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STUDY OF LUBRICANT CIRCULATION IN HVAC SYSTEMS

Volume I - Description of Technical Effort and Results

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ABSTRACT

The United States heating, ventilation and air-conditioning (HVAC) industry and refrigeration industry are rapidly and aggressively moving away from the currently used chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants to new, non-ozone depleting hydrofluorocarbon (HFC) refrigerants across a wide range of product lines. This massive undertaking requires careful assessment of the performance, operational capabilities, durability, and long-time reliability of HVAC products with these new HFC refrigerants and lubricants. Among the more important considerations in the change toward HFC refrigerants is the selection of lubricants that provide the same or improved characteristics relative to traditional mineral oils and alkylbenzene lubricants. Two of the more important characteristics of an acceptable lubricant, in addition to stability, cost and lubricity, are miscibility and solubility characteristics with the new HFC refrigerants.

In general, the lubricant circulation behavior of new HFC refrigerants with mineral oils (MO), polyolesters (POEs), and alkylbenzenes (AB) has not been well characterized for HVAC systems. Universal guidelines need to be developed for various compressor types, configurations, piping arrangements and other system features.

The purpose of this program was to conduct experimental and analytical efforts to determine lubricant circulation characteristics of new HFC/POE pairs and HFC/mineral oil pairs in a representative central residential HVAC system and to compare their behavior with the traditional HCFC-22/mineral oil (refrigerant/lubricant) pair.

A dynamic test facility was designed and built to conduct the experimental efforts. This facility provided a unique capability to visually and physically measure oil circulation rates, on-line, in operating systems. A unique on-line ultraviolet-based measurement device was used to obtain detailed data on the rate and level of lubricant oil circulated within the operating heat pump system.

The experimental and analytical data developed during the program are presented as a function of vapor velocity, refrigerant/lubricant viscosity, system features and equipment. Both visual observations and instrumentation were used to understand "worst case" oil circulation situations.

As a result of the comprehensive analytical and experimental efforts undertaken during this program, heat pump system operational regimes where poor oil management situations can occur have been defined and can be explained for an HFC-blend (R-407C) and HCPC-22 with a variety of miscible and immiscible lubricants (POEs and mineral oils). The operating regimes where poor oil return (i.e., near zero oil return) occurs can be related to low vapor velocity in selected portions of the systems, usually the vertical vapor lines. These vertical line locations, however, are different for the cooling and heating modes. These minimum flow velocities were determined by visual observations of oil return in an operating heat pump, as well as on-line oil concentration measurement instrumentation. The flow velocity levels predicted by the existing ASHRAE guidelines for good oil management were compared with the measured and observed conditions in this test sequence and some good correlations as well as areas for improvement were found to exist.

A surprising result of the program was the relative ease with which good oil return was obtained, even with an immiscible mineral oil and the R-407C blend. The potential for significant cost reductions due to the use of lower cost lubricants such as mineral oils rather than POEs should be noted. However, mineral oils, used as lubricants in combination with HFC refrigerants, must also be proven to provide good lubricity and thermal stability in modern high performance scroll compressors before their use can be widely adapted for industry. The lower limits of refrigerant/lubricant mixture viscosities were not severely tested in the programs, although normal operating regimes of heat pumps were explored. Ten to twenty degree lower temperatures and perhaps as low as -20°F saturation conditions should be tested to fully explore viscosity effects on oil management.

This report is presented in two volumes. Volume I contains a complete description of the program scope, objective, test results summary, conclusions, description of test facility and recommendations for future effort. Volume II contains all of the program test data essentially as taken from the laboratory dynamic test facility during the sequence of runs.

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During the formulation of the program tasks and interpretation of results and instrumentation needs and location, the team received substantial direction and assistance from the ARTI technical and program subgroup. The individuals are listed below.

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LIST OF ACRONYMS AND TERMS

AB	alkylbenzene
ACRC	Air-Conditioning and Refrigeration Center, University of Illinois
AC&R	Air-Conditioning and Refrigeration
AREP	Alternative Refrigerants Evaluation Program
ARI	Air-Conditioning and Refrigeration Institute
ARTI	Air-Conditioning and Refrigeration Technology Institute, Inc.
ASHRAE	American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.
CFC	chlorofluorocarbon
cSt	centistoke
DOE-A	Department of Energy - Standard Cooling Rating Condition
fpm	feet per minute
HCFC	hydrochlorofluorocarbon
HFC	hydrofluorocarbon
HVAC	Heating, Ventilation and Air-Conditioning
IEC	indoor environmental chamber
MCLR	Materials Compatibility and Lubricants Research
MO	mineral oil
POE	polyolester (lubricant)
PTAC	package terminal air conditioner
SUS	Saybolt Universal Seconds
UTC	United Technologies Corporation
UTRC	United Technologies Research Center
UV	ultraviolet
WRAC	window room air conditioner

PROGRAM OUTLINE

Program Objectives

The overall objectives of this analytical and experimental program were: 1) to determine the fundamental lubricant return parameters for HFC/POE mixtures so as to characterize how different refrigerant/lubricant mixtures affect the return of lubricant to the compressor, and 2) to determine fundamental lubricant return parameters for HFC/mineral oil mixtures to assess conditions where immiscible systems can provide sufficient lubricant return.

Program Approach and Scope

The overall program approach undertaken to meet this objective was: 1) to identify poor oil return scenarios and, therefore, the worst case oil return parameters for conventional residential HVAC systems using HCFC-22 and mineral oils, in terms of compressor, suction and exhaust line vapor velocity, and refrigerant viscosity requirements, 2) design and instrument a test apparatus that simulates such conditions, as well as those that might be achieved with HFC and POE mixtures and HFCs and mineral oils, 3) conduct tests with the range of baseline refrigerants and lubricant mixtures to provide experimental data, and 4) prepare, present and interpret the test data to provide an expanded understanding of the phenomena required for good oil circulation in split-system heat pump systems.

To convert this general approach into the program specifics, three major tasks were defined and pursued. These are described briefly below and in greater detail in the report body as Task 1, Task 2, and Task 3.

Task 1 consisted of gathering and reviewing available data. Within Task 1, an improved understanding of the lubricant circulation mechanism in refrigeration and HVAC systems was developed, along with the industry standard procedures and limitations on vapor velocity, viscosity and miscibility of the refrigerant/lubricant mixture. This included a review of the ASHRAE design guidelines, as well as discussions with system designers and manufacturers. Design parameters and features to be used as guidelines to construct a dynamic test facility were developed in this task. As part of this task, we also attempted to identify specific vapor velocity and viscosity limits that were to be tested in the experimental facilities with the various baseline and experimental refrigerant and lubricant mixtures, i.e., HCFC-22, R-407C and POEs and mineral oils. In the second part of this task, we reviewed the available data for refrigerants and lubricants and selected representative POE and mineral oil lubricants that covered the range of viscosity and miscibility anticipated in HVAC systems.

Task 2 consisted of the instrumentation design, fabrication and installation, and the experiment conduction in the dynamic test facility. The Task 1 data was used to design and build a dynamic test facility in which both good and poor oil management regimes could be experimentally explored so that broad on-line visual and instrumentation observations of oil return could be simulated. Instrumentation for proper observation and on-line oil measurements were screened, designed and installed. A series of tests with the various HCFC, HFC and POE and mineral oil combinations were prepared and extensive data gathered.

Task 3 consisted of data collection, analysis, results, review and formulation of the program conclusions. More than fifty test runs were actually conducted in the dynamic test facility, reviewed and assessed for consistency. Broad as well as very specific conclusions and results were recorded. Much of the data is based on visual observations of poor oil return scenarios. Finally, broad guidelines as to acceptable flow velocities for "good" and "poor" oil management are provided for use by HVAC industry designers, manufacturers and installers.

INTRODUCTION

Under the provisions of the Montreal Protocol, CFCs and HCFCs, the backbone refrigerants for the air conditioning and refrigeration industry products for the last five decades, are being phased out or their usage is being capped. Because of their deleterious effect on the global ozone layer, this phase out and production capping is starting in 1996.

The HVAC and refrigeration industry leaders are rapidly and aggressively moving away from the currently used CFC and HCFC refrigerants to new, non-ozone depleting HFC refrigerants across their extensive product lines. This massive undertaking requires careful assessment of the performance, operational capabilities, durability, and longtime reliability of their products. Among the more important considerations in the change towards HFC refrigerants is the selection of lubricants that provide the same or improved characteristics relative to the traditional mineral oils and alkylbenzene lubricants. The new synthetic lubricants, primarily polyolesters (POEs) are being chosen for use with HFCs because of their favorable stability, lubricity and miscibility characteristics. Because the POE lubricants are miscible with the new HFC refrigerants over a wide range of temperatures, they could provide similar lubricant characteristics to the old, familiar CFC and HCFC systems, which largely used mineral oils. However, each type of HVAC system utilizes different compressor types, configurations, piping arrangements and other features so that universal guidelines are difficult to apply.

Among the new HFC refrigerants that are being evaluated, are a substantial number of refrigerant blends as substitutes for HCFC-22 in residential building air conditioning and heat pump applications. Analytical predictions and system tests have indicated that there are certain performance, capacity, operational advantages and potential size and weight improvements with the use of HFC refrigerants and blends. However, these blends can introduce significant service and operational issues, due to fractionation of the blends. Even in those blends which do not experience fractionation, a serious concern to the HVAC industry are other potential operational and reliability issues. These issues can take the form of long-term reliability in typical system installations. One such concern is associated with the oil return characteristics and potentially different mechanisms that might be observed with new HFC refrigerants and appropriate lubricants such as polyolesters (POEs). The POEs appear to have the correct lubricity and thermal and chemical stability to meet the challenges of today's high performance compressors, but what is largely unknown are the proper oil management techniques to ensure that oil return to the compressor is achieved and that the compressors, and thus overall system, will function reliably for at least fifteen years of operation.

A second issue is the potential use of low cost lubricants, such as mineral oils, even though they are immiscible with most HFC refrigerants. Mutual solubilities of the new HFC/POE pairs are very different from those of the CFC/HCFC refrigerant/mineral oil pairs they are intended to replace. Similarly, interfacial surface tension properties of conventional refrigerant/lubricant pairs show differences when compared with the newer HFC/POE combinations. In actual operating systems, there are still many unanswered questions about the behavioral differences between the old and new refrigerant/lubricant pairs. Thus, the subject program was aimed at studies designed to obtain fundamental data to better understand the oil return parameters for the new versus the old refrigerant/lubricant pairs. The data generated in this study will aid in the assessment of how the differences in the physical properties affect system reliability and oil return mechanisms.

Because of the high cost and operational considerations associated with the POEs, the component and system manufacturers are interested in the possibility of using immiscible mixtures of HFCs and mineral oil or alkylbenzene lubricants in air conditioning/heating and refrigeration systems. A major impediment to this strategy is the uncertainty of the return to the compressor of the immiscible lubricant which still must have a high enough viscosity to effectively lubricate the compressor. Therefore, it is imperative that the fluid properties of immiscible refrigerant/lubricant mixtures and their impact on the system operation be well understood. The data generated in this study will also help determine whether immiscible refrigerant/lubricant mixtures might be successfully applied with HFCs.

United Technologies Research Center (UTRC) has recognized these problems and has taken specific actions to insure that lubricant circulation in systems can be measured and that there are techniques in place to improve the understanding of adequate lubricant return with miscible HFC/POEs and possibly immiscible mixtures of HFCs with mineral oils and alkylbenzene lubricants in air-conditioning and refrigeration (AC&R) systems. In particular, United Technologies Corporation (UTC) has developed and assembled the following background material for application to the present oil circulation study:

1) Detailed computer models have been developed and verified by experimental results that predict the solubilities and fractionation effects of various HFCs mixtures and POE lubricants. These models and data were developed in large part under an ARTI-sponsored UTRC program (MCLR Project No. 660-52300) and under corporate sponsorships;
2) A comprehensive model has been developed under ARTI sponsorship, which describes the concentration of lubricant and HFC blends in typical heat pump systems during startup, shutdown, and various modes of operation;
3) Actual oil circulation rate measurements were carried out at Carrier and included as part of the performance tests and data provided by Carrier under the Alternative Refrigerants Evaluation Program (AREP) Soft Optimization Program efforts on a split system 5-ton heat pump;
4) Accumulated knowledge and detailed installation information were provided by Carrier Residential personnel based on their installation of hundreds of thousands of heat pumps and air conditioners each year;
5) Several new techniques for measuring, *in situ*, oil concentration rates with HFCs immiscible/miscible lubricants were developed and tested at UTRC.

The subject program was aimed at understanding refrigerant/lubricant circulation issues, developing test data and approximate models that can predict operating regimes where good oil management can be assured. A dynamic test facility was constructed and used to examine oil return under varying system operating conditions. The development of industry guidelines for system reliability in using the new refrigerant blends was a goal of this program. To validate the guidelines, techniques and predictions, this dynamic test facility was used to obtain data to compare to the analytical predictions.

TASK 1 - GATHER AND REVIEW AVAILABLE DATA

The objective of this task was to gain a more thorough understanding of the lubricant circulation mechanisms in refrigeration systems, with particular emphasis on certain key factors. These factors are the role of refrigerant vapor mass flux, as well as refrigerant/oil miscibility and solubility and resulting mixture properties such as viscosity and surface tension. Two activities were undertaken: 1) a general review of industry practices and design approaches in HVAC systems, especially for split-system residential air conditioners and heat pumps, and 2) development of parameter limits for viscosity, flow velocity and possibly miscibility, to be investigated for the refrigerant/lubricant pairs in the dynamic test apparatus to be built and used to obtain experimental verification of oil management.

In the sections which follow, background information of the industry practices and problems are provided, followed by a discussion of the specific flow velocity and viscosity parameters to be evaluated. How the refrigerant/lubricant mixtures to be tested fit within these guidelines and a description of the specific data is also presented in this section.

Background

In a typical refrigeration system for comfort cooling and heating, such as the heat pump shown in Fig. 1, the lubricant is typically introduced to the compressor suction port from mist created in the sump region and interaction of the refrigerant with the oil return flow from the bearings. The lubricant is necessary for separating and sealing the compression surfaces and valves of most positive displacement type compressors. Additional lubricant may be directly introduced to the compressor, if the suction mist is insufficient. The lubricant is then discharged with the hot refrigerant gas and, in low-side compressor, circulated throughout the system, such as shown in Fig. 1.1a.

Miscible Lubricants

Miscible lubricants dissolve in the condensing refrigerant (in the outdoor section when the heat pump is functioning in the cooling mode) and are readily carried to the expansion valve. The lubricant separated in the evaporator is returned to the compressor by gravity and drag force from the refrigerant gas. The gas velocity necessary for recommended or good oil return (typically 1350 to 1500 fpm for up flow) is achieved by sizing return lines for these flow conditions. Therefore, a critical or worst oil return condition is likely to be at the evaporator exit (the indoor section in the cooling mode), where the vapor refrigerant line is located that feeds into the accumulator (see Fig. 1.1a). The range of these flow parameters will be discussed in greater detail below.

During the heating mode in a heat pump, the worst case oil circulation condition will tend to be found in the vapor line leaving the compressor that feeds the indoor section, where the evaporator serves as the system condenser. This is shown in Fig. 1.1b. Maximum height parameters are sometimes used to indicated worst case oil circulation conditions. However, it is not uncommon for split-system installations to have vertical separations of 30 to 100 feet and still provide adequate oil circulation. Flow velocity, rather than maximum height, may therefore be the critical design parameter.

Refrigerants and Immiscible Lubricants

Refrigerants and immiscible lubricants will circulate in much the same fashion, except that refrigerant and lubricant stratification and therefore pooling, with oil and refrigerant forming different phases, can exist in the liquid regions of the circuit. However, unlike the situation with miscible refrigerant/oil mixtures, immiscible lubricants could also have a higher effective viscosity since minimum (or zero) thinning will occur due to refrigerant dilution of the lubricant. It is well known that even a small percentage (0 to 10%) of refrigerant can substantially reduce lubricant viscosity. Therefore, one could postulate poorer oil return during cold evaporator operation as a consequence. However, recent experience, as part of the AREP efforts, with HFC-407C and alkylbenzene in a split system heat pump revealed no lubricant return problems during a possible worst case scenario that included elevating the outdoor unit 17 feet above the indoor coil (Ref. 1).

Lubricant Particle Mechanism Formulation

From experience and systems results, it is apparent that refrigerant gas dynamic heat ($1/2\rho V^2$) and lubricant particle size are keys to lubricant transport in the gas phase regions of a system. Considering the relatively low velocities that are adequate for lubricant transport, mechanisms for relatively small lubricant particle size generation must be present in both the compressor and evaporator. The Weber number, or ratio of inertial forces to surface tension forces, correlates well with particle size generation. High pressure, temperature, and gas velocity in the compressor generates high inertial forces and, hence, small particles. Since the drag of the particle is proportional to the diameter squared and the mass is proportional to the diameter cubed, these smaller particles require less dynamic head for transport. In the evaporator, it is the boiling action of the refrigerant that generates inertial forces adequate for small (50 µm) diameter particle creation. In addition, surface tension is reduced, due to lubricant miscibility. At very low temperatures and pressures, the lubricant may become immiscible and its viscosity high. This combination can lead to lubricant accumulation in the evaporator and/or suction line and can be a serious problem preventing oil return to the compressor sump.

In the two-phase and liquid regions, miscible lubricants dissolve into the refrigerant and are readily transported with the bulk flow. Immiscible lubricants may form an emulsion if the surface tension is low and adequate inertial forces exist. In this case, lubricant circulation may be satisfactory. However, at low temperatures, the system mass flow is minimum and lubricant viscosity high. This may lead to lubricant accumulation in low velocity regions.

Refrigerant blends, such as R-407C, introduce the additional complication of composition shift due to potential fractionation in the two phase components. Such composition shifting can influence the lubricant solubility and, therefore, miscibility. UTRC has developed (under contract to ARTI) static and dynamic system models for several HFC blends and lubricant combinations to estimate these effects. These models have been modified for R-407C and selected POE lubricants to make accurate estimates of lubricant miscibility during system operation. The changes in viscosity due to these fractionation effects were not considered significant for the scope of this study. However, a particularly useful feature of the dynamic model is the ability to track the lubricant and refrigerant constituents throughout the system, if sufficient data are available.

In order to understand these fundamental lubricant return parameters for HFC/POE and HFC/mineral oil mixtures, as well as baseline state-of-the-art HCFC-22 and mineral oil mixtures, it is critical to obtain physical data on candidate refrigerant/lubricant pairs. In addition, research conducted on lubricant return mechanisms needs to be identified. A brief review of the available data is described in Subtask 1A. In addition, the current heat pump system configurations and operating conditions must be known to correctly identify situations which could lead to improper or poor lubricant return. System manufacturers, distributors and installers need to be surveyed for their service practices and problems to adequately understand these situations, as is described in Subtask 1A below.

Subtask 1A - Oil Management Industry Service Practices and Problems Survey

Industry Guidelines

Oil management in refrigeration systems is thoroughly discussed in the ASHRAE refrigeration handbook, Ref. 2, as well as numerous other sources. All refrigerant compressors circulate some amount of oil throughout the system. Oil separators are used where excessive oil can affect system performance or where poor oil return conditions exist such as in low temperature refrigeration systems, and it is essential that oil be guaranteed to return in the system.

For split-system heat pumps, which are the focus of this study, separators are seldom used. Rather, the various system components are designed to assure proper oil flow through the system and return to the compressor oil sump. The literature review indicates that hot compressor discharge gas will transport the fine oil mist in the refrigerant vapor through hot gas risers if **adequate mass flux or line velocity is maintained**.

For air-cooled condensers, this minimum velocity must be maintained in condensing tubes for transporting both liquid refrigerant and oil. Miscible oil is typically specified to minimize oil fouling of the tubes and to assure good transport through the liquid lines. However, immiscible oil will also likely be transported through the condenser, since temperatures are relatively high and the resultant oil viscosity low. The exception to this will be for flooded areas where the oil and refrigerant can separate and pool. Here, care must be taken to assure flow of both liquids.

More serious problems can be expected in the evaporator, since the refrigerant evaporates and the liquid phase becomes enriched in oil. Oil viscosities are selected to assure oil transport at minimum anticipated velocities and temperatures. For the case of oil return up suction risers, line diameters are specified to assure that oil flow is maintained under conditions where minimum vapor velocity levels are achieved. Accumulators, receivers, and driers must also be designed to flow the specified oil under worst case circumstances, such as low temperature conditions in these components.

ASHRAE Procedure

ASHRAE publishes tables of minimum refrigeration capacity for oil entrainment up suction and hot gas risers. The experimental and theoretical basis for these tables comes from research at Carnegie-Mellon University (Jacobs *et al*. 1976, Ref. 3). The minimum mass flux, and hence velocity, was found to be proportional to gas density and line diameter through the following relationship:

$$\text{Minimum Mass Flux} = 0.7225[\rho_g g_c D\,(\rho_f - \rho_g)]^{½}$$

where

ρ_g = gas density, lb/ft^3
g_c = 32.2 ft/sec^2
D = pipe diameter, ft
ρ_f = liquid mixture density (oil and refrigerant), lb/ft^3

Viscosity is not directly included in the above expression; however, the equation is stated to be only valid for viscosities lower than 3000 SUS, or about 650 centistokes (cSt). The oils that were considered for this study are all generally less viscous than this limiting value (see Table 1.1), if minimum temperatures in selected components are maintained and some small refrigerant dilution of the lubricant occurs.

Table 1.1 Viscosity - Temperature Characteristics of Lubricants

Lubricant	Type	Temperature (°C)	Viscosity (cSt)
Witco Suniso 1 GS	Mineral oil	-20.	---
		40.	12.0
		100.	---
Witco Suniso 3GS	Mineral oil	-20.	4000.0
		40.	33.0
		100.	4.5
ICI Emkarate RL32S	POE	-20.	2310.0
		40.	32.0
		100.	5.6
ICI Emkarate RL68S	POE	-20.	7356.0
		40.	74.1
		100.	10.1
Castrol SW32	POE	-20.	2000.0
		40.	31.0
		100.	5.9
Castrol SW68	POE	-20.	10,000.0
		40.	68.0
		100.	9.0

Solving the above equation for a typical range of cooling and heating conditions results in the minimum flow velocities shown in Table 1.2 for HCFC-22 and R-407C, with representative POE and mineral oil lubricant viscosities.

Typical equivalent minimum flow velocities are indicated for each refrigerant and cover a range from roughly 369 to 416 fpm for HCFC-22 and a range from 385 to 435 fpm for R-407C in cooling (therefore, in the suction lines to the compressor). In the heating mode, the minimum flow velocities range from about 200 fpm up to almost 260 fpm at representative saturation temperatures feeding the hot gas risers coming out of the compressor.

The method by which the velocity was varied in the actual test system was by means of a manifold of 3/4 inch vapor lines, as discussed in Task 2. Based on the typical refrigerant flow rates that can be achieved in representative, available, scroll-compressors for 2.5 to 3.0 ton cooling systems, it was then possible to calculate the number of 3/4 in. diameter lines needed to achieve the minimum mass flow calculated from the Ref. 1 or 3 guidelines. The line size of 3/4 in. was selected as a compromise between dynamic facility cost and ability to obtain more discrete flow velocity data. Since the project purpose was attempting to test the guideline limits, as well as uncover worst case and poor oil management scenarios, it was desired to be able to go somewhat below the guideline minimal limits.

The Table 1.2 data indicate that guideline minimum flow velocities and below could be achieved with three (3) to four (4) lines in cooling at the representative saturation temperatures for the two refrigerants; while in the heating mode, only two or three, 3/4 in. lines would reduce the flow below the recommended limits, and the potential for poor oil management or zero oil return would be observed.

In addition to the limits shown by the number of 3/4 in. lines to be opened during testing, it was further decided that worst case oil return scenarios could be explored by reducing the saturation temperature in heating, i.e., by lowering the evaporator fan coil airflow or by operating the test system at lower evaporator temperature levels in the cooling mode.

Table 1.2 Approximate Limiting Minimum Vapor Velocities for Marginal Oil Return Representative Refrigerants in Typical 2.5 to 3 ton Capacity Heat Pump Split System

Data for Pure HCFC-22 and R-407C; Based on Ref. 3 Guidelines

Refrigerant	**Velocity in Cooling Mode, ft/min**		**Velocity in Heating Mode, ft/min**	
	Location - Suction Inlet Line to Compressor		**Location - Discharge Line Compressor to Indoor Coil**	
	Temperature, °F		**Temperature, °F**	
	45°F	**32°F**	**120°F**	**90°F**
HCFC-22	369	416	200	255
No. of 3/4 in. diameter Lines to Achieve Velocity Requirements Minimum Flow and Lower	3 to 4	3 to 4	1	2
R-407C	385	435	202	256
No. of 3/4 in. diameter Lines to Achieve Velocity Requirements Minimum Flow and Lower	3 to 4	3 to 4	1	2

Guidelines (Ref. 3) Apply for Maximum Lubricant Viscosity = 650 cSt or 3000 SUS units.

Industry Personnel Interviews

UTRC interviewed personnel at both Carrier Residential Products Group and Carrier Commercial Unitary Group to obtain information on oil circulation problems that their dealers, installers and distributors had experienced and to help develop "worst case" scenarios for typical split-system heat pumps to be evaluated in this program. Problem areas tended to be in the two phase regions where the gas phase velocity was low and the liquid phase was mostly or all oil. These regions are at the entrance to the condenser, exit of the evaporator and in the accumulator.

The equipment manufacturer has control over some of these locations, so system geometry can be made specific to prevent marginal oil return problems.

Contractor installations tend to be the greater problem. The manufacturer often has a set of guidelines to ensure proper system installation in long line applications. In fact, installations where the height between the outdoor and indoor units was greater than 100 ft were not unheard of. However, long vapor lines may be of incorrect diameter or pitch, which results in liquid oil accumulating in low vapor velocity regions. Rather than identify a particular set of geometries that represent "worst case" oil return, the manufacturer believes that quantifying the velocity required for oil transport up vertical lines for various temperature, oil concentration, and liquid viscosity levels would be more useful for industry information and could be used to warn of "worst case" scenario installations and improper installation procedures.

The Table 1.2 data were reviewed by system manufacturer personnel who are concerned with poor installation practices and oil return issues and were generally accepted to be reasonably sound.

Subtask 1B - Miscibility and Viscosity of Refrigerant/Lubricant Pairs

The objectives of this subtask were to develop the miscibility and viscosity data and to select the proper set of refrigerant/lubricant pairs to provide the necessary range of data mixture properties to draw conclusions on refrigerant/lubricant oil management and circulation parameters.

The primary considerations in choosing a lubricant are its chemical compatibility with the refrigerant type and the required viscosity for the service application. In the case of refrigerant blends, a new problem arises since the individual refrigerant components may exhibit different solubilities in the lubricant.[1] Different component solubilities may give rise to fractionation effects in the system which differ from the vapor-liquid equilibrium conditions in the absence of a lubricant. These differential solubility effects may also result in conditions where limited refrigerant/lubricant miscibility is observed. Operation with lubricants that exhibit limited miscibility and with lubricants that exhibit high viscosity could potentially adversely impact both cycle operating parameters and the overall system durability and performance.

The circulation behavior of the refrigerant/lubricant mixture could depend in part on the thermophysical property data of the mixture. An extensive property data base on single refrigerant/lubricant mixtures has been developed at UTRC with input from the major fluorocarbon chemical producers and lubricant formulators. Recently data have become available on the properties of several refrigerant blend/lubricant mixtures, including the R-410A/POE, R-404A/POE and R-407C/POE systems. The R-410A and R-404A blends both exhibit an upper miscibility dome (upper consolute temperature condition) in mixtures with many of the commonly used POEs. With certain POE formulations, this immiscibility results in two liquid phase formation in the condenser, in contrast to formation of two liquid phases in the evaporator where the lower critical solution temperature often is reached. A careful study of the solubility of R-407C with several POEs indicates that the upper critical solution temperature is

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[1] This problem is one subject that was studied at UTRC under the ARTI program "Investigation into Fractionation of Refrigerant Blends": (ARTI MCLR Project No. 660-52300).

above 60°C for most POE formulations. Thus only low temperature immiscibility was examined in choosing the refrigerant/lubricant pairs for this study.

For this program, the baseline for comparison of lubricant circulation was the refrigerant/lubricant pair, R-22/mineral oil, which is widely used in industry. An equally acceptable combination would be R-22/alkylbenzene since commercial equipment has been successfully operated with both systems. Extensive data on the solubility and viscosity of R-22 in either mineral oil or alkylbenzene are available in ASHRAE tabulations or from manufacturers. Two viscosity grades of mineral oil were selected for study: Suniso grade 3GS (150 SUS), which matches an ISO 32 grade POE in viscosity and Suniso grade 1GS (65 SUS), which mimics the dilution effect of viscosity with a POE ISO 32 grade/refrigerant mixture. The kinematic viscosity and solubility characteristics of R-22/3GS mixtures are shown in Figs.l.2 and 1.3, respectively. The miscibility, data are shown in Fig. 1.4, which indicate a lower consolute temperature of ~ -4°C at 20% mass fraction of mineral oil. The Fig. 1.2 viscosity data should be examined at the typical operating temperatures in Table 1.2 during representative cooling and heating conditions. Furthermore, extreme low temperature operation at 0°F or below, representative of refrigeration applications, should be examined before general oil management conclusions are stated for refrigerant/lubricant pairs.

For comparison against the baseline, the R-407C/Suniso 3GS and R-407C/Suniso 1GS pairs were studied to examine the effects of refrigerant/lubricant immiscibility. These HFC/MO systems should exhibit nearly total immiscibility patterns in the refrigerant rich region, <20% MO. Data are available on the limited miscibility of R-407C in Zerol 150, an ISO 32 alkylbenzene lubricant, (Ref. 4), as illustrated in Fig. 1.5. The R-407C/MO systems should exhibit a similar miscibility pattern. The miscibility of Emkarate RL68S and RL32S are also shown on Fig. 1.5 for comparison.

The circulation behavior of these two baseline refrigerant/lubricant pairs, one partially miscible (R-22/MO) and one nearly totally immiscible (R-407C/MO), was then to be compared with four R-407C/POE lubricant pairs. Preliminary data, taken at Carrier Corporation, on the refrigerant/lubricant circulation patterns in a split system heat pump suggest that viscosity effects should be studied using POEs which exhibit significantly different ISO ratings. Several .low viscosity (ISO 32) R-407C/POE mixtures have well characterized thermophysical properties. A search for data in the high viscosity region (>ISO 100) was less fruitful, especially for R-407C/POE pairs that have high miscibility. ISO 68 POEs were finally selected since solubility data were available with R-407C.

An extensive analysis of several thermophysical property data sources (Refs. 5-12) indicated that the following refrigerant/lubricant pairs would separately test the effects of lubricant viscosity and lubricant miscibility with R-407C:

R-407C/ICI RL32S	:	low miscibility, low viscosity
R-407C/ICI RL68S	:	low miscibility, high viscosity
R-407C/Castrol SW32	:	high miscibility, low viscosity
R-407C/Castrol SW68	:	high miscibility, high viscosity

Miscibility data for R-407C/lubricant mixtures were obtained from Castrol and ICI. Data for the R-22/MO pairs were obtained from Witco. These data are summarized in Table 1.3.

Table 1.3 Refrigerant/Lubricant Miscibility Data

Refrigerant/Lubricant Pair	**T(lower consolute)**
R-22/Suniso 3GS	-4°C at 20 wt% oil
R-22/Suniso 1GS	-10°C at 20 wt% oil
R-407C/Castrol SW32 (low visc., high misc.)	< -50°C at 10 wt% oil
R-407C/ICI RL32S (low visc., low misc.)	-5°C at 10 wt% oil
R-407C/Castrol SW68 (high visc., high misc.)	-30°C at 10 wt% oil
R-407C/ICI RL68S (high visc., low misc.)	-3°C at 10 wt% oil

The kinematic viscosity of R-407C/Castrol SW32, R-407C/Emkarate RL32S, R-407C/Castrol SW68, and R-407C/Emkarate RL68S, are shown in Figs. 1.6 - 1.9, respectively. The viscosity reduction upon dilution with R-407C is somewhat greater with the ICI formulations than with the Castrol formulations. The viscosity data for the various selected baseline and alternative refrigerant/lubricant pairs does span the range of applicable viscosities recommended in Ref. 3 and followed by the industry for current oil management practices.

The R-407C/Castrol SW68 and R-407C/ICI RL68S mixtures could exceed the (Ref. 1) 650 centistoke restrictions at high lubricant, low temperature conditions. Therefore, the compressor pump should be monitored carefully to identify if any oil management and refrigerant/lubricant immiscibility and separation problems occur with the refrigerant/oil mixtures. R-22/MO mixtures at similar low temperature, high oil content conditions, should be monitored especially with an eye toward the lower consolute temperature behavior of this mixture, as shown in Fig. 1.4, when a high percentage of lubricant (> 5% oil) is encountered.

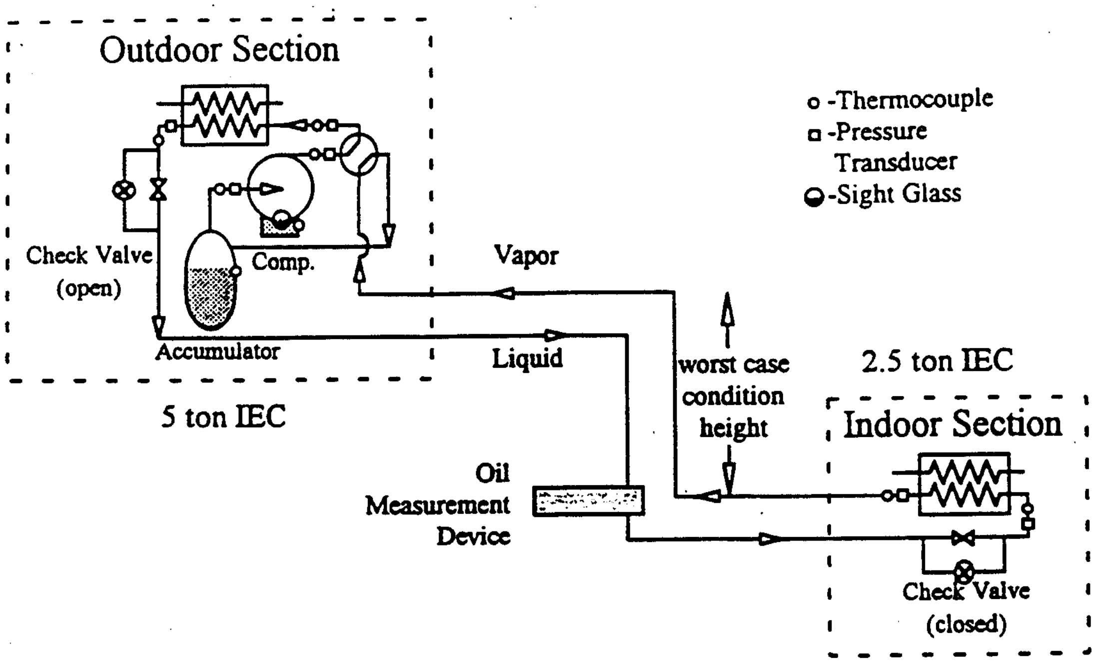


Figure 1.1a. Heat Pump Layout (Worst Case Cooling Model)

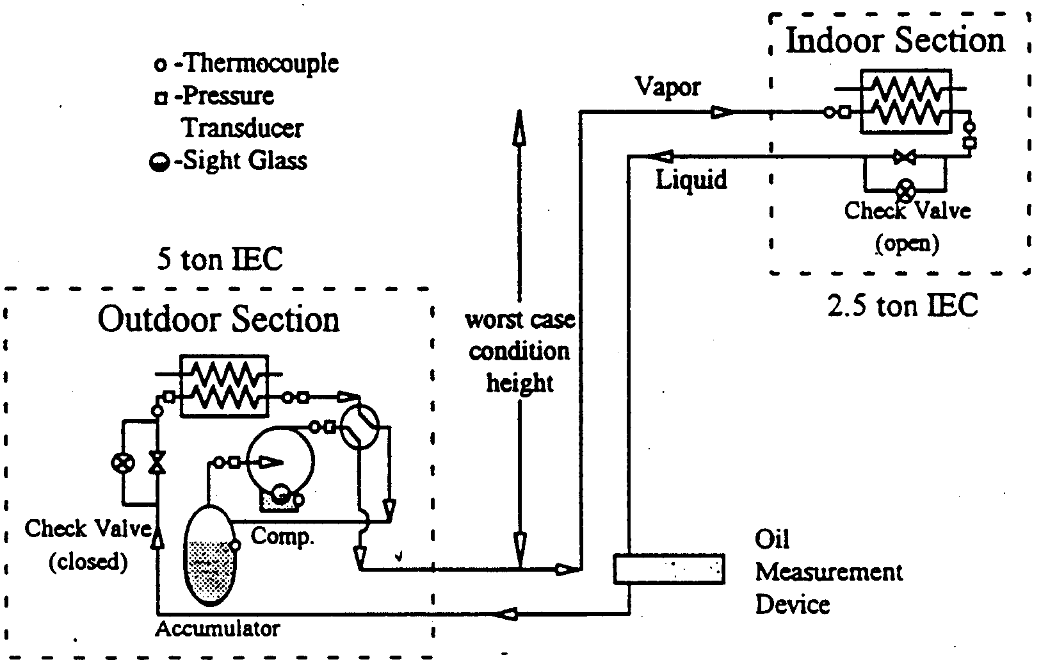


Figure 1.1b. Heat Pump Layout (Worst Case Heating Model)

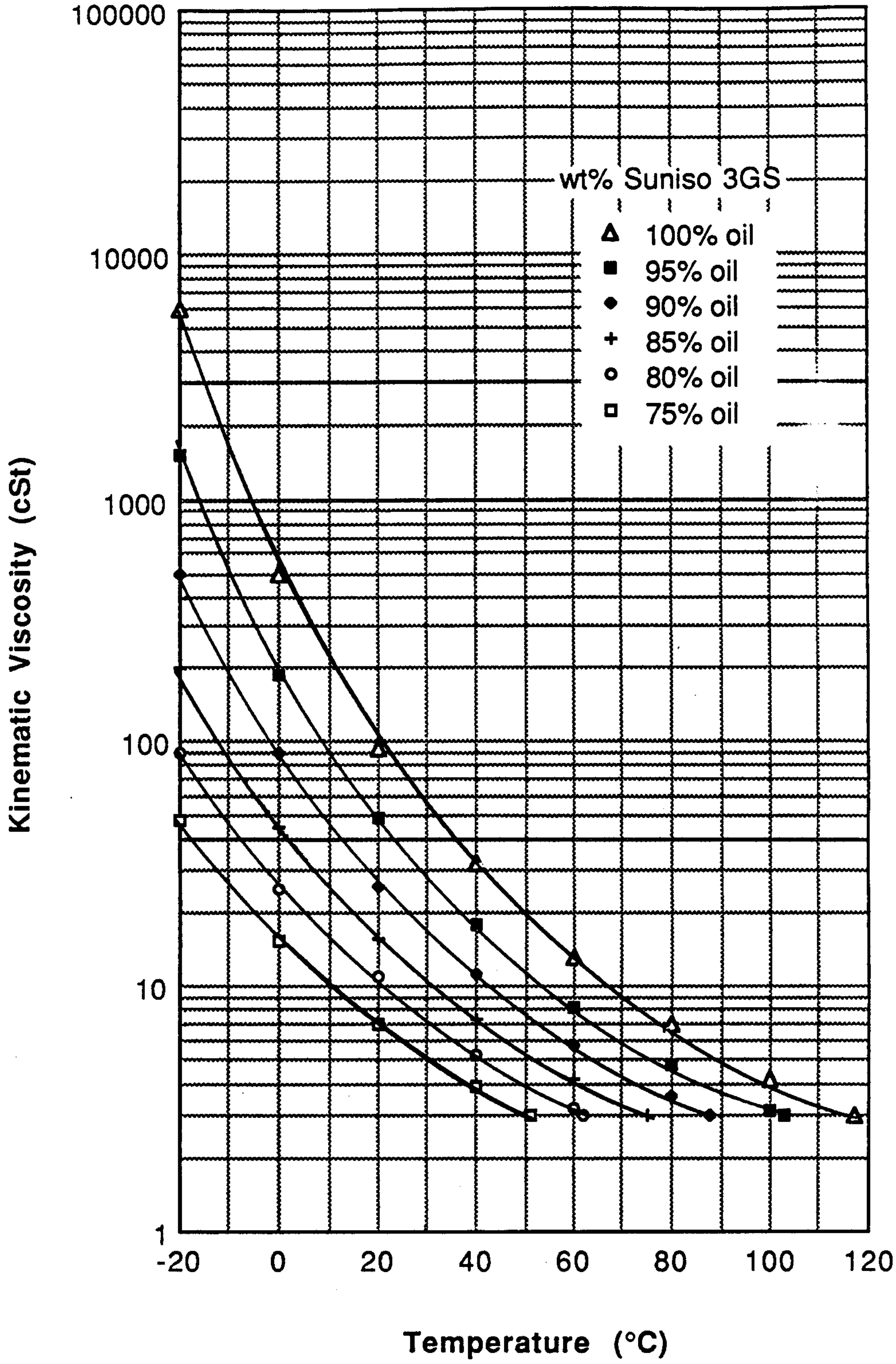


Figure 1.2. Kinematic Viscosity of R-22/Suniso 3GS Mixtures

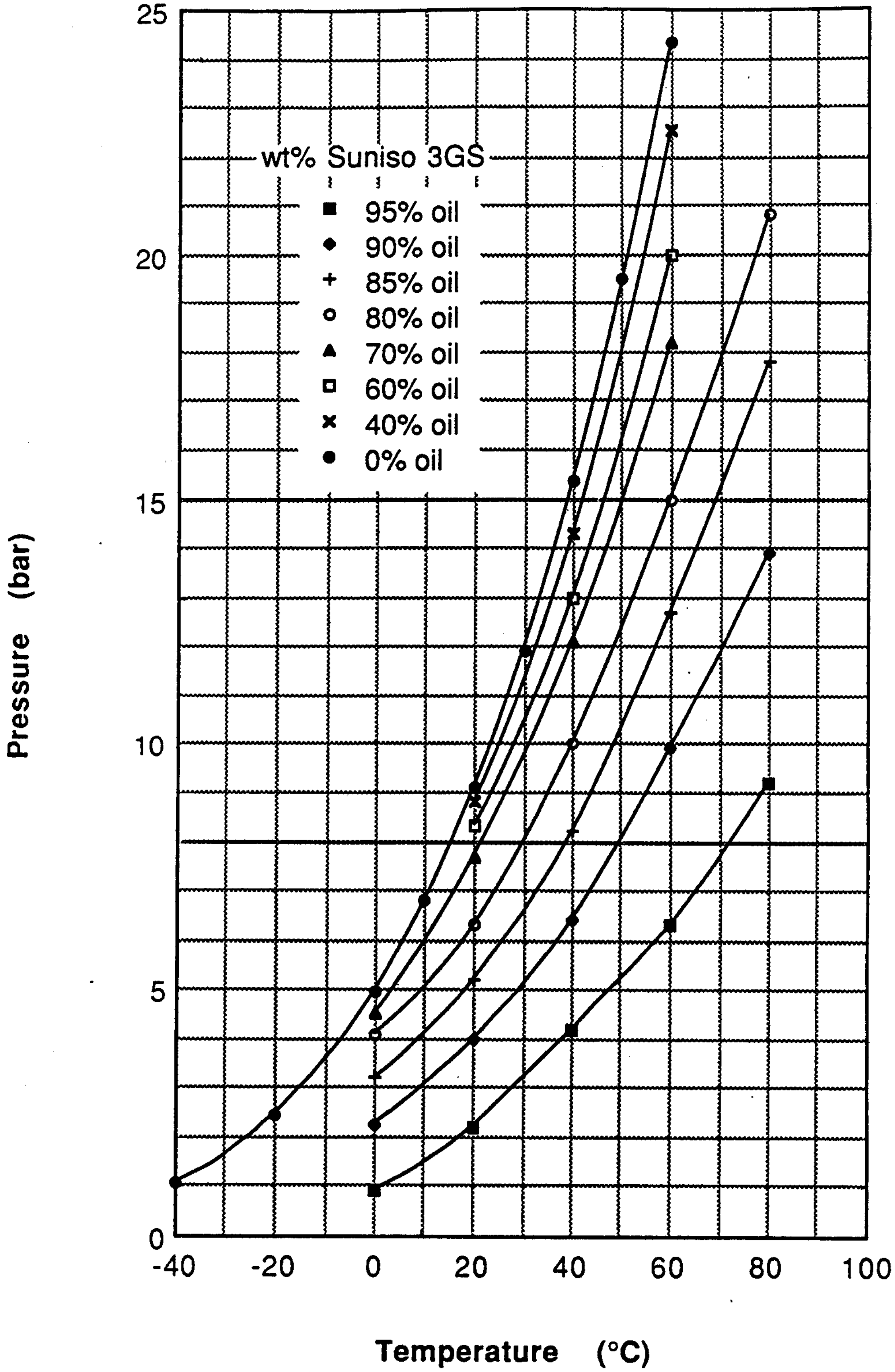


Figure 1.3. Pressure-Temperature Diagram for R-22/Suniso 3GS

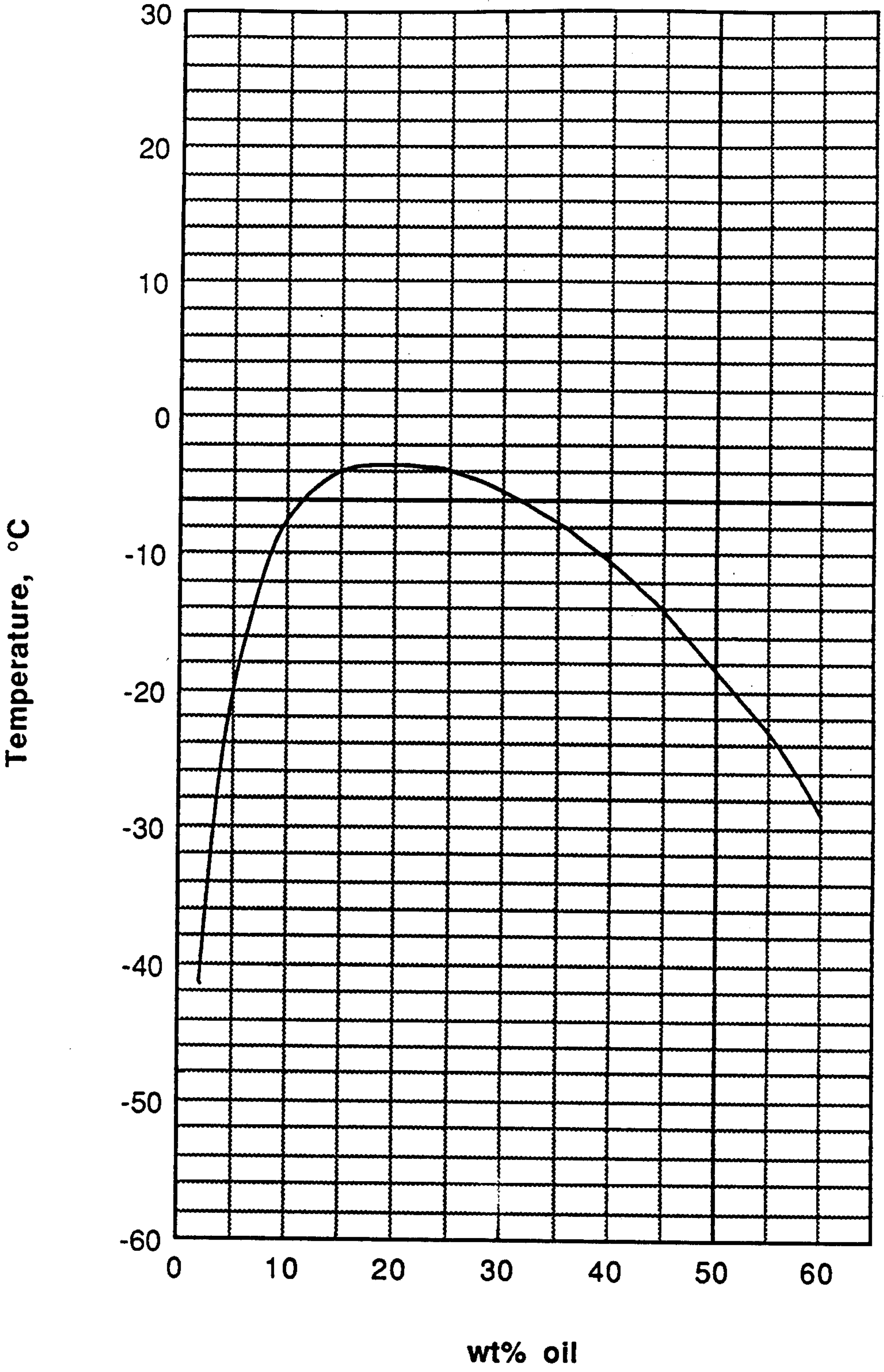


Figure 1.4. Miscibility of R-22 in Suniso 3GS

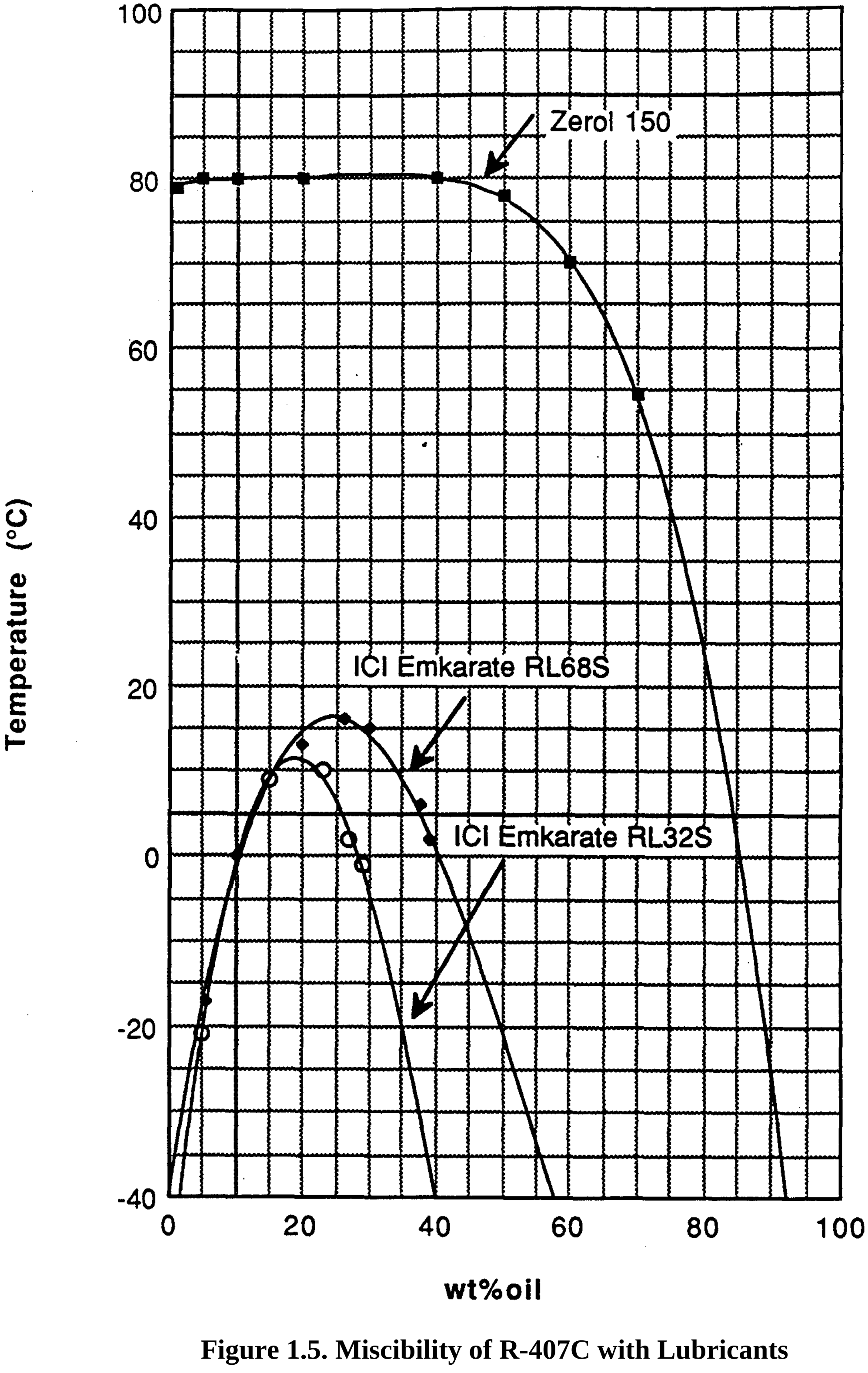


Figure 1.5. Miscibility of R-407C with Lubricants

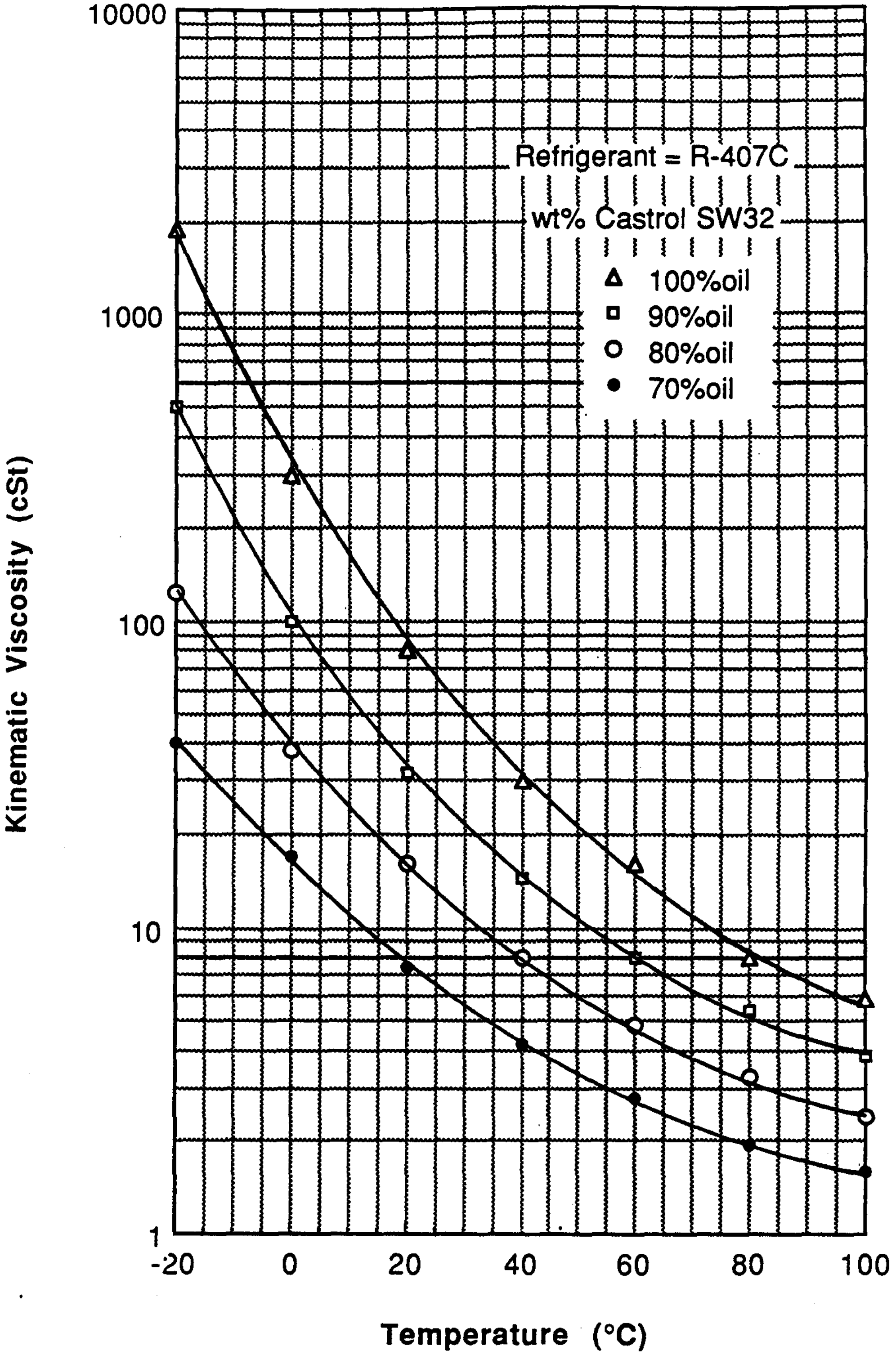


Figure 1.6. Kinematic Viscosity of R-407C/Castrol SW32 Mixture

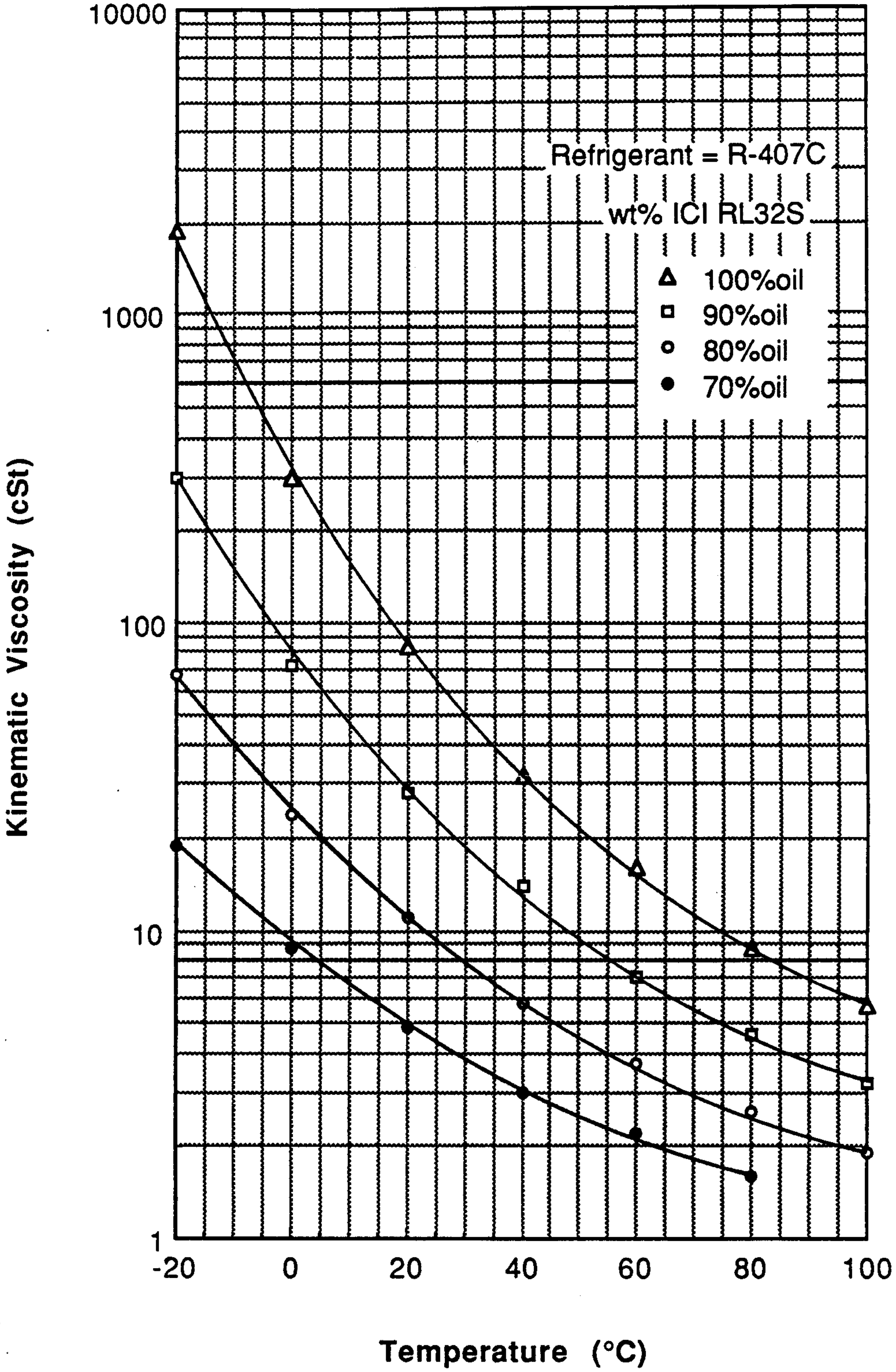


Figure 1.7. Kinematic Viscosity of R-407C/ICI Emkarate RL32S Mixture

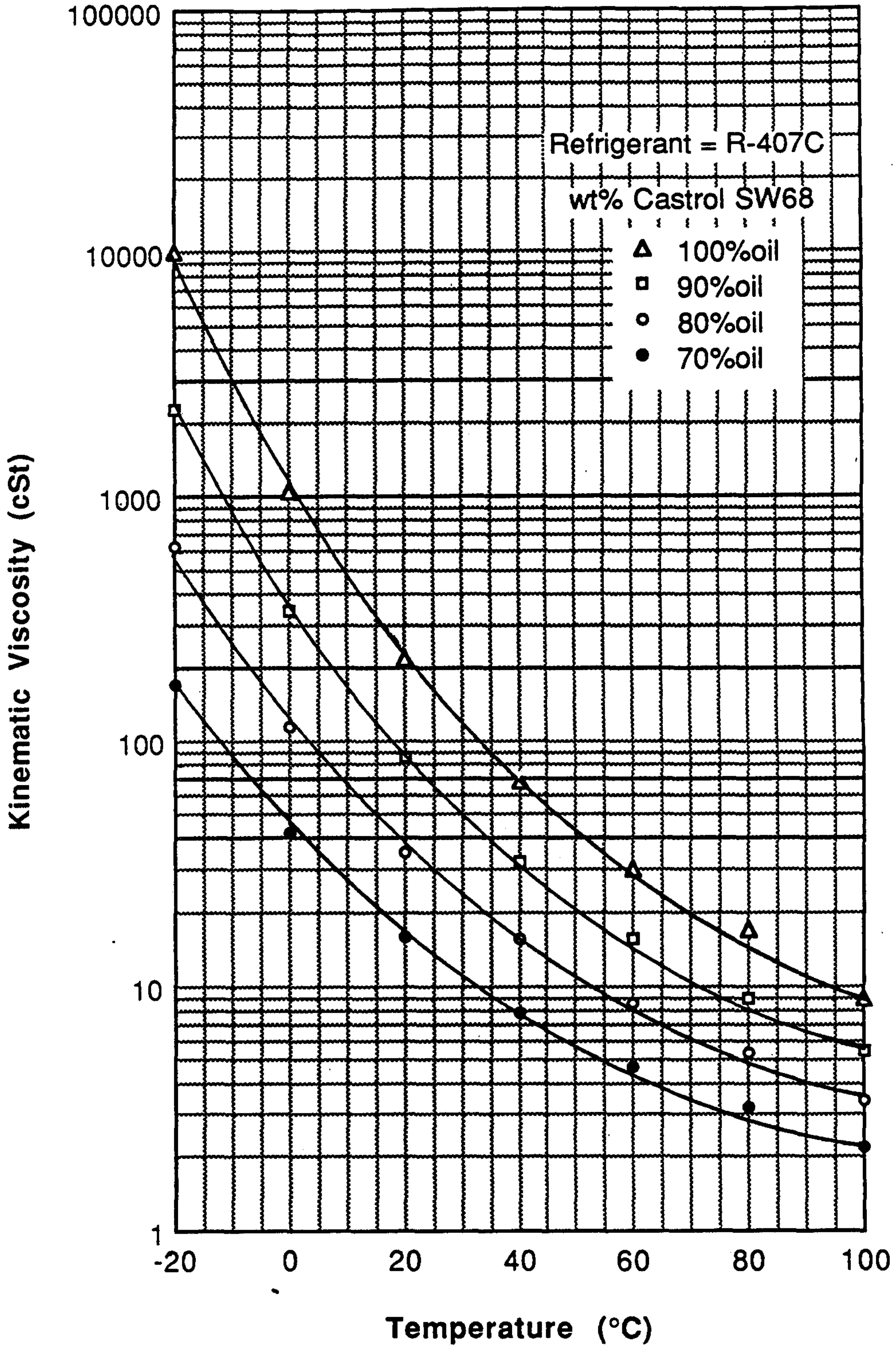


Figure 1.8. Kinematic Viscosity of R-407C/Castrol SW68 Mixture

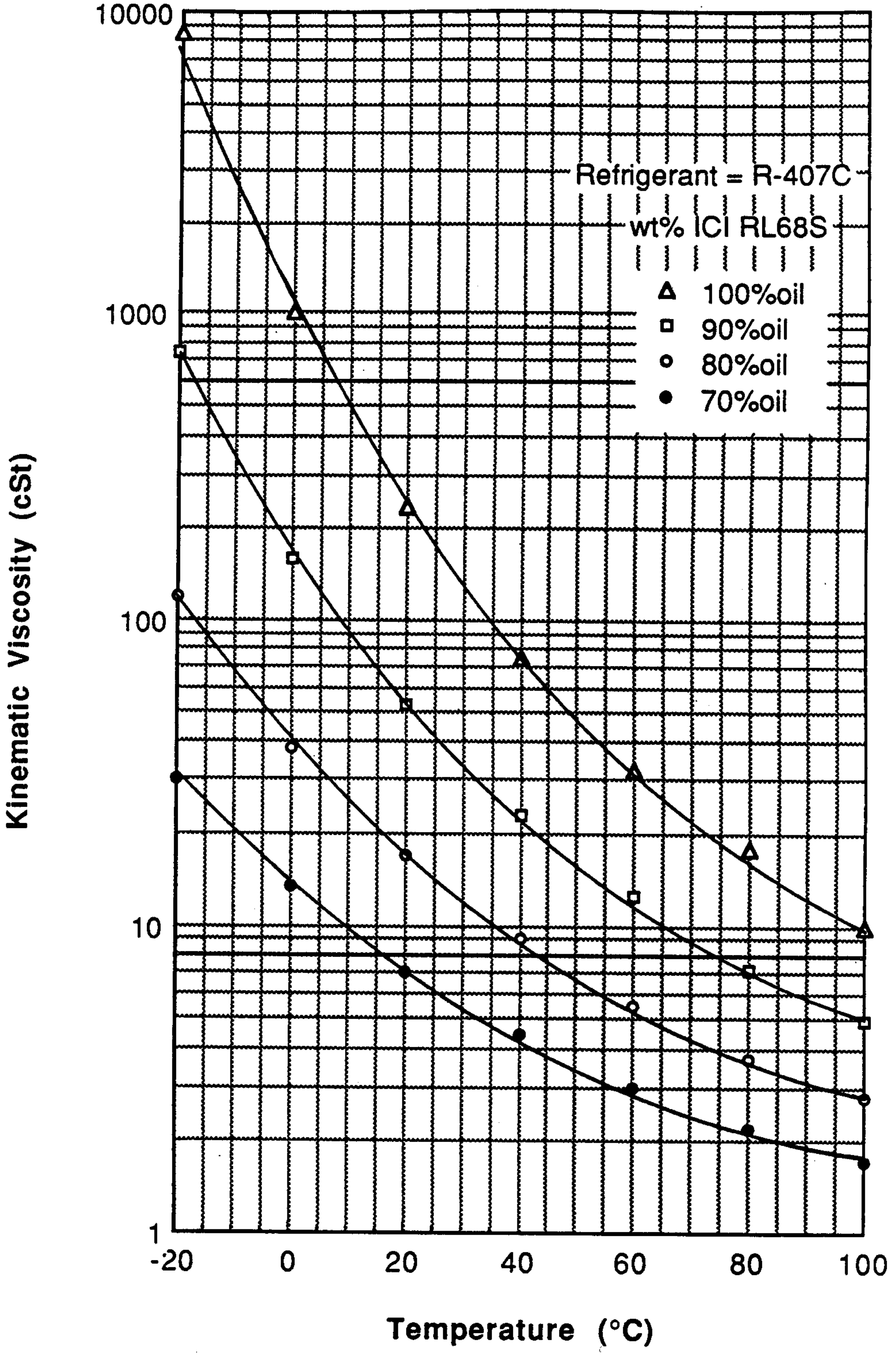


Figure 1.9. Kinematic Viscosity of R-407C/ICI Emkarate RL68S Mixture

TASK 2 - DESIGN, FABRICATE, INSTALL INSTRUMENTATION AND CONDUCT EXPERIMENTS IN DYNAMIC TEST FACILITY

Introduction

The overall objective of this task was to utilize the data developed previously in Task 1 and (summarized in Part 1 of this task), wherein the poor oil management situations for split-system heat pumps could be encountered, and to design a test facility to simulate these worst case scenarios. Furthermore, selection and installation of instrumentation to obtain pressure, temperature, and oil concentration measurements on-line during a wide range of operating conditions was to be completed in this task, as well as development of a test plan and conducting of tests. This overall task was conducted under Part 1 and Part 2 efforts. Part 2 consisted of five subtasks. The efforts undertaken under each part and subtask are described in the following sections. The range and scope of tests actually conducted are described in the last task of this section, while the actual test results and program conclusions are presented in the Task 3 section.

Task 2 - Part 1. Identification of Worst Case Oil Return Situations

The central theme of this ARTI oil management and circulation program was to: 1) identify worst case oil return conditions and design a test apparatus that simulates such conditions, 2) establish experimental baseline performance with R-22/mineral oil, and 3) evaluate R-407C with both high and low miscible POEs and immiscible mineral oils.

The significance of refrigerant/lubricant vapor flow rate and lubricant miscibility and viscosity was discussed in Task 1. The objective of the dynamic test facility was to provide a range of conditions to determine when and where the worst case oil management or zero oil return condition would be encountered. These conditions were then related back to identifying minimum vapor velocities (in different locations during the heating and cooling modes) and comparing them to predicted values. To determine the impact of viscosity on oil management, a range of refrigerant/lubricant mixtures was to be tested. These included refrigerant/lubricant mixtures with low, high and immiscible characteristics. The dynamic test apparatus for achieving such conditions is described below, along with the factors impacting the dynamic test facility features.

Dynamic Test Facility Design Guidelines

Oil return up-suction or hot gas risers require that there be sufficient gas momentum to overcome opposing gravity and viscous forces acting on the oil. By definition, oil traps which violate these conditions must be avoided. Long lines between condenser and evaporator add pressure drop to a system and result in reduced mass flow and capacity. Line diameters can be increased to minimize this effect, but minimum velocities must be maintained to insure oil return.

Empirical relationships, such as that of Ref. 3, have been developed to calculate minimum velocity. Knowledge of the compressor flow characteristics allows one to determine a maximum allowable gas line diameter for any specified suction and discharge condition, as was shown in Table 1.2 and described previously. A test apparatus with a minimum velocity vertical test section effectively simulates worst case long lines and vertical riser combinations. Viscosity is not directly addressed in the empirical relationships, since miscible oils tend to have sufficiently low values, as described in Subtask 1B and in the data of Figs. 1.2 to 1.8. The viscosity of pure immiscible oils exceeded recommended values at operating conditions of about 0°F, especially since there is little dilution with refrigerant which would reduce the viscosity.

Criteria for Design of Test Apparatus

For a **cooling** mode operating condition of 40°F/90°F (T_{sat} in evaporator and T_{sat} in condenser) and the type of compressor selected for the tests, four wide open 3/4 inch diameter suction lines were determined to create a minimum velocity condition for either R-22 or R-407C, as shown in Table 1.2. Lower evaporator temperatures can be achieved by reducing (or blocking) evaporator fan flow.

For **heating mode operating conditions,** the available UTRC Test Facility Indoor Environmental Chambers (IECs) were limited to achieving a lower saturation temperature range of about 0 to 15°F. A 10°F/90°F operating condition (T_{sat} in ambient outdoor exchanger and T_{sat} in indoor exchanger) was determined to cause insufficient velocity in two 3/4 inch hot gas risers. Since the suction line is the factory installation in the outdoor unit during **heating** operation, it must be assumed that adequate provisions will be made for oil return. That is, there are no internal piping connections or configurations that will result in oil trapping or poor oil management conditions introduced by the heat pump system. supplier.

During the test sequence, it is possible that highly immiscible lubrication with high oil viscosity (>650 cSt) may result in oil trapping in the accumulator or suction line. Evaporator temperatures during blocked fan tests should provide low temperatures, which may cause critical behavior in this regard.

In summary, the installation of four 3/4 inch gas lines between indoor and outdoor units, each with appropriate valves at either end, allowed configuring the test loop for a worst case oil management and minimum oil return during either **heating** or **cooling** operation. The vertical height needed only to be sufficient to establish steady state conditions. Even though much higher heights (30 ft or above) were considered as necessary to conduct the tests, the currently available 16 ft height should be adequate. (As indicated previously (in Task 1A), some installations with vertical lines of 100 to 150 ft actually have **no** oil management problem.)

Test Rig Design Considerations and Test Procedure

Given a nominal worst case oil management scenario for R-22, and projected for R-407C, the test plan allowed for evaporator or condenser T_{sat} **variations to confirm a minimum velocity** for the given refrigerant/oil combination being tested. The impact of oil miscibility was determined by establishing maximum viscosity characteristics. Saturation temperature levels were varied further by adjusting either air temperature or fan flow for the appropriate heat exchanger. The test procedure was as follows:

1) Charge with refrigerant/oil combination.

2) Select oil injection rate (at least two rates can be selected, one normal for the compressor, the other(s) to be provided by an oil injection procedure).

3) Run system to nominal **cooling** condition (40°F/90°F) with one 3/4 inch suction line.

4) Open two, three, then four suction lines and confirm oil return management (problem encountered or **not** encountered).

5) Vary evaporator fan flow until compressor oil sump level drops (note critical velocity. and estimate maximum viscosity, also estimate oil return rate, if any, versus injection rate).

6) Operate each suction line independently to return oil to sump.

7) Switch to **heating** condition (10°F/90°F) and repeat procedure.

8) Open second hot gas riser, then third, then fourth, until an oil return problem is encountered.

9) Vary condenser fan flow until oil sump level drops (note critical velocity and estimate maximum viscosity, also estimate oil return rate, if any, versus injection rate).

10) Repeat above at least once for each refrigerant/oil combination.

In summary, the dynamic test facility was designed to evaluate the following parameters and operating ranges and to have the following measurement and visual capabilities.

Table 2.1 Dynamic Test Facility Features and Operating Range

	Cooling Mode	**Heating Mode**
(1) - Minimum Vapor Ranges, ft/min (temperature dependent)	350	200
(2) - Maximum Lubrication Viscosity cSt (at least	650	650
(3) - Achieve Low Temperature IEC Operation T_{sat} with Reduced Fan Flow or Blocked Fan	45°F (32°F)	10°F (0 to 10°F)
Visual Observations of • Oil Return Mechanism in Vapor Lines • Compressor Oil Sump Level and Change of Level (use of remote TV camera and on-line display)	Compressor Suction Lines Both	Compressor Discharge Lines Both
On Line Measurement • Liquid/Lubricant Concentration in System, 0% to 5 to 10% • Include Provisions for Metering in Excess Lubricant, if desired • All Pressure, Temperature, Flow Rates, and, as needed. Refrigerant Composition		

Task 2 - Part 2. Design and Fabrication of Test Apparatus

Subtask 2A - Design and Fabrication of Dynamic Test Apparatus

As previously described, poor oil management and inadequate oil or resupply to the compressor can be attributed to problems in either or both the liquid or vapor side of the system. Each has its own set of important parameters which affect oil return. On the vapor side, the issue is mostly one of insufficient velocity to carry the oil, as shown in Table 1.2. However, the physical geometry of the system can also play an important role. Liquid side problems are mainly related to the solubility and miscibility properties of the refrigerant/lubricant mixture, although the layout of the system also has an effect.

Since the physical arrangement of the indoor and outdoor sections of a heat pump and air conditioning system are usually factory designed and **not** modified on the job side, only installation specific system parameters could be addressed in this study. It should be recognized that component arrangements, i.e., the geometry and slope of the indoor or outdoor exchangers and connecting piping can lead to system oil traps. However, these issues were beyond the scope of this study to evaluate.

The main area of investigation during the course of this study was the lubrication return problem on the vapor side, which was different for either the heating or cooling modes. In order to do this in a representative system, the vapor velocity needed to be altered without affecting the flowrates throughout the rest of the system. In a real system, the velocity would be reduced by installing an oversized vapor line. Often, oversized lines are used to overcome the pressure losses associated with running long lines. However, in order to alter the velocity in the vapor line, the dynamic test was based on the use of **four** oversized vapor lines, connected by a manifold at each end. Each line in the manifold system had a diameter of 3/4 inches, while the normal vapor lines inside of the representative system were 5/8 inches. By opening the manifold lines one at a time, the total velocity running through the lines could be lowered gradually in discrete steps and poor oil management and zero oil return scenarios could be demonstrated. A sight glass was installed at each end of both manifolds to enable visual observation of the flow directly through each manifold.

Figures 2.1 and 2.2 show the dynamic test apparatus schematic and the location of the indoor and outdoor sections during both cooling mode and heating mode operation (Figs. 2.1 and 2.2). An actual 2.5 ton split-system heat pump manufactured by a major supplier was used as the basis for the dynamic facility.

Vapor Velocity Simulation

Due to the physical layout of the dynamic test system, each of the manifolds was constructed somewhat differently, as shown in Figures 2.3 and 2.4. The indoor manifold, BB, shown in Figure 2.3 was relatively simple; four 3/4 inch lines rising from a 3/4 inch manifold tube with a 5/8 inch line to the indoor unit connecting at the bottom. This manifold was installed in the location shown in Figs. 2.1 and 2.2. The outdoor unit manifold, AA, was much more complex, as seen in Figure 2.4. Its location is noted on Figs. 2.1 and 2.2. Four 3/4 inch lines rise from a 3/4 inch manifold tube with a 5/8 inch line to the outdoor unit also rising from the manifold tube. Shut off valves are located six inches above the manifold tube on each vapor line. In operation, this manifold tended to act as an accumulator, with the additional disadvantage of shutting off the flow to one or more of the vapor lines when flooded. To alleviate this particular problem, an additional connecting manifold was constructed to connect each of the four 3/4 inch vapor lines, with shut-off valves in the connecting lines between each of the four vapor lines. This served to equalize the pressure and flow in each of the tubes, and reduced the impact of manifold flooding on the test.

Configuration and Length

The four 3/4 inch vapor lines extended approximately 16 feet vertically from one manifold, then ran another 16 feet horizontally before dropping another 16 feet vertically to rejoin the other manifold (see Figures 2.1 and 2.2). As previously noted, shut off valves were only located on one manifold.

Manifold Behavior

Each of the manifolds behaved somewhat differently during operation. In cooling mode, the vapor line between the indoor and outdoor units contained low pressure gas from the evaporator discharge fed to the accumulator inlet. In heating mode, on the other hand, the vapor line contained high pressure gas from the compressor discharge fed to the condenser inlet. Both the geometry of the manifolds and the operational mode affected the behavior of each manifold and observations taken in each manifold, as will be described below.

Cooling mode

In cooling mode, the supply manifold was on the indoor side unit, and the return manifold was on the outdoor side unit. The vapor return line fed the supply manifold from the bottom, where the vapor flow was distributed to each of the open 3/4 inch vapor lines. As more lines were opened to flow, the velocity in each line decreased, until the local velocity became insufficient to carry oil up the tube. At this point, the oil started to flow back down the tube and eventually collect inside the manifold itself. As the manifold flooded with more and more oil, the oil eventually started to flow back down the supply tube, where the local velocity was sufficient to carry it back to the manifold. Eventually, the flood of oil in the manifold caused one or more of the vapor lines to be blocked, whereupon the local velocity in the remainder of the vapor tubes became sufficient to carry oil up. This floodback of oil through the vapor tubes lasted only as long as one or more tubes remained blocked, and as soon as the liquid level in the manifold dropped sufficiently to open the blocked tubes, the oil return stopped again, and the manifold began to flood again. This cycle was repeated again and again, and was a key visual indication of an oil return problem. This manifold flooding behavior is illustrated in Figure 2.6 and is described in detail below in Subtask 2E. On the other side of the vapor lines, the return manifold fed the flow from all supply lines back through the 5/8 inch vapor return line to the accumulator. Because of the geometry of the outdoor unit manifold, it also acted as an accumulator. In fact, if there was any two phase flow from the evaporator, this is where it tended to collect first.

Heating mode

In the heating mode, the supply manifold was on the outdoor side unit, and the return manifold was on the indoor side unit. The compressor discharge line fed the outdoor unit manifold from the top, where the high pressure gas flow was fed to each of the open 3/4 inch vapor lines. Once again, as more valves were opened, the velocity in each decreased, and the oil eventually had insufficient velocity to be carried up the vapor lines. At this point, the liquid level in the manifold started to rise. The difference in this manifold as compared to the cooling mode supply manifold was that the liquid had no place to drain. Also, when the liquid level became sufficiently high to block one or more vapor tubes, the redistribution manifold would redirect the flow from the open tubes to the blocked tubes. The net result was that no fluctuation in oil flow was expected, unlike the fluctuations expected in the cooling mode. However, as in the cooling mode, manifold flooding was expected to be a key indication of an **oil return** problem.

Viscosity and liquid line flow

The determination of oil return problems caused by liquid flow is extremely difficult to make, particularly because these problems are mostly due to the physical properties of the refrigerant-lubricant mixture and the actual geometry of the system. When an immiscible combination is used, the mixture separates into an oil rich liquid phase and a refrigerant rich phase. The oil rich phase, because of its higher viscosity, may become trapped in certain parts of the system. No attempt was made in this study to create liquid side oil return problems other than by the selection of immiscible lubricant-refrigerant pairs.

Subtask 2B - Selection and Installation of Instrumentation

The objective of this subtask was to select the instrumentation technique best suited for making *in situ* measurements of oil concentration in the system and to define all of the other key instrumentation required to be used in the program. A plan for acquiring, installing and calibrating the instrumentation was also provided in this task, as well as the instrumentation locations and measurements.

Data collected for each refrigerant/lubricant pair and condition included the following:

Refrigerant Gas Velocity/Mass Flow Rate

A mass flowmeter (such as a Micromotion Meter) was used to obtain mass flow and density, which when combined with pressure and temperature and refrigerant properties was used to obtain refrigerant gas velocity.

Temperature and Pressure Profiles

Temperatures and pressures were collected throughout the system with K-type or T-type thermocouples and pressure transducers, using a PC-based data acquisition system. Values were averaged for an accuracy of (±) 1°F for temperature and (±) 1% accuracy for pressure.

Change in the Compressor Oil Level

The scroll compressor shell was replaced with a bolted shell containing a sight glass, which allowed precise determination of oil level in the compressor at all times. Provisions were also included for metering in additional lubricant, if desired, in the bolted shell.

To determine oil return conditions, a representative 2.5 ton heat pump with a scroll compressor was instrumented and was run at nominal conditions for heating and cooling. The indoor and outdoor units of the heat pump were placed into separate Indoor Environmental Chambers (IECs), which provided precise temperature, humidity, and airflow conditions for simulation of various operative conditions. During the course of each test, transient pressure and temperature data were taken, as well as dynamic indications of oil circulation by a number of means.

Indoor Environmental Chambers (IECs)

The experimental facility included two IECs, a nominal 2.5 ton capacity and a nominal 5 ton capacity system. The IECs can provide exact control of supply temperature from 25°F to 150°F, relative humidity from ambient to 100%, and airflow up to 3000 cfm. The supply air side of the indoor unit of the heat pump was connected directly to the 2.5 ton IEC. In contrast, the outdoor unit was placed in a large chamber which was connected to the 5 ton IEC. This is shown schematically in Figure 2.5. In normal operation, the indoor unit air flowrate is set to 1000 cfm while the outdoor unit air flowrate is set to 2000 cfm.

Pressure and Temperature Data Acquisition

Eight thermocouples and six pressure transducers were installed in the system to provide an accurate determination of the refrigerant state throughout the cycle. The instrumentation also included two air-side thermocouples at the entrance and exit of each test section. Type J thermocouples were selected for their good low temperature response. Figures 2.1 (cooling mode) and 2.2 (heating mode) show the instrumentation locations as installed for system tests. The thermocouple and pressure transducer data were sampled once per second by a dedicated PC (PC2 on Figure 2.5). A Labtech Notebook, a commercially available software package, was used to display and record the data.

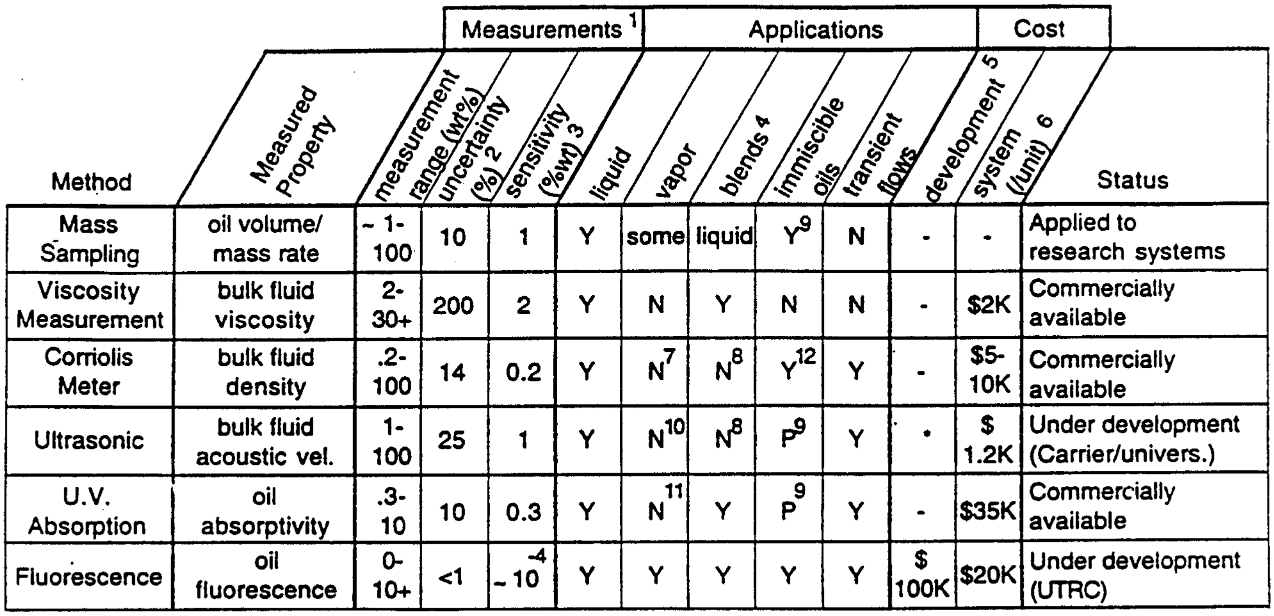
Oil Flow Rate and Distribution of Oil In the System

The various methods for measuring oil concentration in operating HVAC systems were reviewed in detail by UTRC/UTC. A summary of basic features, limitations, and range of applicability for six possible techniques together with development status and/or acquisition cost and current status is provided in Table 2.2. It is concluded from this review that two available techniques, ultraviolet and ultrasonic were viable candidates for use under this program.

Ultraviolet absorption device

Briefly, the ultraviolet absorption meter measures the amount of incident ultraviolet light absorbed by the refrigerant/oil mixture. Since the absorption characteristics are dependent on the constituents, the meter can be calibrated for each mixture of interest. A commercially available system is now operational at UTRC. Tests have been conducted using a 3-ton scroll compressor installed on a calorimeter test stand operating with a R134a/Mobil 68 refrigerant/oil mixture. The instantaneous on-line results were compared with an independent oil sampling measurement technique, as presented in Fig. 2.8. As shown in Fig. 2.8, the two methods are in very good agreement and accordingly, the ultraviolet meter can be used with confidence to make dynamic measurements with other refrigerant/oil mixtures. This calibration information is stored in the unit software for real-time use during system tests.

Table 2.2 Comparison of Oil Concentration Measurement Methods



Method	Measured Property	Measurements[1]			Applications					Cost		Status
		measurement range (wt%)	uncertainty (%)[2]	sensitivity (%wt)[3]	liquid	vapor	blends[4]	immiscible oils	transient flows	development[5]	system (/unit)[6]	
Mass Sampling	oil volume/ mass rate	~ 1-100	10	1	Y	some	liquid	Y[9]	N	-	-	Applied to research systems
Viscosity Measurement	bulk fluid viscosity	2-30+	200	2	Y	N	Y	N	N	-	$2K	Commercially available
Corriolis Meter	bulk fluid density	.2-100	14	0.2	Y	N[7]	N[8]	Y[12]	Y	-	$5-10K	Commercially available
Ultrasonic	bulk fluid acoustic vel.	1-100	25	1	Y	N[10]	N[8]	P[9]	Y	•	$ 1.2K	Under development (Carrier/univers.)
U.V. Absorption	oil absorptivity	.3-10	10	0.3	Y	N[11]	Y	P[9]	Y	-	$35K	Commercially available
Fluorescence	oil fluorescence	0-10+	<1	$\sim 10^{-4}$	Y	Y	Y	Y	Y	$ 100K	$20K	Under development (UTRC)

Notes: unk - unknown, P - possible, Y - yes, N - no

* No known commercial system

1. Measurement performance based on mfgr's spec.'s. For fluorescence, estimate based on qualitative measurements at UTRC and component spec.'s.
2. (Wt% uncertainty at 1 wt%)/(1 wt%) x 100%.
3. Minimum measurable oil concentration.
4. "N" in this column indicates blend significantly affects measured property. (In all cases, blend concentration affects %wt.)
5. Cost to produce a system from current state of technology.
6. Cost of commercial system or estimated cost of a unit once technology is developed.
7. Buoyancy, inertial response limited.
8. Uncertainty in blend at measurement point can have large effect on oil concentration measurement.
9. Must be homogeneously mixed.
10. Acoustic scattering interferences.
11. Optical scattering interferences.
12. Must be single phase (e.g., liquid oil and refrigerant)

Ultrasonic device

The ultrasonic meter is under development at the University of Illinois, Air Conditioning and Refrigeration Center (ACRC) and at Purdue University, both under support from the Carrier Corporation. A schematic diagram of the meter and typical calibration data are presented in the Capabilities Section of this proposal for an R22/alkyl benzene (Lunaria) mixture. This type of meter operates by measuring the speed of sound in the refrigerant/oil mixture via transmitter and receiver transducers. Since the mixture speed of sound is dependent on the oil concentration, this meter can also be calibrated for each mixture of interest.

Since the traditional ASHRAE standard method for measuring the proportion of lubricant in the liquid refrigerant (BSR/ASHRAE 41.1-1984R) cannot be conveniently used for the subject program, both the ultraviolet and ultrasonic methods were candidates for use in measuring *in situ* time dependent oil concentration levels in operating systems. The final choice of the ultraviolet method was based on both the availability and maturity of the technology.

Oil Circulation Data and Measurement Techniques

Oil circulation was determined by a number of means. A Jasco UV oil concentration meter was used to measure the oil concentration in the liquid line. Visual observations of flow through the vapor line manifolds provided an indication of good oil return in the vapor line, as described in the previous section. Finally, the oil sump level in the compressor was continuously monitored and recorded for an indication of the amount of oil in the system.

UV Oil Meter Calibration Technique

The JASCO ultraviolet oil concentration meter measures oil content by determining the amount of ultraviolet (UV) light that a refrigerant/oil mixtures absorbs. The instrument does this by monitoring the absorption occurring at a specific uv wavelength at a point in the system. The device uses a spectrometer system to narrow the spectrum of the light source for accurate measurements at a single wavelength. Since the absorption characteristics are dependent on the constituents, the meter can be calibrated for each mixture of interest. It is generally limited to oil concentrations of about 10% or less by weight because of loss of signal at higher concentrations. In addition, measurement accuracy may be compromised if oils are used having inherently low uv absorbance levels.

Most oils (POEs, AB, mineral oils) have characteristic absorbance in the ultraviolet region while refrigerants show little absorbance at these same wavelengths. The oil concentration in a refrigerant (vol/vol or wt/vol) is proportional to the absorbance according to Beer's Law which states:

$$A = \varepsilon * C * 1$$

where ε is the extinction coefficient of the oil at a given wavelength, C is the vol/vol or wt/vol concentration, and 1 is the optical pathlength of the flow cell. Since the extinction coefficient is unknown for any given oil, calibration is necessary for each type of oil. An oil calibration can be performed by preparing a series of oil concentrations (wt/wt) in some appropriate solvent and finding the absorbance of each sample.

A more accurate calibration takes into account the compressibility effects of the refrigerant at various pressures and temperatures. Since the refrigerant is compressible, the fractional volume of the refrigerant will change relative to the constant volume of oil as the mixture temperature or pressure changes. This, in turn, means that the vol/vol ratio will change with compression but the wt/wt concentration of the oil will remain the same. These effects can be compensated for by performing a refrigerant calibration at various temperatures and pressures. Since most refrigerants are vapor at room temperature, and calibration must be performed using liquid samples, a high pressure vessel is required for this calibration. To avoid this, it is possible to calculate the P and T correction coefficients using refrigerant properties data as outlined in the analytical refrigerant calibration section in the JASCO instrumentation. Due to the high degree of oil concentration measurement accuracy required for the tests and the limited time available for performing the oil/refrigerant mixture calibrations, the analytical refrigerant calibration method was chosen. Typical results from the calibration are shown in Figure 2.7 for one particular oil/refrigerant mixture of interest in the program (R-407C/Castrol SW32). The results show that by calculating approximately four absorbent concentrations, a straight line of the log of oil/refrigerant concentration can be determined.

To verify the accuracy of the calibrations performed, tests were conducted using a 3-ton scroll compressor installed on a calorimeter test stand operating with an HFC-134a/Mobil 68 refrigerant/oil mixture. The on-line results from the meter were compared with an independent oil injection/measurement technique that used a digital flow meter for monitoring the amount of oil being injected. The two methods were found to be in very good agreement, as shown in Figure 2.8. Accordingly, the UV meter can be used with confidence for other refrigerant/oil mixtures.

Separate calibrations of the UV meter, both at normal oil conditions and at much higher concentrations of 2.5 to 5.0%, were also encouraging. These data for HFC and POE are shown in Figure 2.9. The entire range of absorbance concentrations for the five miscible refrigerant/lubricant mixtures are shown in Table 2.3. The data highlights that the R-407C/POE ICI RL325 can be extremely difficult to calibrate because of the extremely low UV absorbance characterization (i.e., the oil is extremely clear to UV light).

Table 2.3. Experimental Calibration Data for UV Oil Concentration Meter with R-407C at Temperatures from 16.5°C to 18.8°C (61.7°F to 64.2°F)

Lubricant	Concentration, C	Absorbance (abs)
ICI SW 32	0.3869	0.0129
	1.0079	0.0473
	2.505	0.1290
	4.8871	0.2640
ICI SW 68	0.4182	0.0184
	0.9151	0.0434
	3.2187	0.1471
	5.2570	0.2378
ICI RL 32	0.4630	0.0016
	0.7187	0.0042
	2.2810	0.0189
	4.1793	0.0369
ICI RL 68	0.3135	0.0061
	0.7810	0.0161
	2.5874	0.0600
	4.2013	0.1007

* These low levels may be source of inaccuracy

Other Data

The tests also included a number of other data samples. The flow through the liquid line was measured by a S025 Micromotion flow meter and recorded every 5 seconds. By determination of the saturated vapor density of the refrigerant charged in the system and the number of vapor lines open, the vapor velocity through the vapor lines was calculated. A highly accurate scale was used to determine the amount of refrigerant charged into the system. The scale, an A&D model EP-60KA, has a resolution of 0.005 lbs. The charge weight was determined by weighing the refrigerant cylinder before and after charging.

Subtask 2C - Oil Selection and Property Verification

As discussed under Subtask IA, two mineral oils, Suniso 3GS and Suniso 1GS and four POE formulations, ICI RL32S, ICI RL68S, Castrol SW32 and Castrol SW68, were chosen as lubricants for the circulation tests. The manufacturers specifications for viscosity, total acid number (TAN), and water content are summarized in Table 2.4.

Table 2.4 Lubricant Physical Properties

Lubricant	Kinematics Viscosity (cSt)		TAN (mg KOH/g)		Water (ppm)	
	-20°C	40°C	Mfg. Spec.	Measured	Mfg. Spec.	Measured
Suniso 1GS	---	12	<0.03		<35	
Suniso 3GS	4000	33	<0.03		<25	
ICI Emkarate RL32S	1800	32	<0.02	(0.025)	<50	(46)
ICI Emkarate RL68S	7356	74	<0.02	(0.028)	<50	(39)
Castrol SW32	-	32	<0.15	(0.142)	<50	(97)
Castrol SW68	-	68	<0.15	(0.144)	<50	(125)

Water content and acid number are verified on an on-going basis for selected samples of these lubricants at the Central Engineering Services Group at Carrier in Syracuse, NY since these lubricants are used in several Carrier HVAC equipment lines. Data collected to date are shown in parenthesis in Table 2.4. The lubricants tested were close to manufacturer's specifications, with the exception that the Castrol POEs measured somewhat higher water content. This could be due to the sampling technique; these lubricants are very hygroscopic. The POE lubricants were handled under N_2 purge to minimize water absorption upon system charging.

Subtask 2D - Test Plan

The refrigerant-lubricant pairs selected in the previous task were run in the test system according to the plan developed in this task, shown in Table 2.5. Each refrigerant/oil combination was tested at two environmental conditions. One corresponded to a typical cooling condition (DOE-A) in cooling mode operation, and one corresponded to a typical heating condition (no lower than 20°F outside air) in heating mode operation. These tests were each repeated at least once for verification. In addition, most combinations were also tested with additional oil injected into the circulating flow to simulate systems with higher nominal oil circulation rates.

Table 2.5 Original Test Plan

Test Refrigerant/Lubricant Combinations	Test Conditions Run
1) R22/Mineral Oil (3GS), Baseline	Cooling Mode, Cooling Mode w/Oil Injection Heating Mode, Heating Mode w/Oil Injection
2) R407C/Mineral Oil (3GS), MO1	Cooling Mode, Cooling Mode w/Oil Injection Heating Mode, Heating Mode w/Oil Injection
3) R407C/Mineral Oil (1GS), MO2	Cooling Mode, Cooling Mode w/Oil Injection Heating Mode, Heating Mode w/Oil Injection
4) R407C/Castrol SW32, POE1	Cooling Mode, Cooling Mode w/Oil Injection Heating Mode, Heating Mode w/Oil Injection
5) R407C/Castrol SW68, POE2	Cooling Mode, Cooling Mode w/Oil Injection Heating Mode, Heating Mode w/Oil Injection
6) R407C/ICI RL32S, POE3	Cooling Mode, Cooling Mode w/Oil Injection Heating Mode, Heating Mode w/Oil Injection
7) R407C/ICI RL68S, POE4	Cooling Mode, Cooling Mode w/Oil Injection Heating Mode, Heating Mode w/Oil Injection

The baseline refrigerant/oil combination was HCFC-22 and 3GS mineral oil. This combination was then replaced with the baseline blend/oil combination: R407C (R32/R125/R134a with 23/25/52 mass fractions) with mineral oil. The baseline oil was also to be replaced in series by four different POE oils, as specified in the previous section. Two of the oils had a lower miscibility and two had a higher miscibility with the HFC blend. Two also had higher viscosities (ISO-68), and two were lower in viscosity (ISO-32). This is described in further detail in Task 1. A total of seven refrigerant/oil combinations were planned to be tested.

Oil Replacement Procedure

When replacing an oil, the following procedure was followed. The compressor sump was drained of the old oil, and replaced with the new oil. The old refrigerant charge was recovered and replaced with a fresh charge. The system was turned on and cycled through heating and cooling mode conditions. The compressor sump was once again drained and replaced with a fresh charge of the new oil. The refrigerant charge was recovered and replaced with a fresh charge. The procedure was repeated once more, for a total of three refrigerant and oil changes before a new test was run. This ensured that as much of the old oil was removed from the system as possible, and the contamination potential of the new oil by the old oil was minimized. All applicable environmental procedures were followed during the recovery and disposal of the oil and refrigerant charges.

Subtask 2E - Test Sequence and Data Analysis

The experimental apparatus was run with the objective of finding the minimum velocity at which oil return through the vapor line was possible. The data collected during the course of each test run was plotted to determine oil circulation trends. Both visual observations in the manifolds and the compressor oil sump level as well as measured data were used to determine oil circulation problems.

Test Procedure

The same procedure was used for each test run and generally followed the Part 2 (Task 2) guidelines. First, the system was turned on and let warm up to a steady state condition with one vapor line open. The charge in the system was next adjusted to achieve ten degrees of superheat at the compressor inlet. This was done to minimize the amount of liquid refrigerant in the suction line and accumulator to prevent fractionation of the refrigerant. After the system was again at steady state, the conditions inside the manifolds were noted on a log and an additional vapor line was opened. If the test was in heating mode, the valve connecting the two vapor lines in the redistribution manifold was also opened. The system was given at least 15 minutes to stabilize before the manifold conditions were again logged and another vapor line was opened. This was repeated until all four vapor lines were opened. Then, for some tests, the evaporator temperature was additionally lowered by lowering the evaporator airflow in order to lower the vapor density and thus the vapor velocity. Finally, all vapor lines except one were closed, and the test was repeated.

After the test had been run, the data collected was processed and plotted. Since the data acquisition system did not record when vapor lines were opened and closed, the record of these events had to be recorded by hand, and later correlated to the proper data system times. The data acquisition system recorded data every five seconds. This data was averaged to one minute intervals for plotting. The one exception was the liquid line oil concentration from the UV meter, which was averaged to 15 second intervals for better resolution. The system temperatures, system pressures, and air side temperatures were each plotted on separate graphs. In addition, the liquid line oil concentration from the UV meter was overlaid with the vapor line velocity on one graph. This graph was used to determine when the velocity was insufficient to carry the oil. For reasons mentioned in Subtask 2A, poor oil return in cooling mode was indicated by a fluctuation in the oil concentration. For the heating mode, a drop-off in the oil concentration was expected to indicate a problem.

Oil Return Measurements and Observations

Along with the UV meter indications, the visual observations made of the vapor manifolds and oil sump also gave indications of oil return problems. During operation, the liquid inside the vapor manifolds was not all at one level. As the flow through the vapor lines decreased and oil began to collect in the manifolds, it was typically forced to one side by the flow through the manifolds. This is shown schematically for the indoor side manifold in Figures 2.6a-d. In Figure 2.6a, one valve is open and oil travels normally through the manifold with little to none being trapped in the manifold. With two valves open, as in Figure 2.6b, the velocity is still not low enough to cause a problem. Figure 2.6c shows oil starting to accumulate on one side with three valves open. Note that the oil is accumulating on the side with both vapor lines open, meaning that the local flow inside the tubes became insufficient to carry all the flow up. The system reaches steady state with this type of liquid level, which would correspond to a certain amount of oil being carried from the level in the manifold and a certain amount without sufficient velocity, which returns to the manifold. With all four vapor lines open (Figure 2.6d), one side has totally flooded, with the level fluctuating slightly as it alternately blocks and opens the fourth line, as was explained in Subtask 2A. The same order of events did not occur in all of the tests. In some tests, the manifold flooded much more quickly (less open vapor lines needed), and in others additional inducements were needed (such as lowering the evaporator temperature) in order to cause flooding. In general, however, the flooding occurred very dramatically, as between Figures 2.6b and 2.6c, where the liquid level jumped rapidly. These liquid level jumps were very repeatable, and the velocities at which they occurred was recorded.

Oil Sump Video Monitoring and Display

During the course of testing, an on-line monitoring of the compressor sump oil level was conducted and recorded on video for review and confirmation of oil management issues.

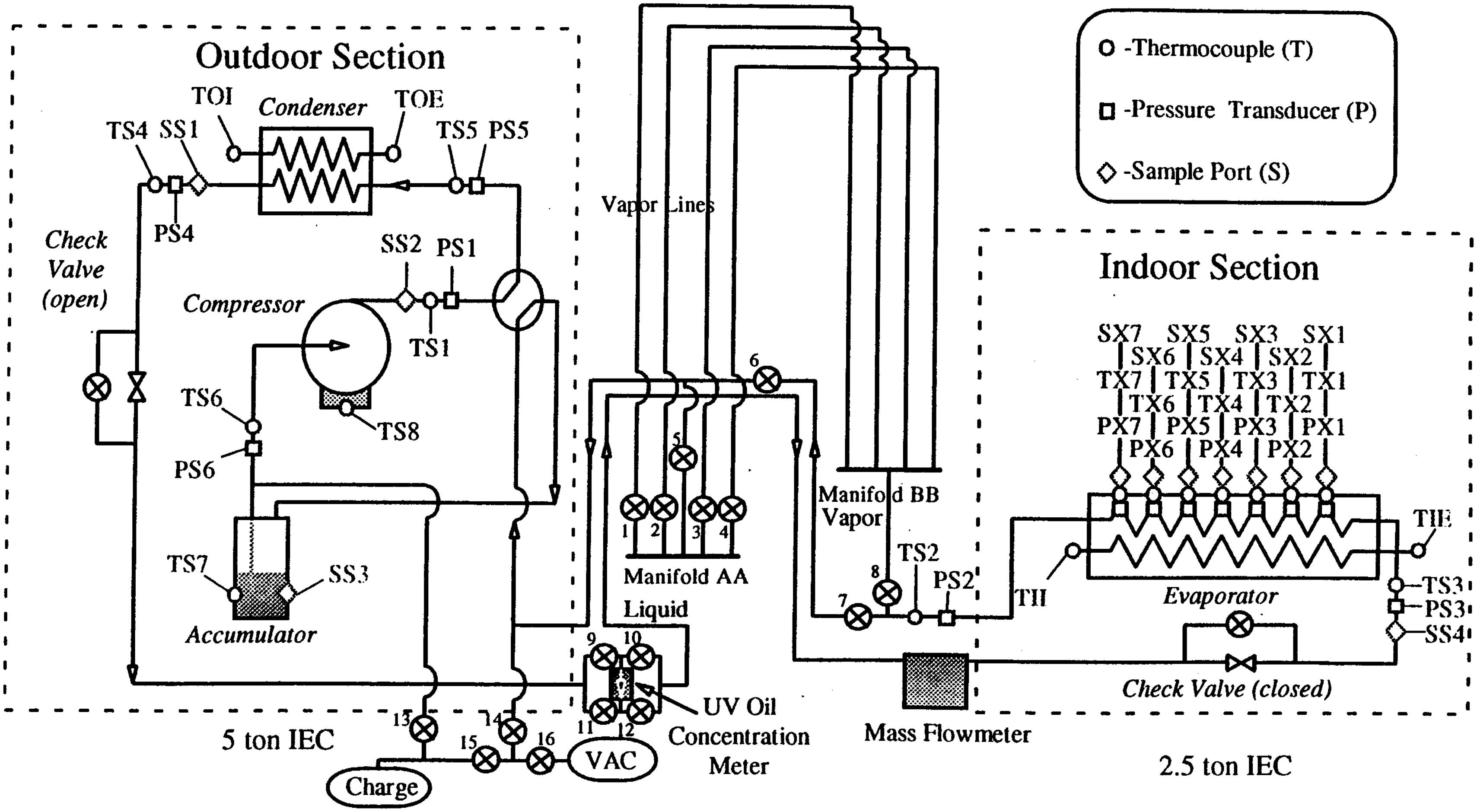


Figure 2.1. Oil Circulation Program Dynamic Test Facility - Cooling Mode

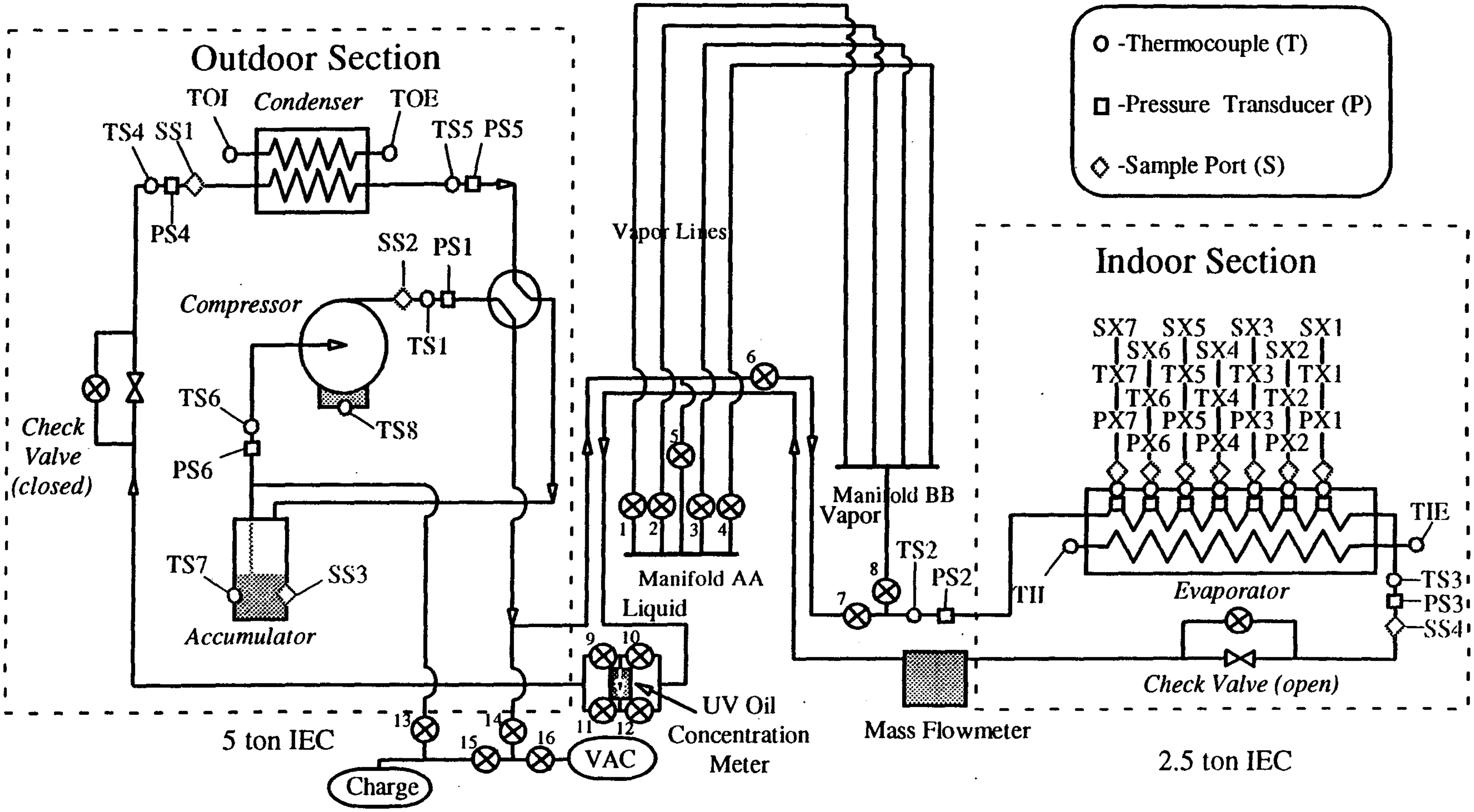


Figure 2.2. Oil Circulation Program Dynamic Test Facility - Heating Mode

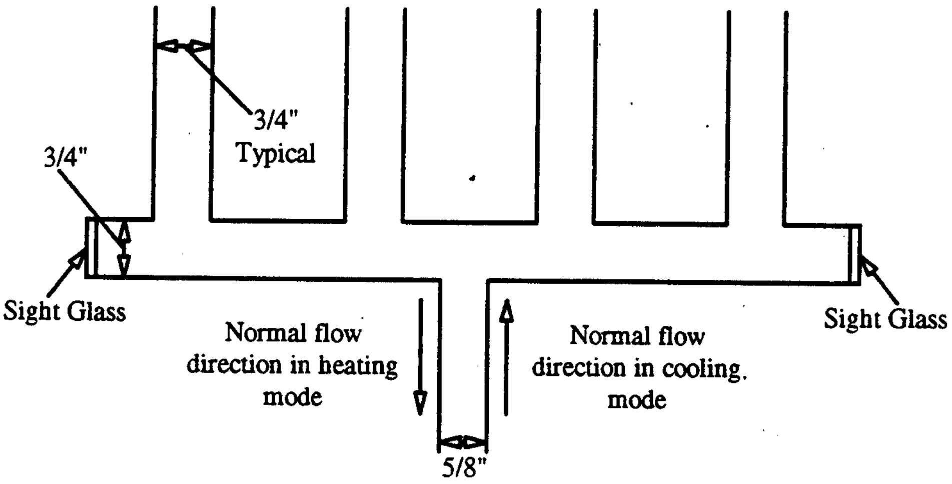


Figure 2.3. Indoor Unit Vapor Manifold (BB)

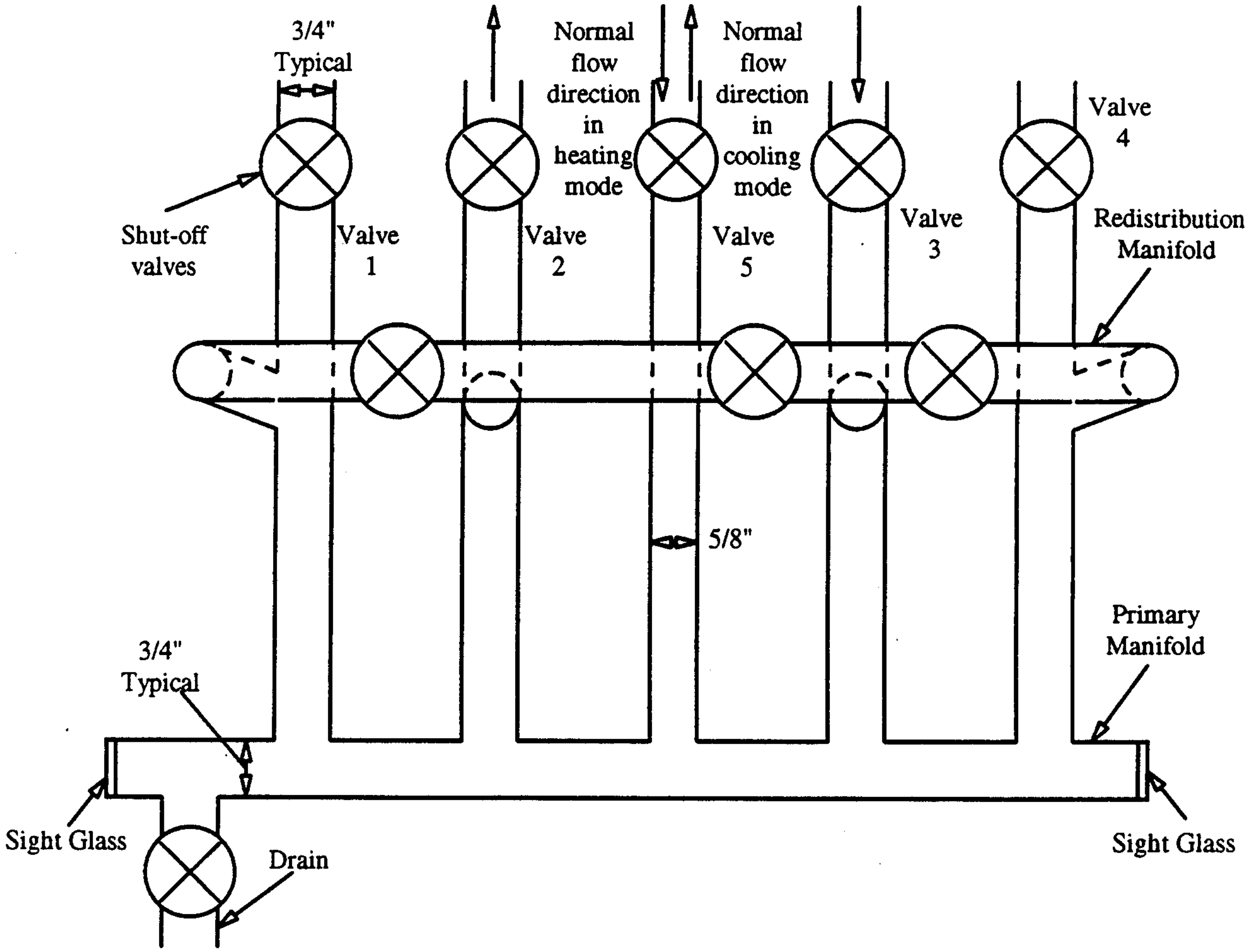


Figure 2.4. Outdoor Unit Vapor Manifold (AA)

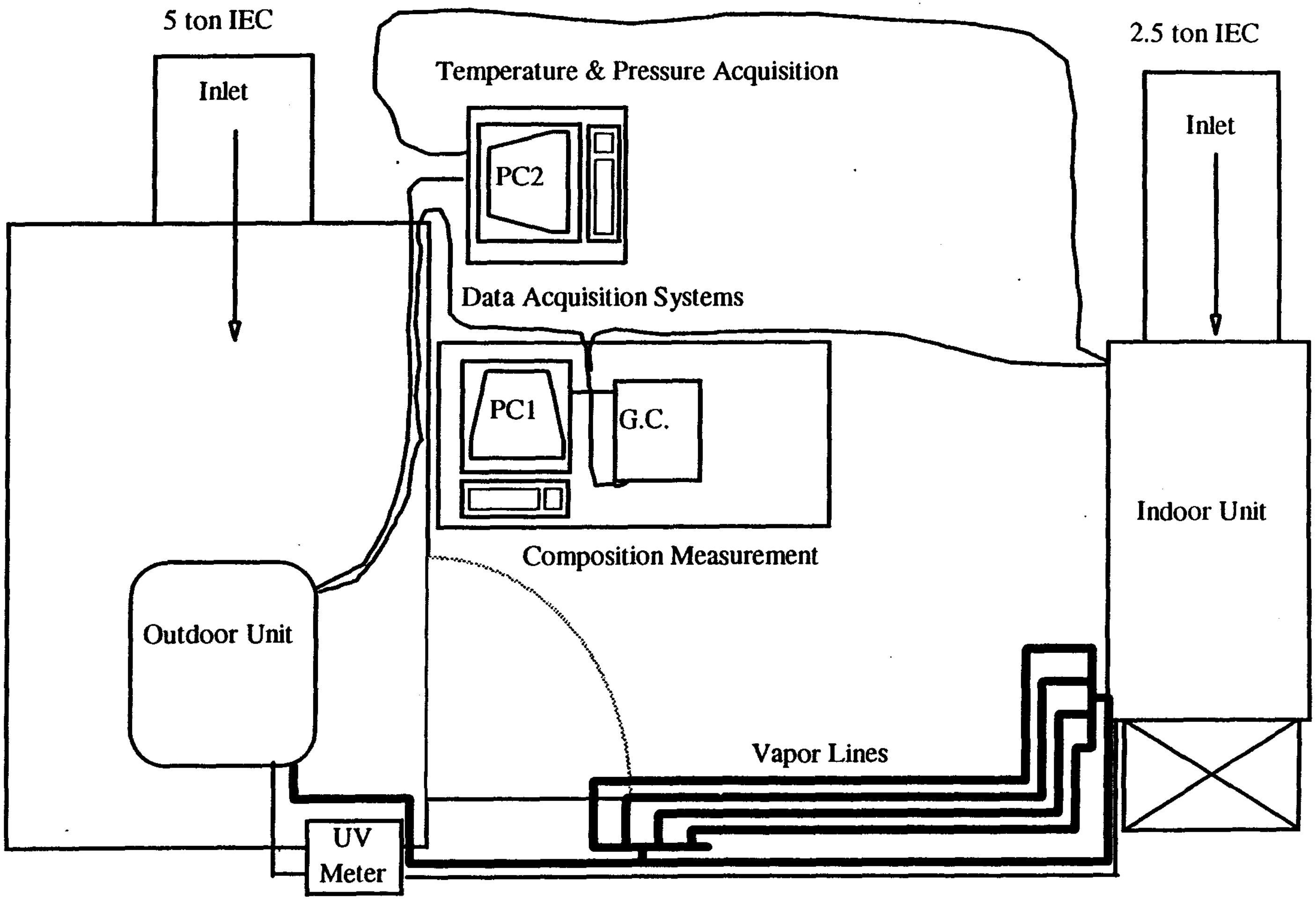


Figure 2.5. Lab Arrangement

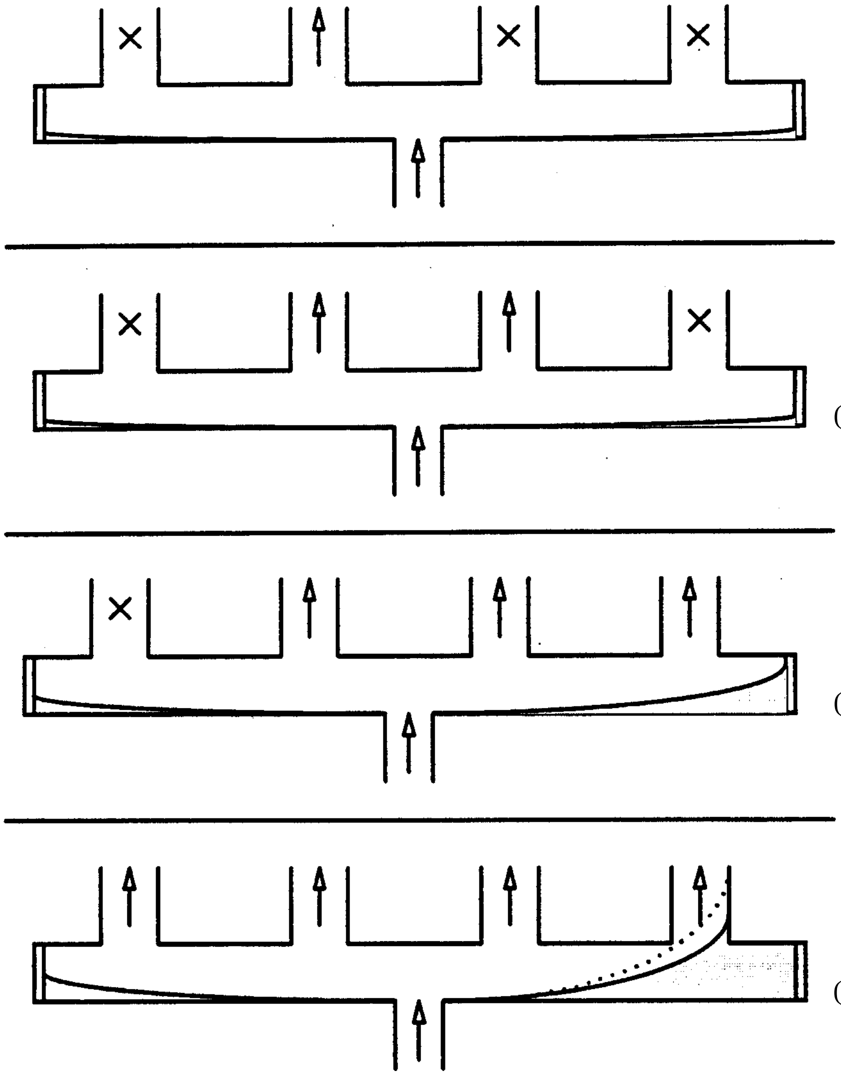


Figure 2.6a
(1 vapor line open)

Figure 2.6b
(2 vapor lines open)

Figure 2.6c
(3 vapor lines open)

Figure 2.6d
(4 vapor lines open)

Figure 2.6. Typical Manifold Flooding Behavior

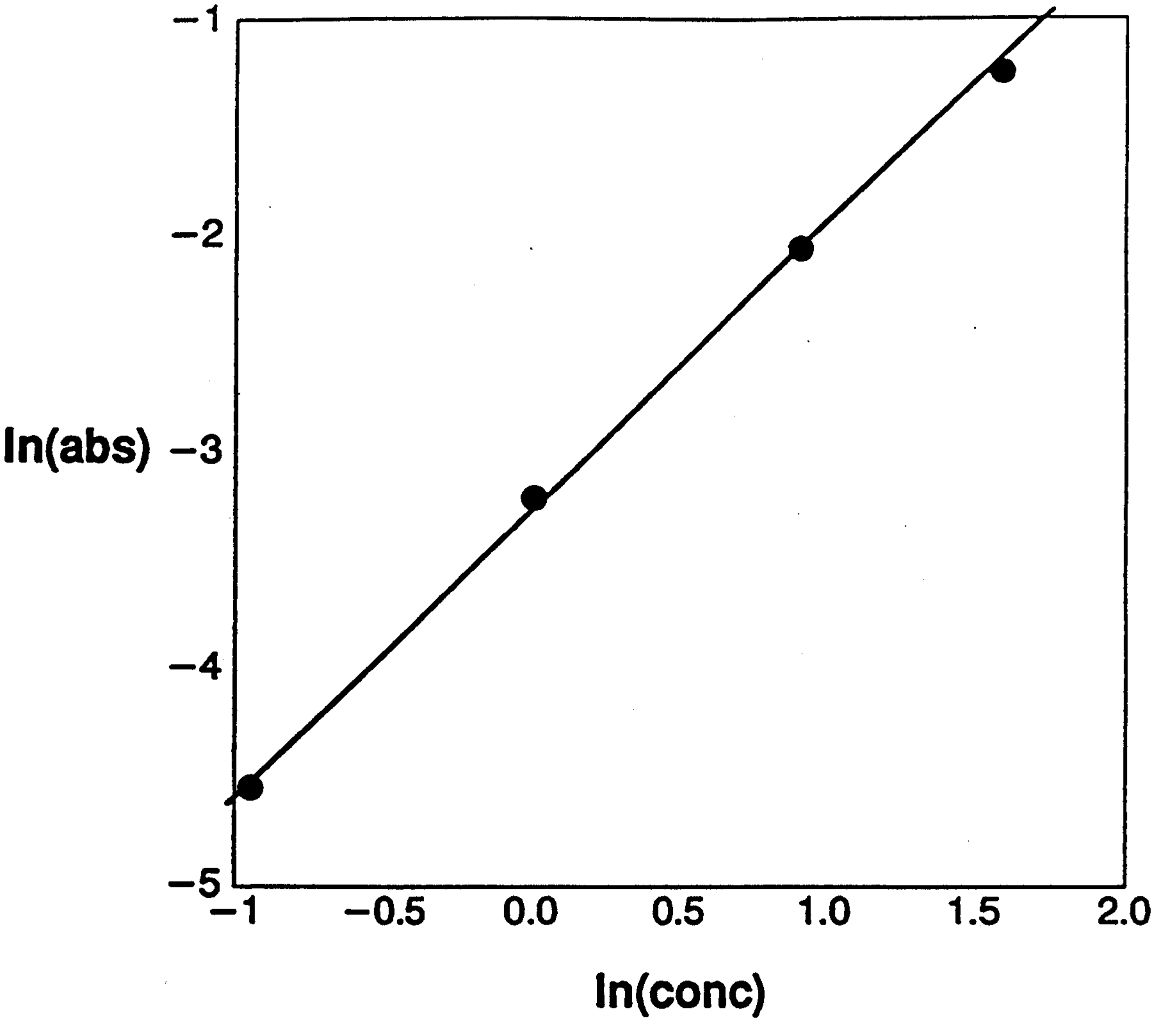


Figure 2.7. Absorbance vs. Concentration for R407C/Castrol SW32

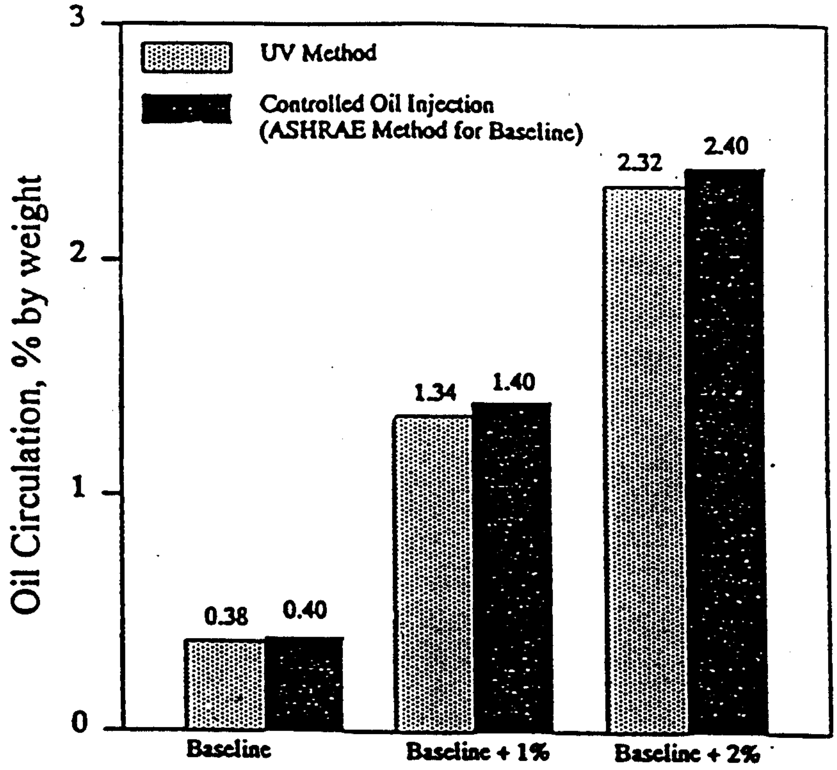
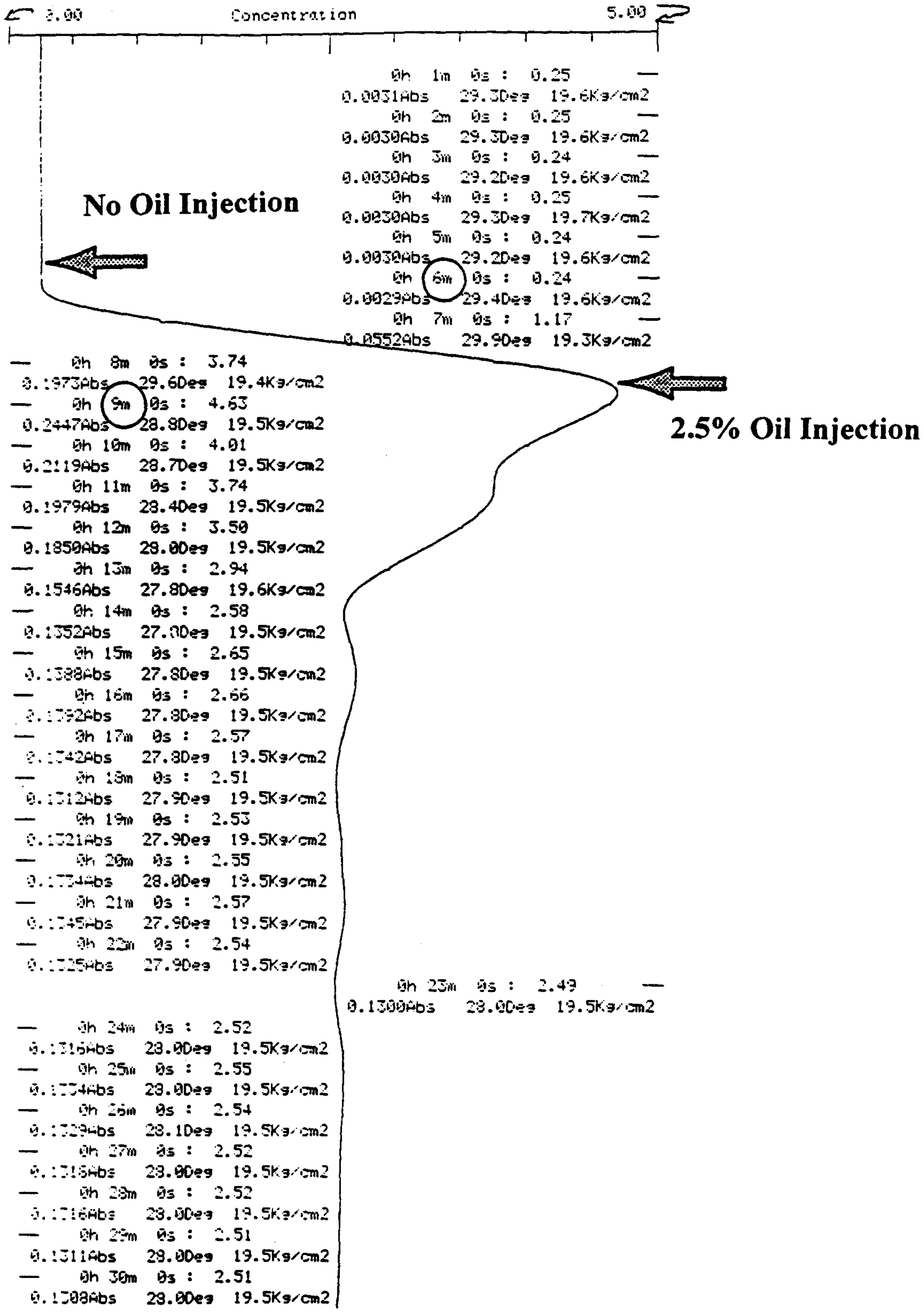


Figure 2.8. Compressor Oil Circulation Measurement with R134a/POE



```
       0h  1m  0s :  0.25        —
0.0031Abs   29.3Deg  19.6Kg/cm2
       0h  2m  0s :  0.25        —
0.0030Abs   29.3Deg  19.6Kg/cm2
       0h  3m  0s :  0.24        —
0.0030Abs   29.2Deg  19.6Kg/cm2
       0h  4m  0s :  0.25        —
0.0030Abs   29.3Deg  19.7Kg/cm2
       0h  5m  0s :  0.24        —
0.0030Abs   29.2Deg  19.6Kg/cm2
       0h  6m  0s :  0.24        —
0.0029Abs   29.4Deg  19.6Kg/cm2
       0h  7m  0s :  1.17        —
0.0552Abs   29.9Deg  19.3Kg/cm2
—      0h  8m  0s :  3.74
 0.1973Abs   29.6Deg  19.4Kg/cm2
—      0h  9m  0s :  4.63
 0.2447Abs   28.8Deg  19.5Kg/cm2
—      0h 10m  0s :  4.01
 0.2119Abs   28.7Deg  19.5Kg/cm2
—      0h 11m  0s :  3.74
 0.1979Abs   28.4Deg  19.5Kg/cm2
—      0h 12m  0s :  3.50
 0.1850Abs   28.0Deg  19.5Kg/cm2
—      0h 13m  0s :  2.94
 0.1546Abs   27.8Deg  19.6Kg/cm2
—      0h 14m  0s :  2.58
 0.1352Abs   27.8Deg  19.5Kg/cm2
—      0h 15m  0s :  2.65
 0.1388Abs   27.8Deg  19.5Kg/cm2
—      0h 16m  0s :  2.66
 0.1392Abs   27.8Deg  19.5Kg/cm2
—      0h 17m  0s :  2.57
 0.1342Abs   27.8Deg  19.5Kg/cm2
—      0h 18m  0s :  2.51
 0.1312Abs   27.9Deg  19.5Kg/cm2
—      0h 19m  0s :  2.53
 0.1321Abs   27.9Deg  19.5Kg/cm2
—      0h 20m  0s :  2.55
 0.1334Abs   28.0Deg  19.5Kg/cm2
—      0h 21m  0s :  2.57
 0.1345Abs   27.9Deg  19.5Kg/cm2
—      0h 22m  0s :  2.54
 0.1325Abs   27.9Deg  19.5Kg/cm2
       0h 23m  0s :  2.49        —
0.1300Abs   28.0Deg  19.5Kg/cm2
—      0h 24m  0s :  2.52
 0.1316Abs   28.0Deg  19.5Kg/cm2
—      0h 25m  0s :  2.55
 0.1334Abs   28.0Deg  19.5Kg/cm2
—      0h 26m  0s :  2.54
 0.1329Abs   28.1Deg  19.5Kg/cm2
—      0h 27m  0s :  2.52
 0.1316Abs   28.0Deg  19.5Kg/cm2
—      0h 28m  0s :  2.52
 0.1316Abs   28.0Deg  19.5Kg/cm2
—      0h 29m  0s :  2.51
 0.1311Abs   28.0Deg  19.5Kg/cm2
—      0h 30m  0s :  2.51
 0.1308Abs   28.0Deg  19.5Kg/cm2
```

Figure 2.9. Ultraviolet Oil Concentration Meter Test Results

TASK 3 - DATA COLLECTION, ANALYSIS, RESULTS, REVIEW, AND FORMULATION OF PROGRAM CONCLUSIONS

The objective of this task was to collect the data generated in the dynamic test apparatus and to analyze the data to prepare clear results so that understandable conclusions could be drawn.

This section is organized in the following manner. First, a listing of the tests that were conducted is provided, then typical data for a few test runs are presented and explained to orient the reader. Finally, a summary of the test results is provided and described, and key program conclusions are presented.

A representative 2.5 ton heat pump with a scroll compressor modified to allow additional injection of oil was used for this study. The system was run at DOE-A (95 F outdoor / 80 F indoor) for the cooling mode tests and at the minimum supply temperature available for the heating mode tests (25 F outdoor / 70 F indoor). A total of 52 tests were run. A summary of the tests numbers is given in Table 3.1.

Table 3.1 Tests Run with Test Number

Refrigerant/Lubricant Test Series	**Actual Chronological Test Run Number**			
	Cooling	**Cooling Inj.***	**Heating**	**Heating Inj.***
1) R22-3GS	1, 2, 35	41, 42	3, 4, 36, 40	37, 38, 39
2) R407C-3GS	5, 43,44	45	6, 46, 47	48, 49
3) R407C-1GS	50, 51	†	52	†
4) R407C-SW32	7, 8	9, 10	11, 12, 13	14
5) R407C-SW68	15, 16	17, 18	19, 20, 25, 26	21, 22, 23, 24
6) R407C-RL32S	27, 28	31, 32,33	29, 30, 34	†
* additional oil injected into compressor † no test run				

The UV oil concentration meter was not calibrated for the mixture of R-407C and 3GS mineral oil since the mixture was immiscible. However, the UV meter still generated useful data using the R22-3GS calibration curve. No calibration data were available for the mixture of R-407C and 1GS mineral oil, which was again an immiscible mixture. The determination of oil return problems for this mixture was made by visual indications only.

Description of Typical Data

For each of these tests, a set of four graphs was generated, describing the system temperatures, system pressures, air side temperatures, and oil concentration and vapor velocity. A typical set of cooling mode and heating mode test results are given below. In addition, visual observations of the oil sump level, oil level in the appropriate manifolds, changes in oil characteristics and oil concentration meter stability (when appropriate) were noted. These are noted in Table 3.2.

Cooling Mode Tests

Figures 3.1, 3.2, 3.3, and 3.4 show the results of a set of typical cooling mode test runs. In this case, the refrigerant-lubricant mixture is R-407C and Castrol SW68. Figure 3.1 shows that the compressor inlet and discharge temperatures rise slightly with decreasing velocity, while Figure 3.2 shows that the system pressures all stay relatively constant over the course of the test. The air side temperatures also stay relatively constant, as shown in Figure 3.3. Refer to Figure 2.1 for the location of all instrumentation. The indoor unit exit temperature remains constant at about 58 F. Figure 3.4 shows the UV oil concentration meter results on left axis, along with the velocity in the vapor lines on the right axis. This graph shows the effect of decreasing vapor velocity on the oil concentration in the liquid line. The results demonstrate that oil concentration fluctuations start to occur with two valves open, and become much worse with three valves open. In the cooling mode, these fluctuations are an indication of poor oil return. This is because the indoor unit manifold is alternately flooding partly and then entirely, with the drop occurring when one or more of the vapor return lines becomes blocked. This was described previously in further detail in Task 2, Subtask 2A (design of test apparatus), and Subtask 2E (data analysis). What is unknown is the reason for the increase in the average oil concentration as the vapor velocity is lowered.

Along with the data logged and displayed on the typical four figures for each test run, additional indications of oil return were the visual observations of the liquid level inside the vapor manifolds and the oil level inside the sump. The observed level of the indoor unit manifold on both sides is shown in Table 3.2. The table indicates that the vapor velocity started to become insufficient between two and three valves open for both tests. At four valves open, the manifold had flooded completely. On Figure 3.4, the change from two to three valves open results in a dramatic increase in the level of the oil concentration fluctuations. The oil sump level did not change appreciably during the course of the two tests. Figure 3.4 also shows the vapor velocity corresponding to each valve opening condition. Based on the vapor velocity levels calculated, these became the numbers to compare to the ASHRAE guidelines.

Table 3.2 Indoor Unit Manifold Liquid Levels During Cooling Mode Tests

	Test 15 Liquid Level Noted		**Test 16 Liquid Level Note**	
# of lines open	**Left side**	**Right side**	**Left side**	**Right side**
1 (❍●●●)	1/4	1/4	1/4	1/8
2 (❍●●❍)	1/4	1/8	1/4	1/8
3 (❍❍●❍)	3/4	1/8	3/4	1/8
4 (❍❍❍❍)	Full	1/8	Full	1/8
❍ - Open vapor line ● - Closed vapor line				

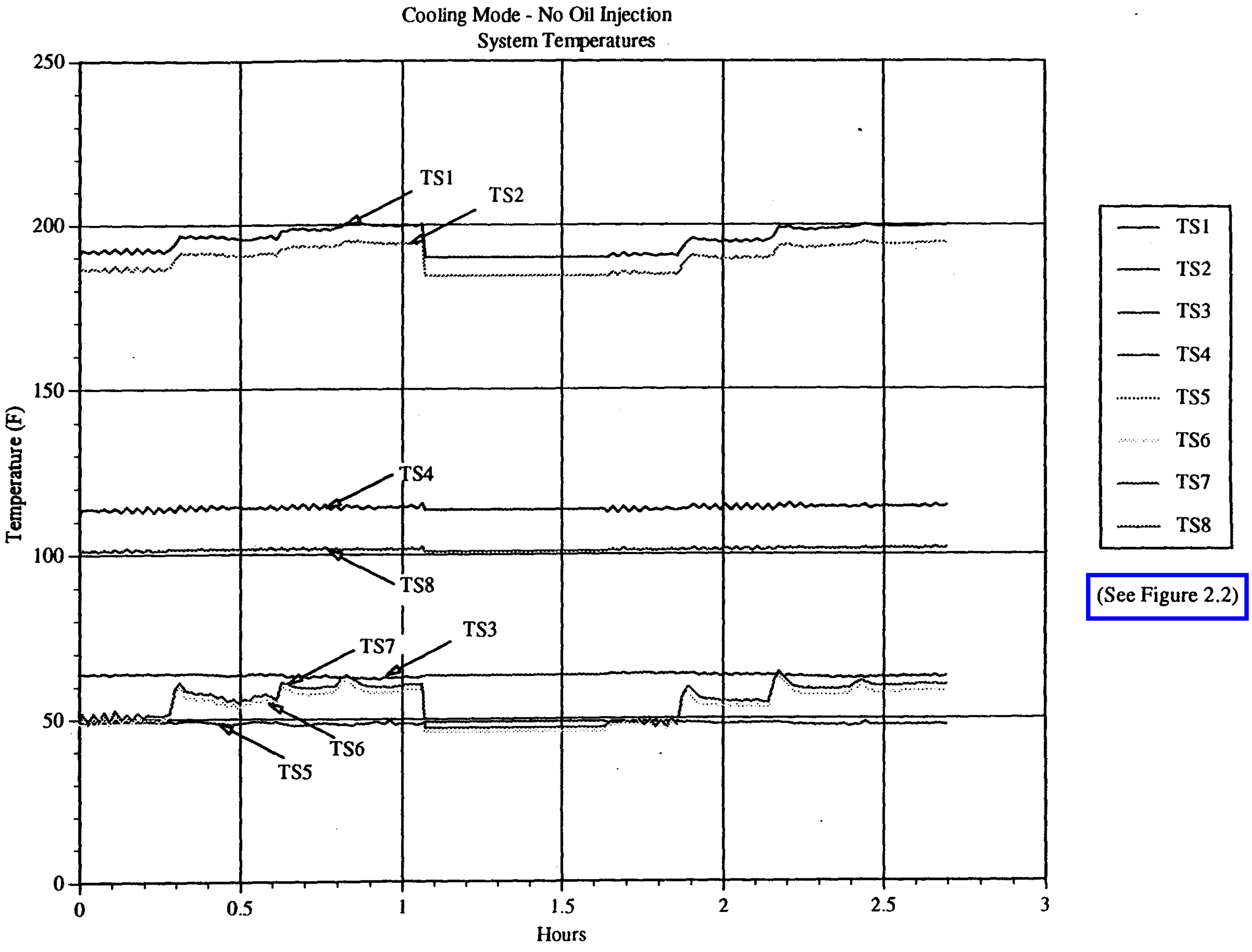


Figure 3.1. Heat Pump R407C - Castrol SW68 Oil Circulation Tests (15 & 16)

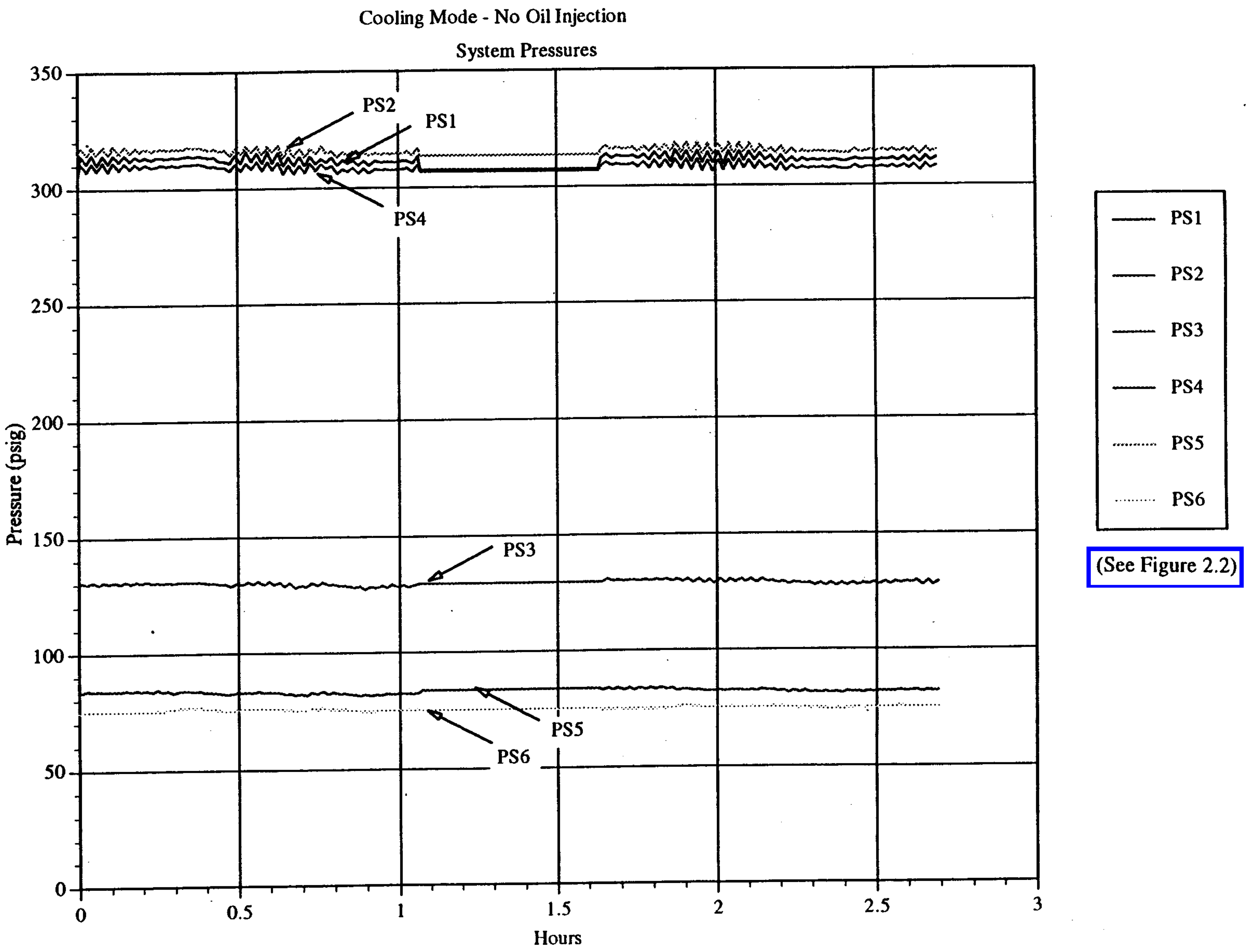


Figure 3.2. Heat Pump R407C - Castrol SW68 Oil Circulation Tests (15 & 16)

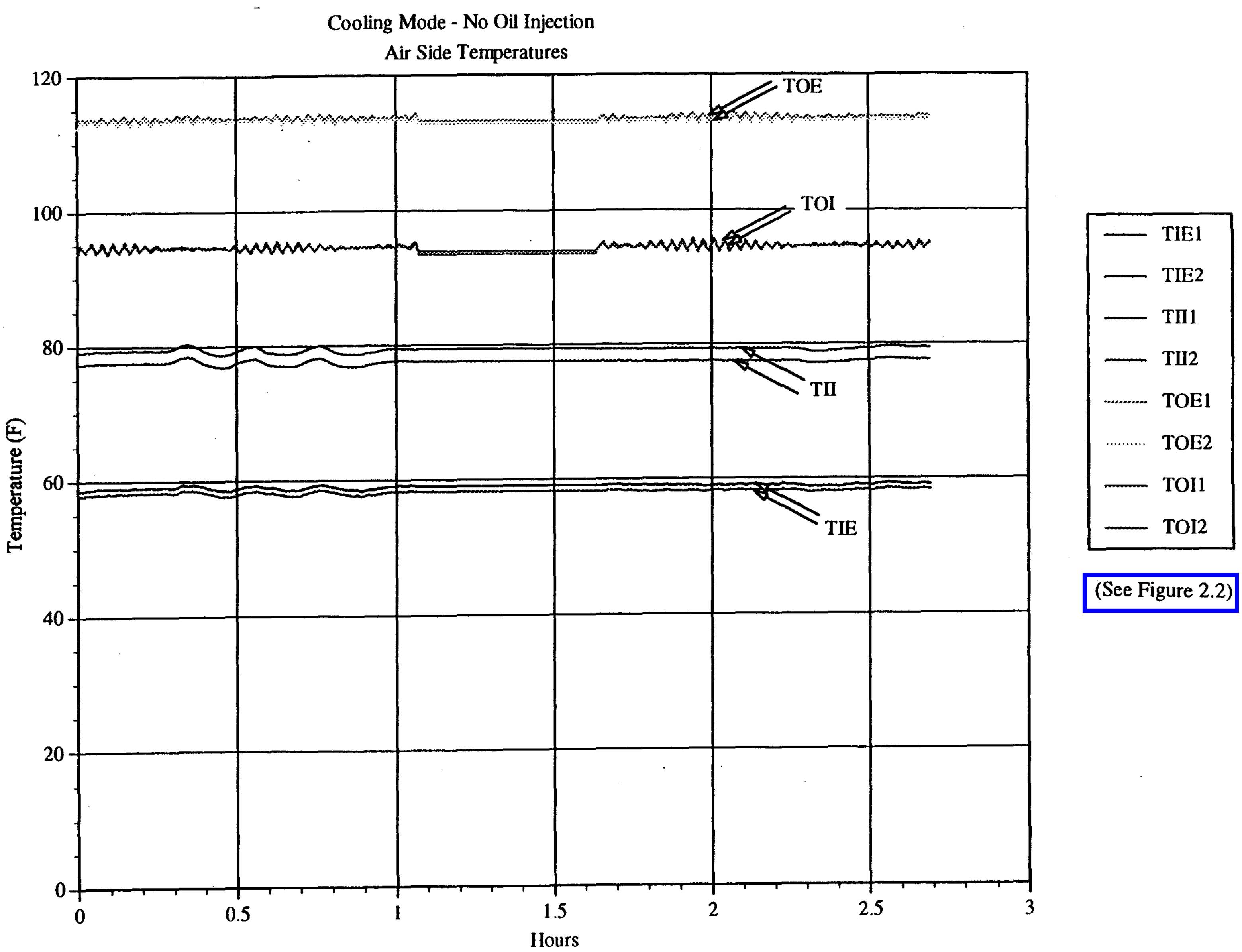


Figure 3.3. Heat Pump R407C - Castrol SW68 Oil Circulation Tests (15 & 16)

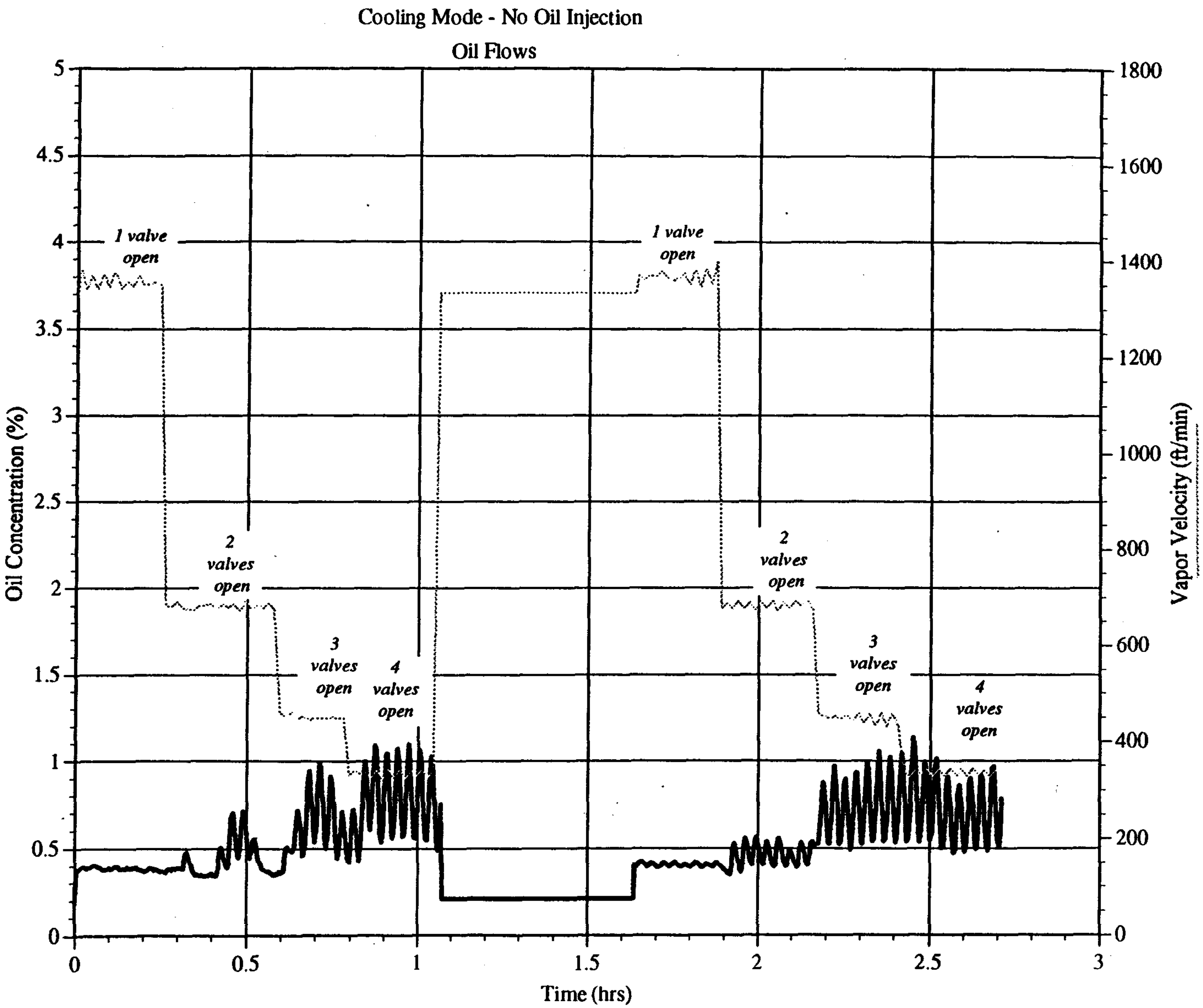


Figure 3.4. Heat Pump R407C - Castrol SW68 Oil Circulation Tests (15 & 16)

Heating Mode Tests

Typical results for a set of heating mode tests are presented in Figures 3.5, 3.6, 3.7, and 3.8. Once again, the refrigerant-lubricant mixture is R407C and Castrol SW68. These figures exhibit time scales that are considerably longer than the cooling mode figures, although the actual tests took about the same amount of time. The tests actually took place between 1.2 and 2.4 hours, and 4.1 and 5.1 hours on the figures. The rest of the time was spent in setting the test conditions, such as suction superheat (set by charge), correctly and performing defrost cycles. The charge level in the system was difficult to maintain exactly, because of the dynamics of the system.

In the heating mode, hot gas exits the compressor and enters the outdoor unit manifold, where it is divided into the open vapor lines. The gas returns on the indoor side manifold, where it flows to the condenser. However, since all vapor lines are open on the indoor side (the valves are actually located on the outdoor unit manifold), the high pressure vapor can and does fill the non-flowing vapor lines. As the vapor rises, it cools and condenses. If the condensation occurs above the outdoor unit manifold, the liquid will flow down and fill the closed vapor line just above the shut-off valve. This is illustrated in Figure 3.9. This continual condensation gradually reduces the amount of circulating charge, increases the suction superheat, and decreases the subcooling level. When the refrigerant starts to exit the condenser in a two-phase state, the UV-oil concentration meter becomes unreliable. This is because the UV meter works by absorption of light passing through a homogeneous liquid stream. When bubbles start to appear in the liquid stream, somewhat more light is transmitted through the liquid. The more light that is transmitted, the less oil is theoretically in the liquid sample. This explains the negative oil concentration readings shown in Figure 3.8. To minimize the effect of this process, all manifold valves were opened briefly (to empty them) when opening another manifold line in the testing sequence.

Figure 3.5 shows the system temperatures in the heating mode. As mentioned previously, the tests were actually run between 1.2 and 2.4 hours (test 25), and 4.1 and 5.1 hours (test 26). During these times, it appears that the condenser inlet temperature drops off a number of times and then returns to a somewhat lower steady state level. The initial drop-off in temperature is caused by the return of the condensed liquid trapped out in the vapor lines. At about 2.2 hours (during test 25), the evaporator air fan flowrate was reduced (to 250 cfm from 2000 cfm) to simulate a blocked fan. At 4.9 hours (test 26) the flowrate was again reduced (this time to only 1000 cfm). The evaporator temperatures dropped between 10 and 15 degrees on average, while the compressor discharge temperature increased by about the same amount. The system pressures are shown in Figure 3.6. The figure indicates a slight increase in pressure throughout the system with more vapor lines open. The pressure also drops slightly throughout the system at the reduced evaporator air flowrate condition. The air side temperatures are presented in Figure 3.7. The outdoor temperature is about 25F, while the indoor temperature is about 70F. These temperatures are representative of all of the heating mode tests. The air side temperatures show little signs of being affected by the vapor line velocities, although the reduction in the evaporator flowrate does drop both the indoor and outdoor air side exiting temperatures. The oil concentration in the liquid line and the vapor velocities are plotted in Figure 3.8. The figure shows that the oil concentration level remains relatively constant (about 0.5 %) throughout the test times until the evaporator cfm is lowered. At this point, the oil

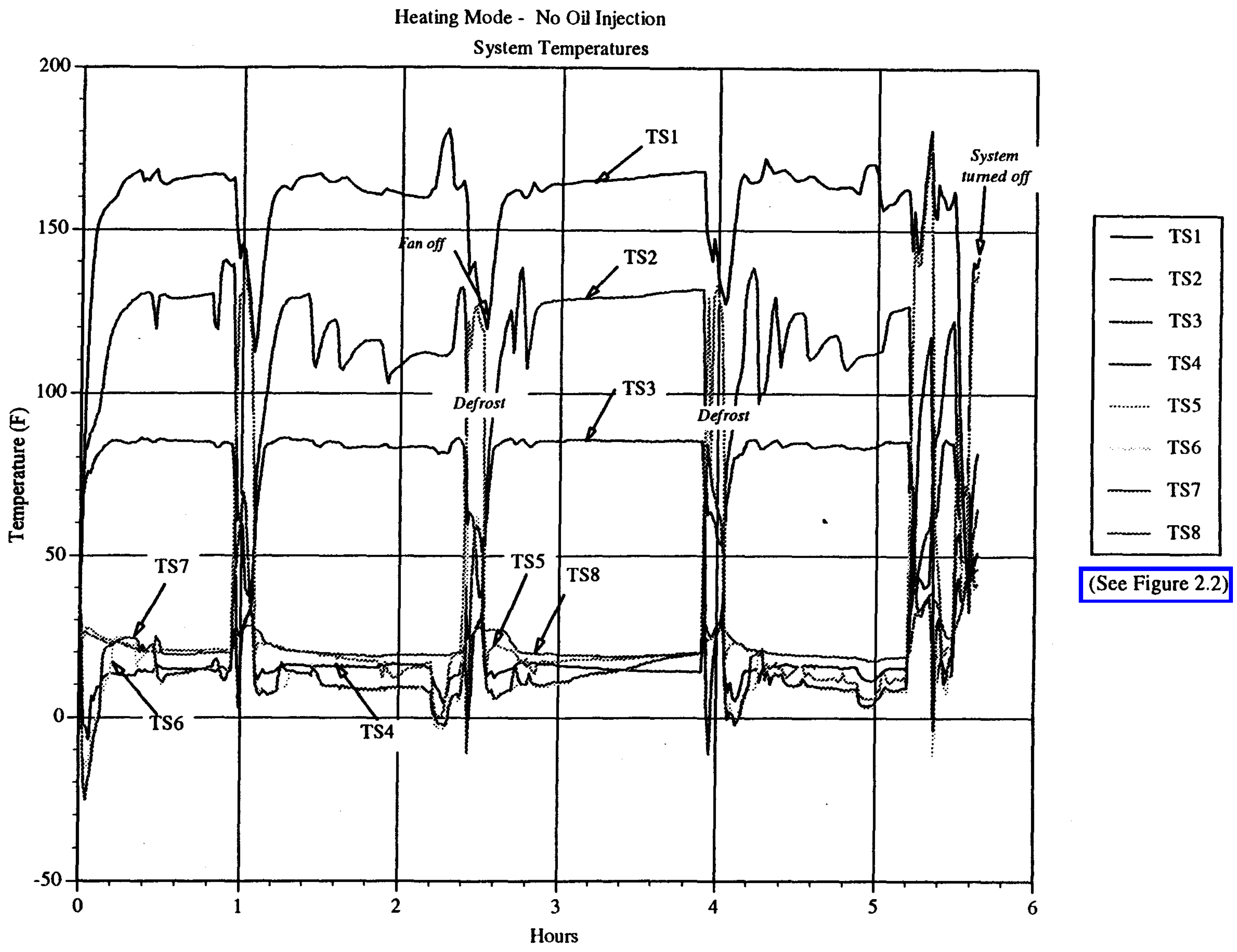


Figure 3.5. Heat Pump R407C - Castrol SW68 Oil Circulation Tests (15 & 16)

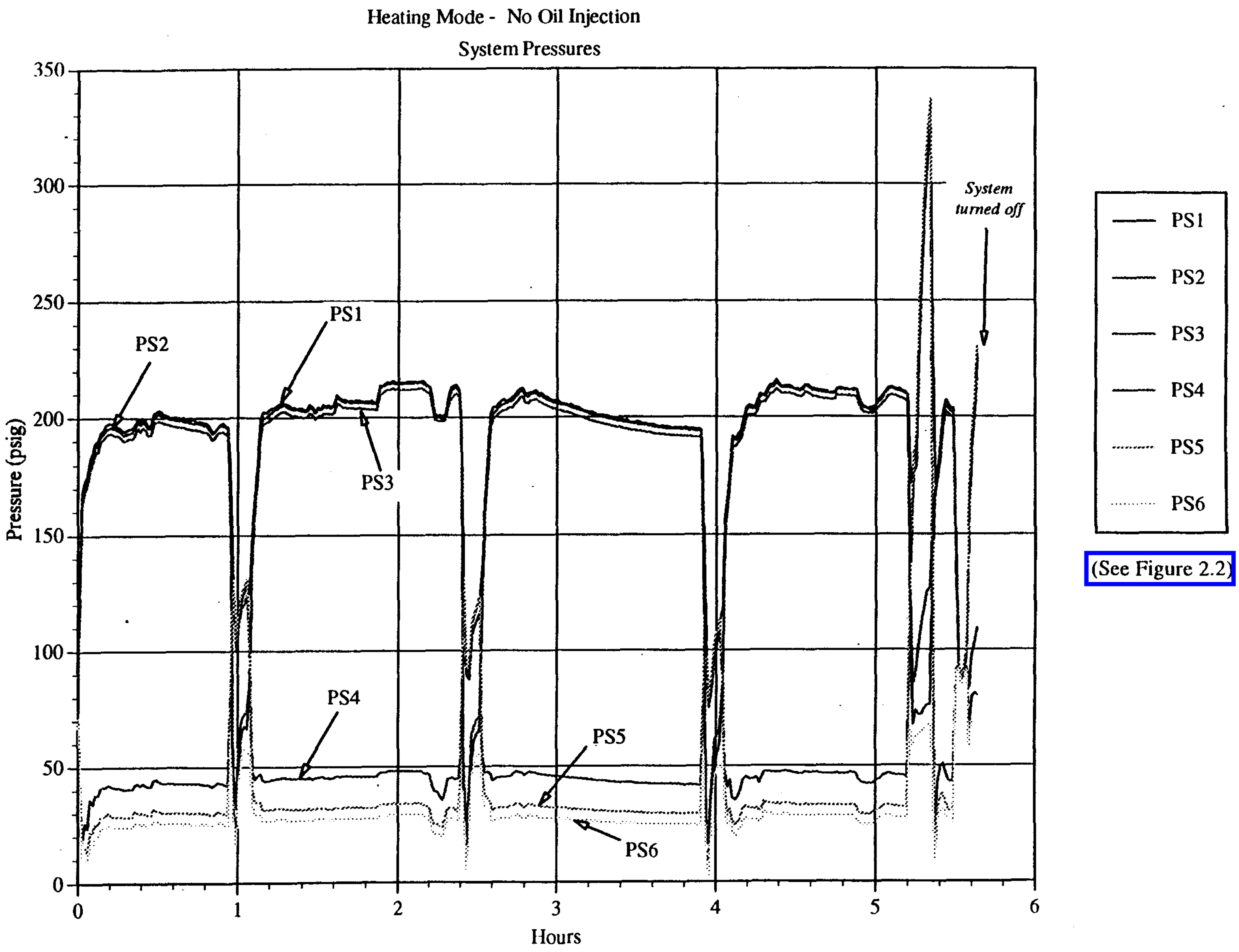


Figure 3.6. Heat Pump R407C - Castrol SW68 Oil Circulation Tests (15 & 16)

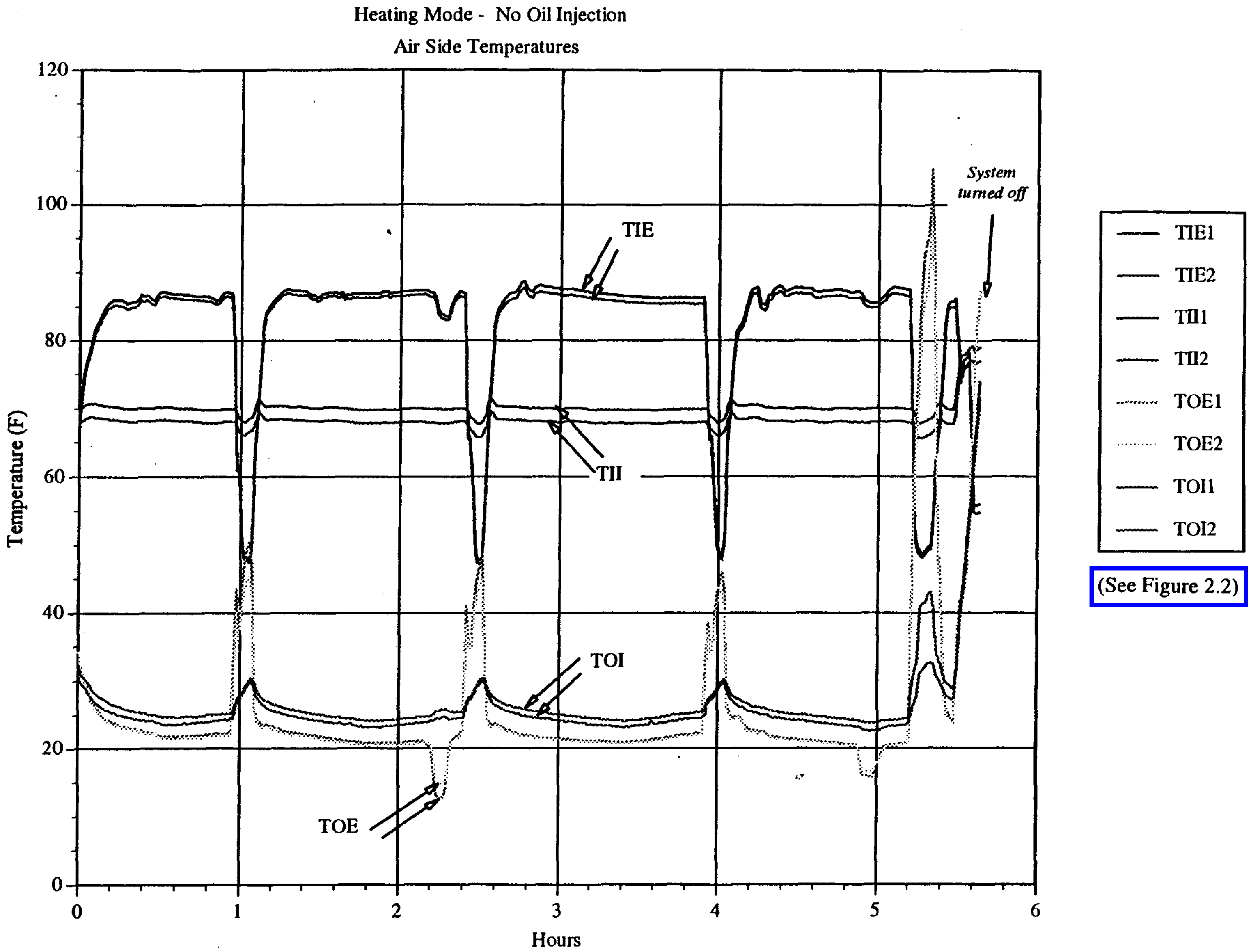


Figure 3.7. Heat Pump R407C - Castrol SW68 Oil Circulation Tests (15 & 16)

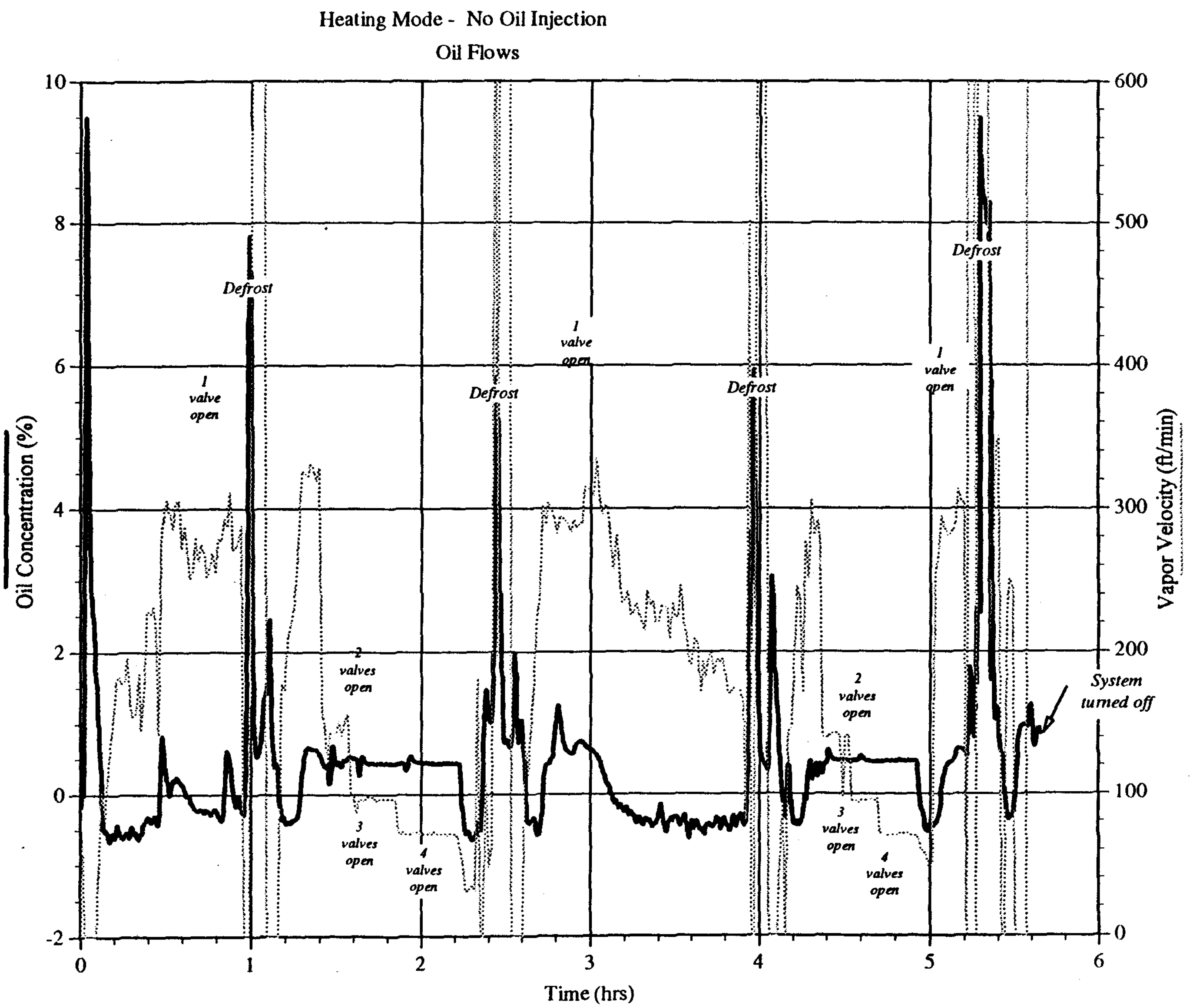


Figure 3.8. Heat Pump R407C - Castrol SW68 Oil Circulation Tests (15 & 16)

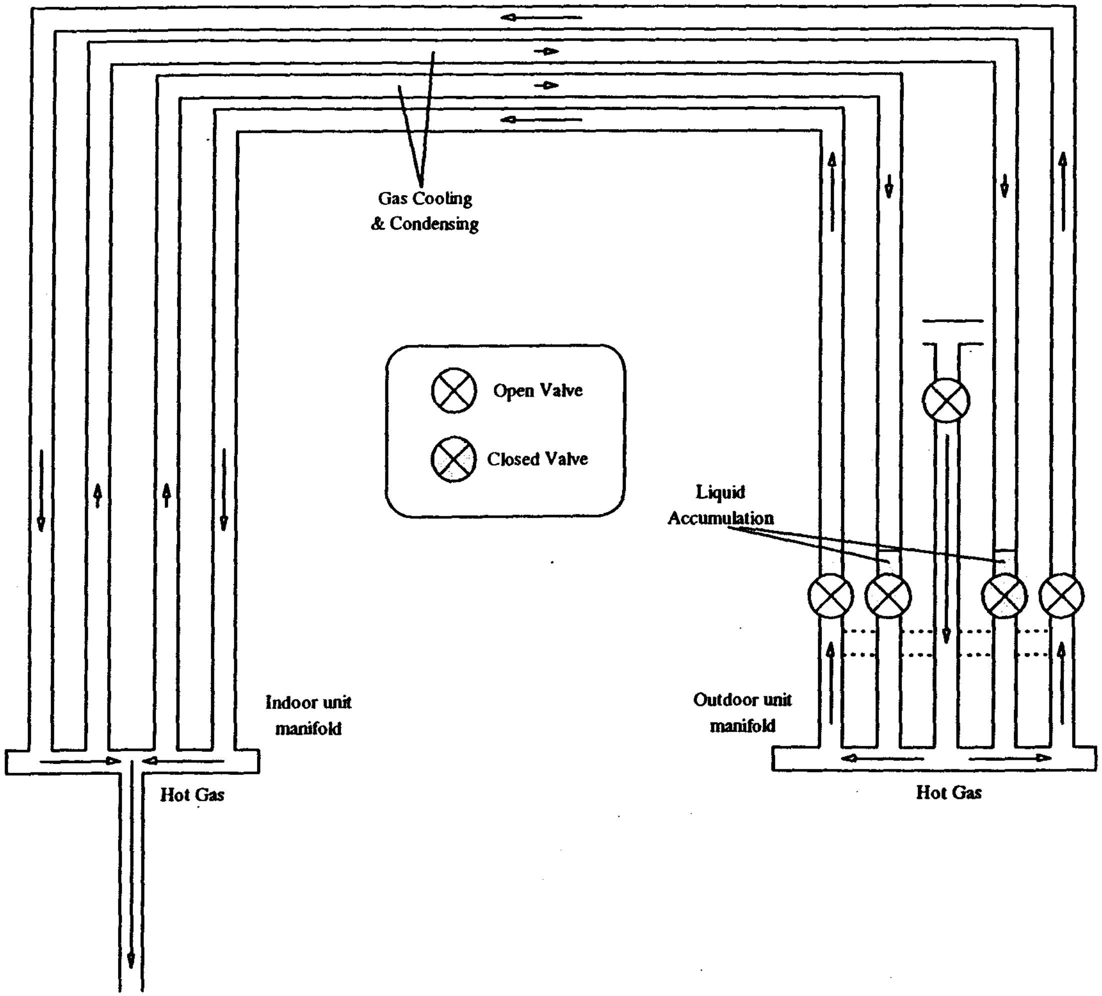


Figure 3.9. Heating Mode Manifold Accumulation

concentration quickly drops below zero. It is not known whether this sharp drop-off is caused by vapor bubbles in the liquid line (two-phase flow) or by an actual drop in oil concentration. However, because of the way the UV oil concentration meter is calibrated, readings below zero are not significant; that is, oil flow is occurring in the system. The slight increases in oil concentration right after another valve is opened are caused by the same slug of liquid (trapped out in the closed-off vapor lines) returning to the vapor line.

As in the cooling mode, the additional indications, and again probably the most reliable, of poor oil return are the visual observations of the manifolds and the oil sump level. In this case, the liquid level in the outdoor unit manifold is important, although visual observations of the flow inside the returning manifold (indoor) are also meaningful. Table 3.3 shows the liquid level for the outdoor unit manifold as observed during the course of the two tests. The table shows that the velocity started to become insufficient with two valves open, with the manifold flooding completely at three valves open. In the second test, the velocity again started to become insufficient with two lines open. This time, however, all four lines were open before the manifold flooded completely. As in the cooling mode test, the oil sump level did not change significantly during the course of the test.

Table 3.3 - Outdoor Unit Manifold Liquid Levels During Heating Mode Tests

# of lines open	**Test 25**		**Test 26**	
	Left Side	**Right Side**	**Left Side**	**Right Side**
1 (●●❍●)	3/8	1/4	1/4	1/4
2 (●❍❍●)	3/8	1/2	5/8	Fog*
3 (❍❍❍●)	7/8	Full	3/4	3/4
4 (❍❍❍❍)	Full	Full	Full	5/8
- Open vapor line - Closed vapor line *accurate indication of level not possible				

Test Results Summary

As previously indicated, more than 52 tests were actually run with extensive data collection and interpretation. In fact, several tests were run with scroll compressors manufactured by different suppliers. One series of tests were run with HCFC-22 and mineral oil (3GS) over the two cooling and two heating mode conditions (w and w/o oil injection) when a compressor malfunction occurred, which led to a fracture of the scrolls. Other tests were sometimes run several times to insure that consistent results were obtained. Still other tests, for example with ICI RL68S, were never run because of time constraints and our belief that no new information would be obtained.

Nonetheless, more than 40 consistent sets of test data were obtained, thoroughly analyzed and utilized to formulate results and conclusions of the program. In an attempt to provide some concise presentation of the results, Tables 3.4, 3.5, and 3.6 were prepared. Table 3.4 summarizes the six combinations of refrigerants and lubricants tested and provides an indication of when the first poor oil return situation was noted, either by visual observation of the manifold flooding, change in the oil sump level, or via the dropping or fluctuation of the oil concentration measurement. Table 3.5 contains the calculated gas line velocities that bracket adequate oil return, arid for completeness, all the test matrix velocity data and oil circulation information are shown in Table 3.6.

Sometimes air flow reductions across the evaporator coil were needed to produce a poor oil return situation in addition to opening all valves (operation at minimal velocities). In addition, in some instances where oil return was marginal, the interpretation could vary, depending upon the observer and their daily perspective. For example, estimates of the oil return gathering and other flooding phenomena could lead one researcher to infer poor oil return, while another would indicate marginal return. Nevertheless, the data shown in Tables 3.4 and 3.5 shows a good consensus among program researchers as to when poor or zero oil return problems were indicated.

Data Observations

The results in Table 3.4, indicate that with the baseline refrigerant/lubricant pair, HCFC-22 and 3GS mineral oil, poor oil return was observed (by the flooding manifold) when four valves were opened and the evaporator fan flow was reduced 50% (bold print in Table 3.4) during cooling mode with no oil injection. At that time both the sump oil level and the UV meter reading were steady. The vapor velocity was calculated to be less than 315 ft/min. Under the same conditions, but with oil injection, manifold flooding was observed when four valves were opened and fan flow was at 100%. For this case, a vapor velocity of 330 ft/min. was calculated. During heating and no oil injection, three open valves caused an observed oil circulation problem at a calculated velocity of 108 ft/min. With oil injection during heating conditions, the oil sump level was observed to constantly drop for all valve settings, indicating that the system may not be designed to accommodate the injected flow rates. This could be attributed to low saturation temperatures and excessive quantities of oil in the evaporator and accumulator.

By scanning the Table 3.4 results, the various indicators of oil return problems can be seen to be dominated by the visual observation of oil collecting in the manifolds, and only a few cases due to sump oil level dropping or oil concentration meter reading. In the cooling mode, this is to be expected, since manifold flooding leads to the closing of one vapor line and therefore higher velocity and adequate oil return. In the heating mode, the UV meter reading was consistently indicating an oil return problem at a lower minimum velocity than the visual manifold observation and the sump oil level tended to remain constant. It appears likely that oil return was achieved in the heating mode after manifold flooding in a similar manor to the cooling test.

In Table 3.5, the Table 3.4 observations have been converted into flow velocities in ft/min., where oil return is good (OK) and poor (Not OK). The velocities calculated using the Reference 3 guidelines are indicated as Model. The flow velocities based on visual observations (either oil sump or manifold flooding) and the velocities based on UV meter trends are indicated as visual and UV meter, respectively. The visual and UV meter velocities were calculated based on the number of vapor lines that were open (in parenthesis).

Based on the Table 3.5 data, a number of general observations can be made. These include:

1) Cooling Mode (no oil injection)

a) All six refrigerant/lubricant combinations would have good oil management if approximately the same minimum velocities, approximately 360 to 400 ft/min. are maintained. The visual observation (usually flooding of the manifold) would show all combinations have about the same range of flow velocities between good and poor oil management (i.e. from 400 ft/min. down to 340 ft/min.). The data further indicate that the immisible refrigerant/lubricant combinations, R-407C with 1GS and 3GS are as viable as HCFC-22 and 3GS or the HFCs with POEs.

b) The model predicted flow velocities are all relatively close to one another (between 355 ft/min. and 384 ft/min.).

c) In addition, the UV meter would predict that much higher minimum velocities have to be maintained to insure good oil return characteristics than those determined from visual observation.

2) Cooling Mode (with oil injection)

a) This set of tests generally indicate somewhat higher minimum flow velocities and generally higher flow velocity indications with the UV meter.

b) Once again, the model predictions for flow velocity appear to be bracketed by the range of flow velocities between good and poor oil management conditions.

c) Conservative values of good oil management would appear to be similar to that for no oil injection.

3) Heating Mode (no oil injection)

a) The test summary data would indicate close correspondence between the flow velocities predicted by the visual observations (flooding) and the UV meter. Generally the values for good oil management are 100 ft/min. or below.

b) The Ref. 3 guidelines predict that flow velocities of 240 ft/min or above should be satisfactory.

4) Heating Mode (with oil injection)

a) The test data would indicate two to three times higher flow velocities with oil injection based on flooding visualization, while the UV meter values remain at 100 ft/min or less.
b) The guideline predictions of flow velocity are now bracketed by the visual results.

Table 3.4. Summary of Oil Return Problems for Refrigerant and Lubricant Combinations

Bold face type shows first indicator of oil return

Operating Mode	***Nominal Oil Circulation Rate (%) from UV***	***First Indication of Oil Return Problem***			***Potential Cause of Oil Return Problem***	
		Manifold Flooding	***Oil Sump Level***	***UV Oil Meter Reading***	***Low Vapor velocity***	***Low Saturat. Temp.***
R22/	Suniso 3GS					
Cooling	.3	**1/2 FAN FLOW**	STEADY	STEADY	**< 315 FT/MIN**	
Cooling + oil inj.	.8	**4 OPEN**	STEADY	STEADY	**330 FT/MIN**	
Heating	.3	**3 OPEN**	STEADY	STEADY	**108 FT/MIN**	
Heating + oil inj.	.8	1 OPEN	**DROPPING**	STEADY		5° F
R407C/	Suniso 3GS					
Cooling	N/A	**4 OPEN**	STEADY	STEADY	**280 FT/MIN**	
Cooling + oil inj.	N/A	1 OPEN	**DROPPING**	STEADY		**40° F**
Heating	N/A	**3 OPEN**	STEADY	STEADY	**79 FT/MIN**	
Heating + oil inj.	N/A	1 OPEN	**DROPPING**	STEADY		**5° F**
R407C/	Suniso 1GS					
Cooling	N/A	**4 OPEN**	STEADY	STEADY	**330 FT/MIN**	
Cooling + oil inj.	N/A	N/A	N/A	N/A		
Heating	N/A	**3 OPEN**	STEADY	STEADY	**82 FT/MIN**	
Heating + oil inj.	N/A	N/A	N/A	N/A		
R407C/	Castrol SW32					
Cooling	.8	3 OPEN	STEADY	**UNSTEADY**	**455 FT/MIN**	
Cooling + oil inj.	3.5	**4 OPEN**	STEADY	STEADY	**330 FT/MIN**	
Heating	.8	**4 OPEN**	STEADY	STEADY	**70 FT/MIN**	
Heating + oil in'.	6.0	**2 OPEN**	STEADY	STEADY	**142 FT/MIN**	
R407C/	Castrol SW68					
Cooling	.45	3 OPEN	STEADY	**UNSTEADY**	**445 FT/MIN**	
Cooling + oil inj.	3.5	**4 OPEN**	DROPPING	UNSTEADY	**305 FT/MIN**	
Heating	.5	**4 OPEN**	STEADY	STEADY	**67 FT/MIN**	
Heating + oil in'.	3.5	**1 OPEN**	STEADY	STEADY	**290 FT/MIN**	
R407C/	ICI RL32S					
Cooling	3.5	3 OPEN	STEADY	**UNSTEADY**	**455 FT/MIN**	
Cooling + of inj.	4.25	**4 OPEN**	STEADY	STEADY	**330 FT/MIN**	
Heating	3.5	**3 OPEN**	STEADY	STEADY	**95 FT/MIN**	
Heating + oil inj.	N/A	N/A	N/A	N/A		

Table 3.5. Minimum Velocities Needed for Oil Return Problem

Refrigerant	Oil Return Status	Cooling			Cooling Injection			Heating			Heating Injection		
Oil		Visual	UV Meter	Model[a]	Visual	UV Meter	Model[a]	Visual	UV Meter	Model[a]	Visual	UV Meter	Model[a]
R22	OK	$315^{(4)}$	$315^{(4)}$	-	$430^{(3)}$	*3	-	$158^{(2)}$	$108^{(3)}$	-	$312^{(1)}$	$105^{(3)}$	-
Suniso 3GS	Not OK	*1	*1	355	$320^{(4)}$	$1325^{(1)}$	377	$108^{(3)}$	$80^{(4)}$	245	$154^{(2)}$	$78^{(4)}$	250
R407C	OK	$450^{(3)}$	*2	-	$675^{(2)}$	*2	-	$130^{(2)}$	*2	-	$262^{(1)}$	*2	-
Suniso 3GS	Not OK	$280^{(4)}$	*2	375	$435^{(3)}$	*2	375	$79^{(3)}$	*2	234	$115^{(2)}$	*2	242
R407C	OK	$450^{(3)}$	*2	-	N/A	N/A	N/A	$120^{(2)}$	*2	-	N/A	N/A	N/A
Suniso 1GS	Not OK	$330^{(4)}$	*2	384	N/A	N/A	N/A	$82^{(3)}$	*2	253	N/A	N/A	N/A
R407C	OK	$455^{(3)}$	$685^{(2)}$	-	$445^{(3)}$	$655^{(2)}$	-	$94^{(3)}$	$70^{(4)}$	-	$275^{(1)}$	$90^{(3)}$	-
Castrol SW32	Not OK	$340^{(4)}$	$455^{(3)}$	375	$330^{(4)}$	$330^{(4)}$	388	$70^{(4)}$	$60^{(4+)}$	267	$142^{(2)}$	$70^{(4)}$	237
R407C	OK	$450^{(3)}$	$685^{(2)}$	-	$430^{(3)}$	*3	-	$97^{(3)}$	$97^{(3)}$	-	*3	$95^{(3)}$	-
Castrol SW68	Not OK	$330^{(4)}$	$445^{(3)}$	375	$305^{(4)}$	$1370^{(1)}$	378	$67^{(4)}$	$50^{(4+)}$	242	$290^{(1)}$	$68^{(4+)}$	245
R407C	OK	$340^{(4)}$	$685^{(2)}$	-	$465^{(3)}$	$705^{(2)}$	-	$150^{(2)}$	$75^{(4)}$	-	N/A	N/A	N/A
ICI RL32S	Not OK	*1	$455^{(3)}$	375	$330^{(4)}$	$320^{(4)}$	375	$95^{(3)}$	$45^{(4+)}$	240	N/A	N/A	N/A

[a] Based on ASHRAE guidelines (Ref. 3)

*1 - No Accurate Measurement of Velocity Available

*2 - Not Calibrated for Refrigerant - Lubricant Combination

*3 - No Upper Bound on Velocity Available

(n) - Number of Vapor Lines Open

Table 3.6. Complete Test Matrix Flow Velocity Details

Refrig -Oil Combo (Test Number)	Manifold flooding 7/8 (1/2)	Test	1 open velocity (ft/min)	1 open oil conc. (%)	2 open velocity (ft/min)	2 open oil conc. (%)	3 open velocity (ft/min)	3 open oil conc. (%)	4 open velocity (ft/min)	4 open oil conc. (%)	* open velocity (ft/min)	* open oil conc. (%)
R22-3GS											1000 cfm	
35	750 (4)	Cooling 1	1310	0.3	655	0.2-0.3	430	0.2-0.3	315	0.2-0.3	*1	0.5-0.6
41	4 (3)	Cooling 1 inj	1310	1.4-1.6	640	1.1-1.5	415	1.0-1.6	310	1.1-1.6	*1	1.8-2.0
42	4 (3)	Cooling 2 inj	1340	1.1-1.4	672	1.3-1.4	448	1.4-1.5	330	1.4-1.6	*1	1.9-2.0
36	3 (2)	Heating 1	310	0.2	158	0.1	108	0.1	80	0.0	50	-0.1
37	2	Heating 1 inj	310	0.0-1.1	155	0.1-1.1	100	0.4-1.5	x	x	x	x
38	2	Heating 2 inj	315	0.1-0.8	155	0.3-0.9	105	0.1-0.4	78	0.1-0.2	x	x
39	2	Heating 3 inj	310	0.1-0.9	152	0.3-1.1	105	0.1-1.0	78	0.3-0.9	x	x
R407C-3GS											1000 cfm	
43	4 (3)	Cooling 1	1385	0.2	690	0.2-0.3	450	0.2-0.3	330	0.2-0.4	x	x
44	3	Cooling 2	1365	0.3	675	0.5-0.6	450	0.4-0.5	280	0.2-0.4	x	x
45	3	Cooling 1 inj	1360	0.5-0.7	675	0.9-1.0	435	1.1-1.3	320	1.1-1.2	x	x
46	3 (2)	Heating 1	(*2)	0.0	125	0.02	79	0.02	60	0.05	44	-0.1
47	3 (2)	Heating 2	255	0.1	130	0.1	87	0.06	65	0.05	60	0.04
48	2	Heating 1 inj	238	0.1-0.3	115	0.3-1.0	75	0.3-1.1	55	0.3-0.4	x	x
49	2	Heating 2 inj	262	0.3-0.8	132	0.3-0.9	87	0.6-0.7	65	0.4-0.5	x	x
R407C-1GS											1000 cfm	
50	4 (3)	Cooling 1	1350	0.10 (*3)	687	0.11 (*3)	450	0.13 (*3)	330	0.15 (*3)	x	x
51	4 (3)	Cooling 2	1345	0.0 (*3)	680	0.0 (*3)	445	0.0 (*3)	330	0.0 (*3)	x	x
52	3 (2)	Heating 1	250	0.0 (*3)	120	0.0 (*3)	82	0.0 (*3)	60	0.0 (*3)	x	x
R407C-SW32												
7	4 (3)	Cooling 1	1375	0.8	685	0.8	455	0.8-1.2	340	0.8-1.1	x	x
8	4	Cooling 2	1360	0.8	685	0.8	455	0.8-1.1	340	0.8-1.3	x	x
9	4 (3)	Cooling 1 inj	1350	1.5-1.9	680	1.6-1.9	450	1.4-2.2	330	1.4-2.5	x	x
10	4 (3)	Cooling 2 inj	1360	1.5-2.0	655	1.5-2.1	445	1.5-2.2	335	1.6-2.4	x	x
13	4 (3)	Heating 1	277	0.9	142	0.9	94	0.9	70	0.8	60	0.0
14	2 (1)	Heating 1 inj	275	5.8	142	4.3	90	3.6	70	1.2	x	x
R407C-SW68											1000 cfm	
15	4 (3)	Cooling 1	1345	0.4	685	0.4-0.7	450	0.5-1.0	330	0.6-1.1	x	x
16	4 (3)	Cooling 2	1365	0.4	680	0.4-0.6	445	0.5-1.0	335	0.5-1.1	x	x
17	4 (3)	Cooling 1 inj	1370	2.2-5.0	640	2.5-5.5	415	3.2-6.2	305	3.4-6.2	x	x
18	4 (3)	Cooling 2 inj	1320	2.2-3.5	660	2.2-4.3	430	2.8-5.0	320	2.8-5.6	x	x
19	4	Heating 1	285	0.3	140	0.2	90	0.2	72	0.2	65	-0.4
25	3	Heating 2	325	0.6	145	0.5	97	0.4	67	0.4	35	-0.5
26	4 (2)	Heating 3	285	0.3	140	0.5	95	0.5	70	0.5	50	-0.4
21	1	Heating 1 inj	270	3.9	138	2.8	90	2.5	70	1.8	68	0.32
22	1	Heating 2 inj	290	2.9	145	3.2	95	5.3	70	0.7	x	x
R407C-RL32S												
27	750 (4)	Cooling 1	1350	3.8-4.3	680	3.7-3.9	455	3.7-5.2	340	4.1-4.8	*	6.2-12.8
28	750 (4)	Cooling 2	1365	3.8-4.4	685	3.8-4.6	455	3.8-5.6	337	4.3-6.1	*	5.1-10.9
31	4 (3)	Cooling 1 inj	1400	4.2	705	4.3	465	4.1	340	4.0	330	6.4-8.6
32	4 (3)	Cooling 2 inj	1400	4.2	705	4.2	465	4.2	340	4.1	330	6.4-8.6
33	4 (3)	Cooling 3 inj	1365	4.2	690	4.1	455	4.1-5.0	330	4.7-7.0	320	7.6-11.9
29	3	Heating 1	*1	3.1	150	3.3	95	3.3	70	3.3	45	-2.0
30	3 (2)	Heating 2	*1	3.4	135	3.3	100	3.2	75	3.3	50	-1.7
34	3 (2)	Heating 3	285	3.6	145	3.6	97	3.6	72	3.6	45	-1.7

MAJOR RESULTS AND CONCLUSIONS

Results

A dynamic test facility was designed, constructed, and instrumentation provided to simulate oil management scenarios and operating conditions that would produce zero oil return (or worst case) situations in ducted-split, residential-size heat pumps. Unique instrumentation and visual observation features were incorporated in the dynamic test facility so that on-line, continuous measurement and monitoring of the oil return behavior was possible in a representative installation.

The dynamic facility was designed to evaluate the primary effect of vapor mass flux and thereby, flow velocity, over a range of conditions likely to be experienced in heating and cooling modes for a heat pump.

Six combinations of refrigerants and lubricant were tested, including a baseline combination of HCFC-22 and mineral oil, and R-407C with two immiscible mineral oils (1 GS and 3GS) and R-407C with a range of low to medium miscibility lubricants including Castrol SW32, SW68 and ICI Emkarate RL32S. These lubricants represent a range of viscosities to test their impact on oil return characteristics.

The flow velocity, under which worst case oil management was noted by visual and measurement techniques, corresponds to approximately 100 ft/min in heating (where low oil concentrations of 0.25 to 0.50% are normal). At higher oil circulation rates, the minimum flow velocities are as much as twice as high (i.e., 200 ft/min). Minimum flow velocities ranging from 350 to 375 ft/min are required in cooling. Based on the results of this study, these should be considered minimum velocities that should be maintained for good oil management. Approximate guidelines for good oil management are well above these levels in heating for low oil concentrations, although they roughly correspond to those indicated in selected ASHRAE literature. The guidelines (Ref. 3) for cooling mode operation were essentially confirmed. (See Table 3.5).

With the miscible lubricants, both HCFC-22 and R-407C have approximately the same minimum velocity limits. The use of immiscible lubricants with R-407C, such as 1 GS and 3GS, did not indicate any worst case oil return scenarios, based on the test conditions produced in the dynamic test facility.

The dynamic test facility included the capability to inject additional quantities of lubricant beyond that which would normally be produced by normal compressor discharge. The results of these oil injection tests provided an indication that excessive lubricant injection could result in localized oil pooling and trapping.

Conclusions

Experimental data, enhanced by visual observations, indicate that R-407C and POEs exhibit similar oil management behavior to the baseline HCFC-22 and mineral oils. Although the flow velocities at which the poor oil management (or zero oil return) is experienced are slightly different, the flow velocity results indicate consistent behavior.

A surprising and unexpected result of the extensive test program was the indication that R-407C and either 1GS or 3GS mineral oil exhibited good, if not better, oil return, i.e., these immiscible mixtures could operate at lower minimum velocities than for HCFC-22 and comparable mineral oils or R-407C and the tested POEs.

However, for all of the refrigerants and lubricants tested, extreme operating conditions (low saturation temperatures where lubricant viscosity is extremely high) may not be as thoroughly explored as necessary. For example, in tests with HCFC-22 and mineral oil, extreme low temperature operation (below 0 to 10°F) may have produced high viscosity conditions, produced an oil pooling condition or have resulted in the mixture entering the region of this refrigerant/lubricant pair where two liquid phases of high viscosity are produced.

Visual observations of poor oil return situations, such as that seen in vertical vapor lines and manifolds built in the dynamic test facility, probably provide a more consistent and reliable indicator of oil management than the on-line data produced by a UV oil concentration meter. The visual observations also included changes and rates of change of the compressor oil sump level during the various test sequences. Changes in these levels were rarely the reason for flagging poor oil management situations.

The UV oil concentration meter provided an on-line indication of circulating oil concentration for the R-407C/POE tests, where calibration data were available. The UV meter is not a reliable measure of oil concentration for immiscible mixtures. Where the UV absorbency of the lubricant is extremely low (as for the ICI RL32S lubricant), the meter readings should not be considered to be highly accurate, but only provided an approximate indication of oil flow in the system.

Although not the subject of this program, opportunities for oil trapping and pooling may exist in the indoor and outdoor sections of the heat pump. These are manufacturers' responsibilities and designs or configurations which permit oil pooling (i.e., bottom of heat exchangers, in accumulators or system sumps) should, of course, be avoided for overall good oil management practices. Furthermore, the compressor flow characteristics appear to have some impact on the oil return behavior in heat pump systems and must be considered in the design and test sequences.

The current industry guidelines developed twenty years ago for CFC and HCFC refrigerants and mineral oil and alkylbenzene lubricants appear to provide appropriate but conservative guidelines for the minimum mass flux (and hence flow velocity) required for good oil management practice. The guidelines, while suggesting an upper viscosity limit, do not adequately address all of the parameters that impact good oil management practices.

The ability to achieve adequate oil return up hot gas risers at lower than anticipated velocities may be due to the existence of very fine (smoke) oil suspension during these operating conditions. It appears that an oil transport model, based on anticipated oil particle size, may be appropriate for determining hot gas riser minimum velocity, particularly at the low oil circulation rate that characterizes modern scroll compressors.

RECOMMENDATIONS

Although the experimental program determined conditions under which the worst case oil return situations were determined for the six lubricant/refrigerant combinations that were studied, additional tests and analysis are recommended.

1. These tests should attempt to explore the use of additional HFCs, such as R-410A with POEs, as well as mineral oils to provide a broader set of guidelines for industry use with new HFCs. Both an upper and lower immiscibility dome have been observed with many R-410A/POE combinations. The impact on oil return should be studied over a range of temperatures where both regions of immiscibility are encountered in system operation.
2. A limited series of tests should be carried out to determine the location of the liquid refrigerant and lubricant during normal operation, shutdown, and startup mode of operation. This would require isolating one or more of the major components to determine the location of key oil traps and pooling locations.
3. Additional tests with another HFC blend such as R-404A, or one of those tested originally, should be undertaken at substantially lower evaporator temperature levels, (perhaps at 0 to 10°F or below) than the levels encountered in the current tests, in order to explore the effect of lubricant viscosity on the oil return correlations in refrigeration cycles and to establish reliable guidelines.

A more comprehensive analytical and experimental program, combined with available experimental data on oil circulation, should be undertaken to investigate the oil circulation patterns in representative HVAC systems. The development of a preliminary model that could provide guidelines for a wider variety of HVAC applications, such as package terminal air-conditioners (PTAC), duct-free splits, window room air-conditioners (WRAC), and ducted residential systems would be especially useful. Such a program would determine both similarities and differences in system operation compared with the split system heat pump studied in the present program.

REFERENCES

1. Ogata, S., Carrier Reliability and Technical Series; "Delchi Hi-wall Split System Heat Pump (Oil Return Testing - AC9000/Lunaria 68)," Oct. 19, 1994.
2. Refrigeration Systems and Applications, ASHRAE Handbook, 1994:2.11-2.16.
3. Jacobs, M. L., F. C. Scheideman, F. C. Kazen, and N. A. Macken, 1976. Oil Transport by Refrigerant Vapor. ASHRAE Transactions 81(2):318-29.
4. H. W. Sibley, Soft Optimization System Test of Refrigerant Blend D3 in a Split System Heat Pump (1994).
5. Private data communication to Carrier Corp., DuPont Co. (1994).
6. J. J. Grebner and R. R. Crawford, ASHRAE Trans. 99-1, 380 (1993).
7. S. Corr, F. T. Murphy, B. E. Gilbert and R. W. Yost, ASHRAE Trans. 99-2, 1123 (1993).
8. D. R. Henderson, Solubility, Viscosity and Density of Refrigerant/Lubricant Mixtures, Final Tech. Rept. DOE/CE/23810-34, Spauschus Assoc., April 1994.
9. R. C. Cavestri, Measurement of Viscosity, Density and Gas Solubility of Refrigerant Blends in Selected Synthetic Lubricants, Tech. Prog. Rep. DOE/CE/23810-20C, Imagination Resources, July 1993.
10. P-T-X Diagram for Castrol Icematic SW68/R134a, private data communication, Castrol, Inc. (1994).
11. W.L. Martz and A. M. Jacobi, Refrigerant-Oil Mixtures and Local Composition Modeling, Technical Report ACRC TR-68, U. of Illinois (1994).
12. P-T-X and Viscosity Data for Icematic SW Lubricants and Various Refrigerants, private data communication, Castrol, Inc. (1994).
13. P-T-X and Viscosity Data for Icematic SW Lubricants and Various Refrigerants, private data communication, Castrol, Inc. (1994).