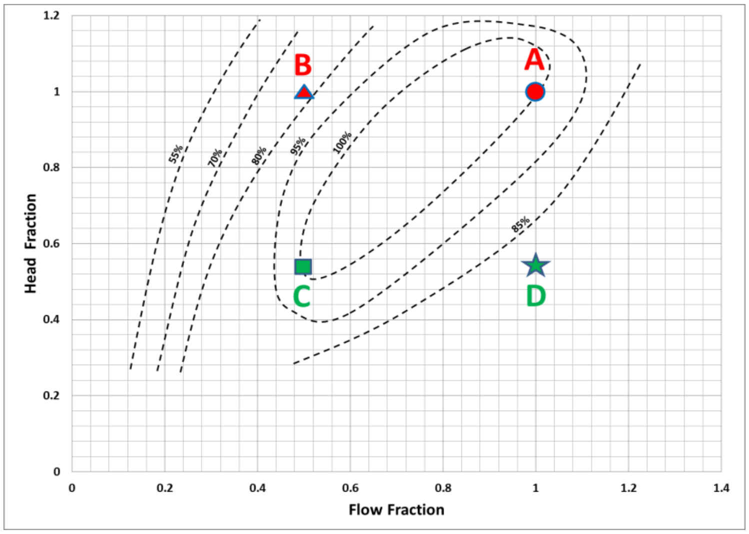

Figure 2: The non-dimensional aerodynamic performance map of a variable speed centrifugal compressor

Compressor capacity is controlled by the volumetric flow rate V̶ which follows from a refrigeration cycle analysis. Variable speed centrifugal compressor performance maps show graphically compressor efficiency as a function of volume flow as the variable on the horizontal axis and head on the vertical axis. Efficiency islands are obtained by connecting the head/flow points of equal efficiency. Figure 2 shows relative efficiencies as a function of flow fraction and head fraction. The meaning of the points A, B, C and D will be explained below. The variables on the compressor map (head, flow and efficiency) are displayed as percentages or fractions of the values at the full-load design point. The saturation temperatures shown in Figure 1 for high and low lift conditions result for an R134a compressor in isentropic enthalpy rises of 19.56 and 10.42 kJ/kg or normalized head fractions (HF) of 1.00 and 0.53, respectively. Under the simplifying assumption that chiller capacity is proportional to compressor normalized volumetric flow fraction (FF) and that motor and inverter efficiency are constant, the flow fraction/head fraction coordinates of the one compressor operating at high lift at 50% chiller duty (point A) are FF=1.0, HF=1.0. The map coordinates of both compressor operating at high lift at 50% chiller duty (point B) are FF=0.5, HF=1.0. Similarly, the flow fraction/head fraction coordinates of the one compressor operating at low lift at 50% chiller duty (point C) are FF=1.0, HF=0.53 The map coordinates of both compressor operating at high lift at 50% chiller duty (point D) are FF=0.5, HF=0.53%. Reading off the efficiencies at these four points it becomes clear that the high lift condition requires single compressor operation at 50% capacity while the low lift requirement requires dual compressor operation at reduced speed as shown in Figure 3 which contains the aerodynamic efficiency of a dual compressor system. By overlaying the compressor map of a single compressor with that of two compressors running in parallel and always selecting the higher efficiency option for given head and flow conditions a new compressor map can be constructed which contains a much larger area of high compression efficiency than can be obtained with a single compressor. This new map shows the advantage of multiple compressor chiller operation.

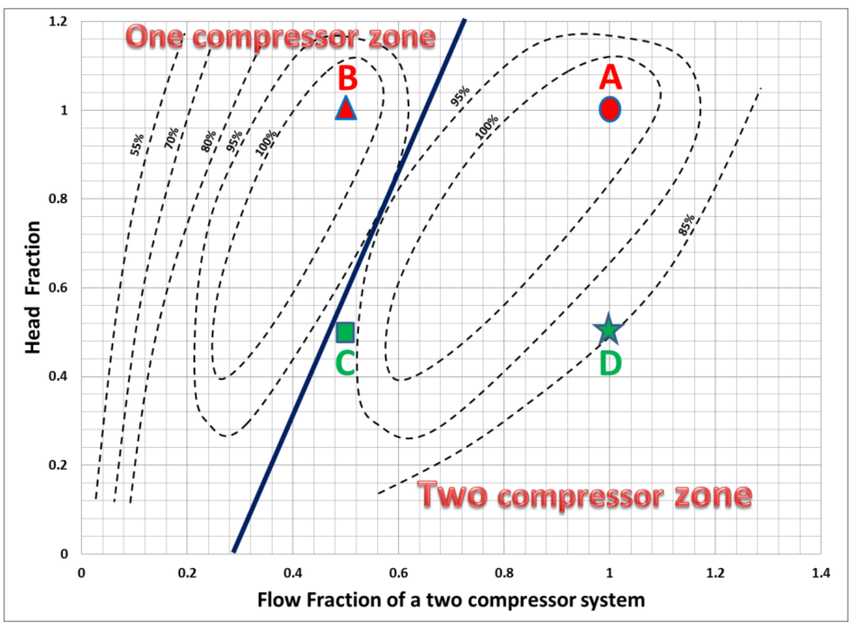

Figure 3: Aerodynamic efficiency islands of a two-compressor system

In order to generalize the example presented above by including the motor and drive efficiencies and allowing the combination of compressors of different capacities, it is necessary to expand compressor aero performance into chiller performance. For a multiple compressor chiller, this can be achieved by taking a look at the performance of each of the compressors and combining them into a single expression that can accurately depict chiller performance. This paper is organized as follows. In section 2 we observe how compressor and chiller performance can be modeled as well as the algorithm developed using a constrained minimization approach, which will estimate the optimum chiller performance for any head/flow combination. Section 3 will present the results of some sample cases in order to show the algorithm's dynamics as well as other relevant results. We conclude in section 4 where key points and an outline of the future work are presented aiming to improve the algorithm's capability to stage more than two compressors.

2. CHILLER PERFORMANCE AND MATHEMATICAL MODEL

Chiller performance depends highly on compressor efficiency, which is a function of two process variables: head (isentropic enthalpy rise) and volumetric flow rate as shown on the compressor map. For a constant pressure ratio, set by the operating temperatures (suction and discharge), the compressor's performance becomes a function of volumetric flow rate only, which better describes a real word scenario where the thermal load and ambient temperature remain fairly constant over significant periods of time. Isentropic efficiencies obtained from the maps can be further related to chiller performance [kW/ton], representing the amount of electric power a chiller requires per ton of refrigeration. Figure 4 shows chiller performance as a function of thermal load (proportional to volumetric flow rate) of a given compressor for different heads (variable condenser water temperature and constant evaporator water temperature). The information extracted from figure 4 can be used to calculate the total chiller performance when operating multiple compressors, provided that similar data is available for each compressor.

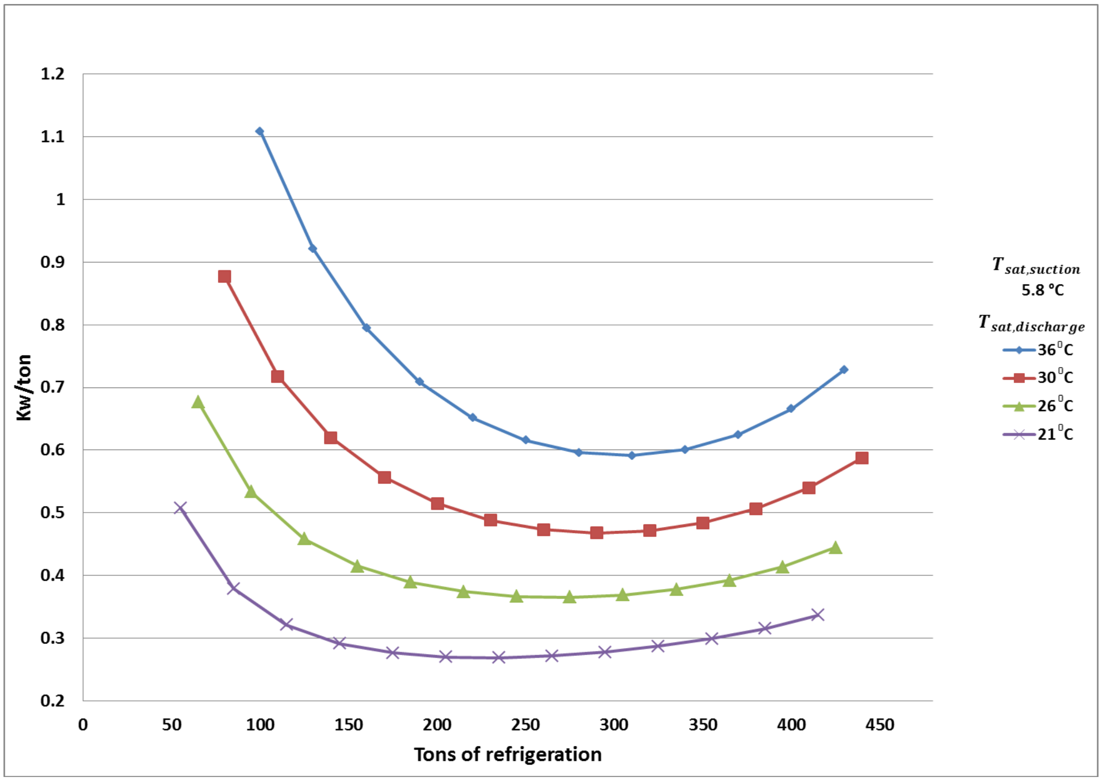

Figure 4: Chiller performance as a function of thermal load at different heads for a single compressor (overall chiller performance vs chiller flow fraction)

2.1 Chiller Performance Mathematical Characterization

As presented above, for a single compressor chiller and for constant head, the performance in [kW/ton] will be dependent on the load experienced by the compressor. This dependence takes into account motor and inverter efficiencies, something the aero map presented in figure 2 disregards. These curves can be accurately fitted using polynomials, and it was found that second order polynomials are sufficient to fit the curves with negligible error. Two functions can then be derived (1) and (2) corresponding to the efficiency of each of the compressors operating on a two compressor chiller. If the compressors operating on the chiller have the same design load, then equations (1) and (2) will be exactly the same. Sample cases with two identical compressors will be discussed in section 3.

$$E_1(x_1) = Ax_1{}^2 + Bx_1 + C \tag{1}$$

$$E_2(x_2) = Dx_2{}^2 + Ex_2 + F \tag{2}$$

2.2 Total Chiller Performance Characterization

When two compressors are present, total chiller efficiency becomes a function of each of the compressor's individual efficiency curves (Figure 4). A single third order multivariable function is created by combing functions (1) and (2) corresponding to the total chiller efficiency, the independent variables in this function are the part-loads for each compressor. Functions (3) and (4) show the function that characterizes chiller efficiency operating with two different compressors, while expression (5) is the criteria that the sum of each of the compressor's load will add up to the total chiller demand, criteria which needs to be met at all times.

$$E_{Total}(x_1, x_2) = \frac{[E_1(x_1)]x_1}{Q} + \frac{[E_2(x_2)]x_2}{Q} \tag{3}$$

$$E_{Total}(x_1, x_2) = \frac{Ax_1{}^3 + Bx_1{}^2 + Cx_1}{Q} + \frac{Dx_2{}^3 + Ex_2{}^2 + Fx_2}{Q} \tag{4}$$

$$x_1 + x_2 = Q \tag{5}$$

2.3 Constrained Optimization

If function (4) were to be visualized, a three dimensional surface would be observed on a 3D coordinate system, with the load of each compressor on two perpendicular axes and with the total chiller performance on the third axis. Function (3) or (4) can be optimized using conventional calculus methods; however the restriction imposed by (5) impedes us from using simple partial derivatives. A constrained optimization approach using a Lagrangian operator is used that takes into account the limitation (5). This method introduces a new independent variable λ and a new function G as shown by expression (6).

$$G(x_1, x_2, \lambda) = \lambda(Q - x_1 - x_2) \tag{6}$$

By combining expressions (4) and (6), we obtain a general expression for the total chiller performance as a function of both capacities, including the constraint that the loads should add up to the total thermal load Q.

$$L(x_1, x_2, \lambda) = \frac{Ax_1{}^3 + Bx_1{}^2 + Cx_1}{Q} + \frac{Dx_2{}^3 + Ex_2{}^2 + Fx_2}{Q} + \lambda(Q - x_1 - x_2) \tag{7}$$

Optimizing function L will yield values for x_1 *and* x_2 that will minimize the power usage.

$$\frac{\partial L}{\partial x_1} = \frac{3A}{Q}x_1{}^2 + \frac{2B}{Q}x_1 + \frac{C}{Q} - \lambda = 0 \tag{8}$$

$$\frac{\partial L}{\partial x_2} = \frac{3D}{Q}x_2{}^2 + \frac{2E}{Q}x_2 + \frac{F}{Q} - \lambda = 0 \tag{9}$$

$$\frac{\partial L}{\partial \lambda} = Q - x_1 - x_2 = 0 \tag{10}$$

Combining expressions (8) through (10) yields:

$$\frac{3(A-D)}{Q}x_2{}^2 - \frac{2(3AQ + B + E)}{Q}x_2 + \frac{(3AQ^2 + 2BQ + C - F)}{Q} = 0 \tag{11}$$

This quadratic expression can be solved to obtain the local minimum for function L; however equation (11) will not necessarily yield the absolute minimum. Equation (11) is of parabolic type and its local minimum will be absolute if and only if its second derivative is positive (concave up parabola), otherwise the parabola will be concave down and its absolute minimum will be at one of the boundaries. If the operating compressors are identical i.e. $A = D, B = E$ and $C = F$ equation (11) will be linear, simplifying the calculations to find x_1 *and* x_2.
Taking the second derivative of equations (8) and (9) with respect to x_1 *and* x_2 respectively yields:

$$\frac{\partial^2 L}{\partial x_2^2} = \frac{6D}{Q} x_2 + \frac{2E}{Q} \tag{12}$$

$$\frac{\partial^2 L}{\partial x_1^2} = \frac{6A}{Q} x_1 + \frac{2B}{Q} \tag{13}$$

When expressions (12) and (13) are positive as shown below, equation (11) will yield an absolute minimum; if they are negative the absolute minimum will be located at one the boundaries.

$$\frac{6D}{Q} x_2 + \frac{2E}{Q} > 0 \tag{14}$$

$$\frac{6A}{Q} x_1 + \frac{2B}{Q} > 0 \tag{15}$$

Manipulating these expressions and using the constraint imposed by (5), the values in which a local minimum corresponds to an absolute minimum can be found through (16) and (17) shown below.

$$-\frac{B}{3A} < x_1 < \left(Q + \frac{E}{3D}\right) \tag{16}$$

$$-\frac{E}{3D} < x_2 < \left(Q + \frac{B}{3A}\right) \tag{17}$$

3. RESULTS

3.1 Sample Cases

In order to test the validity of this method and the accuracy of chiller performance by means of equation (4), two different cases were calculated, and their corresponding efficiencies were compared against those obtained from the centrifugal refrigeration compressor rating method (CPR) described in [Brasz, 2010]. The first case consists of two compressors with identical design loads as outlined in table 1 and a total chiller demand of 700 tons. Case 1 will illustrate the staging of both compressors, i.e. how much load will be allocated to each compressor running under the operating conditions of table 1. Following the same operating conditions as table 1, case 2 illustrates how the algorithm behaves at smaller loads.

Table 1: Chiller operating conditions

$T_{sat,evap}$	5.56 ^{o}C
$T_{sat,cond}$	36.1 ^{o}C (High Head)
Refrigerant	R134a
Suction Superheat	3.6 ^{o}C
Condenser Subcooling	3.6 ^{o}C
Refrigerant	R134a
VTT1200 Design Load	350 tons
VTT1200 limits in tons	$110 \leq x_2 \leq 430$

The results of both cases are presented in table 2. The load allocation in case 1 was determined using equation (11). For the second case, the total thermal load did not meet the criteria set by (14) and (15) and the boundaries needed to be tested. The boundary points correspond to the compressors' limits as shown in table 1. Case 1 splits off chiller demand of 700 tons in half, allocating 350 tons to each compressor in order to optimize chiller performance. In case 2, due to the smaller demand, one of the compressors remains off while the other compressor takes the entire load.

Table 2: Cases results

Compressor	Compressor 1	Compressor 2
Case 1: 700 tons	350	350
Case 2: 300 tons	300	OFF

3.2 Effects of Head

Section 2.1 states that the error introduced by fitting the curves from figure 4 using a second order polynomial is sufficiently small. In order to better quantify this error, a sample case of a chiller operating with a 350 ton and a 110 ton compressor was analyzed (VTT1200 and TT350). Table 3 shows the divergence of using a simple quadratic vs. a 4th order polynomial to fit the curve and how heads affects this divergence. The first two rows show a chiller with a demand of 400 tons at high head ($T_{sat,cond}$ = 36.1 ^{o}C) and how this load is split between the two compressors, third and fourth rows show the same arrangement at a lower head ($T_{sat,cond}$ = 25 ^{o}C). At lower heads the difference between the errors using a 4th order polynomial versus a simple quadratic function increases, however not by much, still making the use of a second order polynomial valid and accurate enough. The results are also compared to those obtained through the CPR method.

Table 3: Second order vs. 4th order regression

Compressor	Simple quadratic [tons]	4th order polynomial [tons]	CPR results [tons]	Difference [tons]	Efficiency [kW/ton]
VTT1200	300	302	302	1.5	0.597
TT350	100	98	98	1.6	
VTT1200	301	299	299	2.1	0.377
TT350	99	101	101	2.1	

For a given demand, the split point that decides whether a chiller uses one or two compressor changes with head. Figure 5 shows the split point on a chiller operating with the same two 350-ton compressors shown in cases 1 and 2. For example, from case 1 we know that each 350-ton compressor takes 350 tons of the total 700 ton demand, this is because at 36.1 ^{o}C or HF =1 the split point occurs roughly at FF =0.56 which corresponds to 410 tons, any total demand above this point would split the load and use both compressors, any load below this point would only be allocated to one 350-ton compressor. At lower heads however, the split point occurs at lower tonnage, for example a total load of 350 tons (FF = 0.50) at a $T_{sat,cond}$ of 21.1 ^{o}C (HF = 0.53) will be more efficiently handled by two compressors than one, while at an $T_{sat,cond}$ of 36.1 ^{o}C (HF =1), one compressor can optimally take all the load.

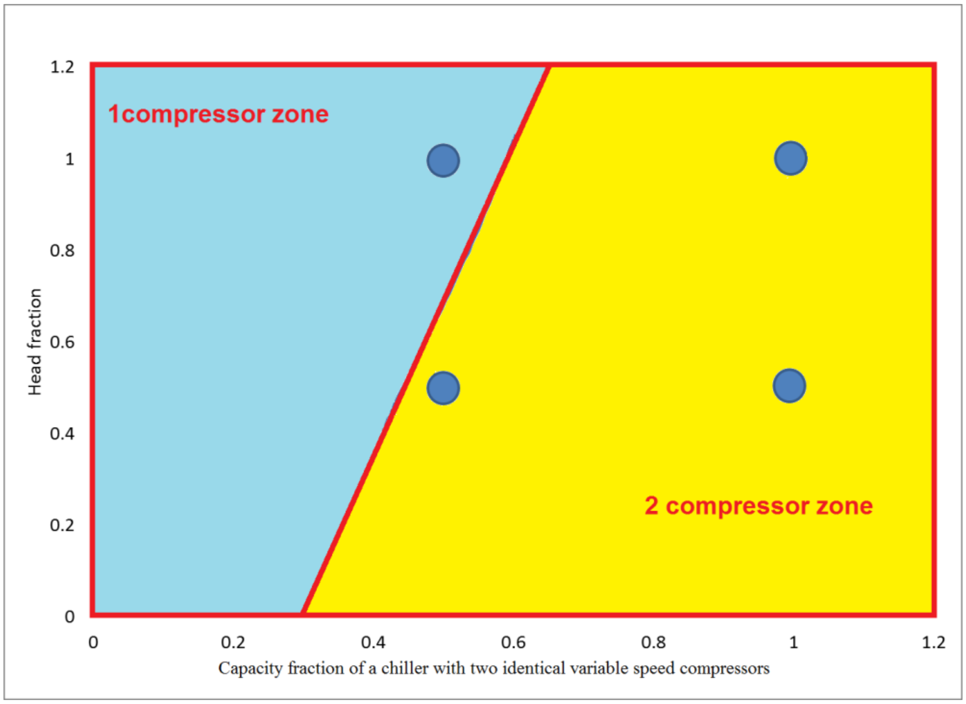

Figure 5: Two compressor chiller split point

4. CONCLUSIONS

- Multiple centrifugal compressor chiller operation has the potential of dramatically increasing centrifugal chiller performance.
- A single function to characterize multiple compressors chiller performance can be derived from each of the compressors efficiency maps.
- Optimum efficiency can be estimated for any head/flow combination a multiple compressor chiller may encounter.
- There is a clear relationship between the compressor's saturated discharge temperature (head) and the way the total thermal load is split between two compressors for a giver demand.
- This algorithm can be used as the basis of a control scheme in order to lower power consumption.
- The results presented can be generalized for more than two compressors.

NOMENCLATURE

Q	Total Chiller Refrigeration Demand	(ton)
A-F	Polynomial coefficients	(-)
x	Individual compressor load	(ton)

Subscript

1, 2	compressor 1, 2

REFERENCES

Brasz, Joost J., 2010, "A Proposed Centrifugal Refrigeration Compressor Rating Method" Proceedings of the 2010. International Compressor Engineering Conference at Purdue. Paper 2006.