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CALCUL DU NIVEAU DES COÛTS DIFFÉRENTIELS DE LA RECONVERSION DES CHAÎNES DE FABRICATION DES ÉCHANGEURS THERMIQUES DANS LES ENTREPRISES RECONVERTISSANT LEURS ACTIVITÉS À UNE TECHNOLOGIE À BASE DE HC-290 (DÉCISION 76/51)

Rappel des faits

1. Lors de la 75e réunion, le Comité exécutif a passé en revue la phase II du Plan de gestion de l'élimination du HCFC (PGEH) pour le Brésil[1]. Lorsqu'il a présenté la proposition de projet, le Secrétariat a expliqué que des travaux supplémentaires seraient nécessaires pour permettre d'évaluer le coût différentiel de la reconversion de deux chaînes d'échangeurs thermiques afin de remplacer le HCFC-22 par le frigorigène HC-290 dans les équipements de climatisation. Le Comité exécutif a approuvé la phase II du PGEH pour le Brésil et a demandé, notamment, au Secrétariat d'entreprendre les travaux supplémentaires sur le niveau des coûts différentiels de la reconversion des chaînes de fabrication des échangeurs thermiques dans les entreprises se reconvertissant à la technologie à base de HC-290, d'en faire rapport à la 76e réunion, et d'adapter le coût de la phase II du PGEH pour le Brésil, le cas échéant, dès réception de la présentation de la demande relative à la deuxième tranche (décision 75/43 f)).

2. En conséquence, le Secrétariat a recruté un expert technique indépendant, qu'il a chargé d'élaborer une étude visant à fournir des renseignements techniques factuels sur les modifications demandées des échangeurs thermiques et des chaînes de fabrication des échangeurs thermiques pour passer des climatiseurs à base de HCFC-22 aux frigorigènes HC-290, HFC-32 ou R-452B[2] , et une estimation du coût différentiel afférent à la reconversion[3]. Cette étude a été l'objet d'un examen[4] de pairs par deux experts, indépendants, en réfrigération, a ensuite été adaptée de façon à tenir compte des toutes les remarques et observations formulées par les deux experts, et présentée à la76e réunion.

[1] UNEP/OzL.Pro/ExCom/75/40 and Add.1.
[2] R-452B (un mélange de HFC-32 (67 pour cent), de HFC-125 (7 pour cent) et de HFO-1234yf (26 pour cent) a été ajouté dans cette étude, car il s'agit d'un mélange qui devrait être utilisé dans le secteur de la fabrication de climatiseurs individuels.
[3] Le cahier des charges de cette étude figure au paragraphe 3 du document UNEP/OzL.Pro/ExCom/76/59.
[4] L'examen par les pairs présentait une évaluation du rapport, et indiquait notamment si l'auteur souscrivait aux conclusions de l'expert. Il y était également analysé la mesure dans laquelle le rapport tenait compte du cahier des charges.

Délibérations de la 76 e réunion

3. À la76e réunion, le Comité exécutif a pris note du rapport et a invité le Secrétariat à soumettre à la 77e réunion un document révisé traitant des questions spécifiques ci-après (décision 76/51):

a) Examiner plus avant, en ce qui concerne la technologie à base de HC-290, les incidences techniques et relatives aux coûts, de la réduction du diamètre du tube de condenseur, passant ainsi de 7 mm à 5 mm, sans changement d'évaporateur; et

b) Fournir de plus amples renseignements sur le nombre estimatif d'unités requises pour chaque type d'outil/équipement dans le cadre de la réduction du diamètre du tube, selon le niveau type de production d'une entreprise d'un pays visé à l'article 5.

Ajouts à l'étude considérée

4. Compte tenu de la décision 76/51, l'expert du Secrétariat a réexaminé en conséquence l'étude dont il est question. La version révisée, jointe au présent document, comporte des précisions sur les points ci-après :

a) Le diamètre extérieur des tubes d'évaporateur peut être réduit, passant de 7mm à 5 mm (comme dans le cas du condenseur) sans augmentation de la chute de pression ou sans dégradation de l'efficacité du système. Il en découlera une réduction du coût des matières pour l'évaporateur de même que pour le condenseur (voir le paragraphe 4 c)), et une simplification du processus de production, avec la production dans l'installation de rouleaux dotés de tubes d'un seul diamètre (5 mm);

b) Pour une capacité de production de référence de 200 000 unités, les équipements et processus de fabrication utilisés pour la production de serpentins de tubes de 7 mm peuvent servir également pour les serpentins de tubes de 5 mm avec certains changements et ajouts ainsi qu'il est indiqué en détail dans l' étude. Les dépenses d'investissement concernant ces changements vont de 215 000 $US à 975 000 $US selon l'équipement de référence et le mode de reconversion ;

c) Changer les tubes de condenseur ou d'évaporateur de 7 mm à 5 mm entraîne une réduction du matériau de cuivre de 30 et 40 pour cent en fonction de l'épaisseur de la paroi des tubes initiaux (7 mm) et modifiés (5 mm) .Ceci équivaut à une réduction globale de 20 à 25 pour cent du coût des matériaux des serpentins. L'Annexe V de l'étude présente un calcul des économies en matériau pour une capacité de production de 200 000 unités;

d) L'expansion du tube est la seule opération de fabrication qui devrait se traduire par une complexité accrue ou entraîner des problèmes supplémentaires (c'est-à-dire davantage de déchets de serpentins) lors de la reconversion en tubes de 5 mm de diamètre extérieur. Pour traiter de ce problème potentiel, des renseignements sur un expandeur pneumatique de substitution de serpentins de tube d'un diamètre extérieur de petite taille ou 5mm ont été inclus. Cet équipement pourrait être nécessaire selon l'équipement de référence et les caractéristiques de la production ; et

e) Les calculs figurant dans cette étude sont une référence utile fondée sur une capacité de production de 200 000 unités; toutefois il y a lieu d'effectuer des évaluations au cas par cas compte tenu des précisions relatives aux serpentins et processus de fabrication de chaque entreprise.

Conclusions de l'étude

5. Les principales constatations et conclusions de l'étude, lesquelles sont semblables à celles qui ont été présentées à la 76e réunion et adaptées d'après les résultats du rapport révisé, sont résumées comme suit :

a) La principale propriété du frigorigène se répercutant sur la conception physique de l'échangeur thermique est la pression nominale maximale :

i) Le HFC-32 et le R-452B ont une pression nominale nettement supérieure au HCFC-22, mais se situant dans les limites de 10 pour cent de la pression nominale du R-410A. Des modifications minimales de conception et l'adjonction d'un essai du cycle de pression au processus de qualification relatif à la fabrication des serpentins permettra l'utilisation des conceptions actuelles de serpentins selon des pressions nominales supérieures pour le R-452B et le HFC-32;

ii) La pression nominale relative au HC-290 est inférieure à celle du HCFC-22; il n'est donc pas nécessaire de modifier la pression nominale ou de fabrication ;

b) D'autres propriétés physiques et thermodynamiques des frigorigènes déterminent les conditions de fonctionnement et le degré d'efficacité du système de climatisation dans lesquelles ils sont utilisés. Pour les trois frigorigènes considérés, ces paramètres entraînent des variations importantes entre les frigorigènes en ce qui concerne l'efficacité du système, la capacité du système, les températures de fonctionnement et les taux de débit du frigorigène. Ceux-ci sont importants lors de la sélection d'un frigorigène pour un type particulier de produit et pour l'optimisation de la conception des composants de l'échangeur thermique. Cependant, ils n'entraînent pas en soi les changements requis dans la conception physique de l'échangeur thermique ou dans les processus de fabrication qui nécessiteraient des dépenses d'investissement ;

c) Les trois frigorigènes ont des propriétés qui rendent possibles des réductions importantes de la charge et du volume du débit du frigorigène par rapport au HCFC-22. Ceci permet, mais ne requiert pas, une nouvelle conception des serpentins de façon à réduire les coûts par l'utilisation de tubes de diamètres plus petits, ce qui représente des économies en matériel mais des coûts d'investissement supplémentaires pour un nouvel outillage de fabrication ;

d) Les trois frigorigènes de substitution dont il est question dans l'étude sont considérés comme étant inflammables : le HC-290 est extrêmement inflammable (A3), tandis que le HFC-32 et le R-452B sont modérément inflammables (A2L). Les codes de sécurité régissant l'utilisation des frigorigènes inflammables dans les espaces occupés varient considérablement selon les lieux, et sont actuellement réexaminés et révisés de manière à prendre en compte les risques moindres d'inflammabilité liés à l'utilisation de frigorigènes de la classe 2L. L'utilisation des frigorigènes de la classe 2L (HFC-32 et R-452B) sera autorisée dans les petits condenseurs séparés (2 kW à 5 kW) placés dans un espace occupé sans système d'atténuation des risques, et dans un équipement unitaire plus grand (monobloc de toiture, 30 kW à 1 000 kW) avec un dispositif de détection des fuites et d'atténuation de la ventilation mécanique. Des limites de consommation maximale par circuit seront vraisemblablement prévues mais elles ne devraient pas constituer un problème pour l'utilisation de produits de cette catégorie. Une réduction de la consommation en deçà des niveaux actuels de consommation n'est donc pas requise pour l'équipement de climatisation utilisant du HFC-32 ou du R-452B;

e) Les restrictions actuelles concernant les codes qui ont été mises en place en Amérique du Nord et en Europe pour les frigorigènes de la classe 3 dans les espaces occupés, ne changeront probablement guère mais seront plus largement appliquées. Ces limites du volume des charges restreindront l'application du HC-290 aux systèmes d'une capacité inférieure à 2 kW sans atténuation des risques, et aux systèmes d'une capacité maximale d'environ 20 kW avec un dispositif actif d'atténuation des risques. Le coût estimatif des dispositifs automatiques d'atténuation des risques atteindrait 30 pour cent du coût d'un appareil de climatisation de petites dimensions. Ceci a pour effet de limiter la taille des équipements utilisant du HC-290 aux réfrigérateurs, refroidisseurs de petites taille et peut-être aux petits appareils de climatisation de type mini-système bi-bloc ;

f) La réduction de la charge de frigorigène et des coûts de matériel est obtenue quand le HCFC-22 est remplacé par le HC-290 dans un système de réfrigération et que le diamètre extérieur du serpentin de l'échangeur thermique est réduit. Avec des modifications mineures du circuit du frigorigène, un serpentin de 5 mm de diamètre extérieur peut être utilisé pour l'évaporateur ainsi que le condensateur sans être l'objet de sanction relative à l'efficacité;

g) Réduire le diamètre du tube à ailettes a des répercussions négatives sur presque tout le dispositif, mais ne devrait pas entraîner de grands changements en ce qui concerne les principaux biens d'équipement comme les presses à ailettes, les cintreuses, le matériel de manutention et l'aménagement de l'usine. Il est estimé dans l'étude le nombre de pièces d'équipement ou outils nécessaires pour produire 200 000 mini-systèmes bi-blocs d'une capacité de 18 000 BTU/h, et le coût afférent au changement du diamètre extérieur d'un serpentin, qui serait réduit à 5 mm. Le coût total de fabrication de l'équipement varie entre 215 000 $US et 475 000 $US à supposer que l'expandeur du tube actuel soit utilisé. Si un expandeur pneumatique est utilisé, le coût varierait entre 750 000 $US et 975 000 $US; et

h) Réduire le diamètre extérieur du tube, qui passerait de 7 à 5 mm dans le condenseur ainsi que l'évaporateur, réduit le volume du frigorigène, simplifie le processus de fabrication du serpentin et permet une importante réduction des coûts des matériels, à savoir le contenu de cuivre d'environ 30 pour cent, soit approximativement une diminution du coût 15 pour cent du serpentin achevé. Sur la base d'un volume de production annuelle de 200 000 unités d'une capacité de 18 000 BTU/h, les économies annuelles en termes de coût du matériel en cuivre se chiffrent 1 15 000 $US, soit 383 000 $US pour l'évaporateur et 767 000 $US pour le condenseur.

Retour d'information des agences d'exécution

6. L'étude révisée a été diffusée à l'intention des agences d'exécution en vue d'un retour d'information de leur part. Compte tenu de celui-ci, le rapport a été l'objet de précisions d'adaptations. Les principaux éléments reçus et les réponses fournies sont présentés ci-après :

a) L'étude est considérée comme étant très utile et traite des points soulevés par le Comité exécutif à la 76e réunion ;

b) Selon l'expérience acquise dans le cadre de la mise en œuvre des projets, la réduction de la charge de frigorigène découlant de la réduction du diamètre extérieur des tubes, passant de 7 mm à 5 mmn dans l'échangeur thermique, est de 15 à 30 pour cent, plutôt que de 50 pour cent selon les calculs figurant dans l'étude. Le pourcentage de 50 pour cent mentionné dans le rapport était fondé sur des calculs effectués à partir d'autres études ; une observation a donc été ajoutée pour indiquer l'expérience présentée par les agences d'après la mise en œuvre des projets ;

c) Il était considéré que les tubes de 5 mm peuvent être utilisés pour le condenseur ainsi que pour l'évaporateur, et que réviser la conception de l'évaporateur est nécessaire pour éviter une chute de pression. Dans le cas du fonctionnement de la pompe à chaleur, il y aurait également lieu de modifier la conception du condenseur. L'expert approuve l'observation formulée; des modifications du circuit seraient requises tant dans le condenseur que dans l'évaporateur des pompes à chaleur, afin qu'ils demeurent efficaces en cas de réduction du diamètre, bien que les pompes à chaleur n'aient pas été envisagées dans le cadre de ce rapport ;

d) L'expandeur pneumatique du tube est obligatoire pour permettre d'éviter une augmentation du taux de déchets. À ce sujet, l'expert a indiqué que dans le cas de l'utilisation d'une expansion de la tige "limitée – réduite", le taux de déchets est semblable à celui de l'expansion pneumatique. L'expansion de la tige "limitée – réduite", et l'expansion pneumatique sont donc des méthodes viables d'expansion du tube. Compte tenu de ceci, le Secrétariat conclut qu'il sera nécessaire de procéder à une évaluation au cas par cas, c'est à dire que la nécessité d'un épandeur pneumatique de tube dépendra de l'équipement de référence et du processus mis en place dans l'usine ;

e) Le coût des outils de l'expandeur de tube (variant, selon les estimations, entre 5 000 $US et 10 000 $US - un ensemble pour chaque taille de serpentin produit) en Chine s'est élevé à 130 000 $US. À cet égard, l'expert a indiqué que ce serait le cas si de multiples postes d'expansion sont utilisés ou bien si l'équipement d'expansion de la tige doit être reconverti de façon à ce qu'elle soit "limitée – réduite". Ceci est possible si les tubes sont mis sous tension en cas d'expansion de la tige ; et

f) Il est convenu qu'une réduction, à 5 mm, du diamètre extérieur permettra des économies relatives au cuivre, mais le niveau de ces économies est inférieur à celui qui est calculé dans l'étude parce que la zone du transfert de chaleur doit demeurer inchangée, et davantage de tubes de cuivre de 5 mm seront donc nécessaires. Sur ce point, l'expert a indiqué que la paroi du tube, la zone de face et le nombre de tubes ne nécessitent pas de modifications dans le cas d'une réduction de 7 mm à 5 mm du diamètre extérieur des tubes. La raison en est que la majeure partie de la résistance au transfert de chaleur concerne le côté de l'ailette du serpentin. Les économies de coût du matériel indiquées dans les calculs sont donc correctes.

Recommandation du Secrétariat

7. Le Comité exécutif souhaitera peut-être :

a) Prendre note du document UNEP/OzL.Pro/ExCom/77/69 , sur le calcul du niveau des courts différentiels de la reconversion des chaînes de fabrication des échangeurs thermiques dans les entreprises reconvertissant leurs activités à une technologie à base de HC-290 (décision 76/51);

b) Demander au Secrétariat d'ajuster le coût de la phase II du Plan d'élimination des HCFC pour le Brésil, le cas échéant, dès réception de la présentation de la demande pour la deuxième tranche, conformément à la décision 75/43 f), compte tenu des renseignements techniques figurant dans le document UNEP/OzL.Pro/ExCom/77/69; et

c) Demander au Secrétariat et aux agences d'exécution d'utiliser les renseignements techniques fournis dans le document UNEP/OzL.Pro/ExCom/77/69 comme référence lors de l'évaluation des coûts différentiels de la reconversion des chaînes de fabrication des échangeurs thermiques reconvertissant les climatiseurs à base de HCFC-22 au HC-290, HFC-32 et à des frigorigènes au R-452B.

A STUDY OF AIR TO REFRIGERANT CONDENSOR AND EVAPORATOR HEAT EXCHANGER MANUFACTURING CHANGES FOR CONVERSION FROM R-22 TO R-290, R-32, AND R-452B

William Plzak

August 2016

Table of contents

Summary

This study was performed at the request of the Multilateral Fund Secretariat (Secretariat) to serve as a basis to evaluate the incremental cost of the heat exchanger conversion proposed in the HCFC Phase-out Management Plan (HPMP) for Brazil as well as future heat exchanger conversions submitted for funding to the Executive Committee (ExCom) for consideration.

The study provides technical information on required modifications to heat exchangers and heat exchanger manufacturing lines when converting from R-22 based AC to: (1) R-290, (2) R-32, and (3) R-452B (DR-55) refrigerants, and provides an estimation of the incremental capital and operating costs associated with the conversion of heat exchanger manufacturing lines from R-22 to the three alternative refrigerants. Where appropriate, a similar study completed in October 2011 for conversion from R-22 to R-410A[1] was used as basis for assessment of costs for conversion from R-22 to these three alternative refrigerants. Two types of direct expansion AC systems were considered, 2 kw to 5 kw mini-spilt systems and 30 kw to 1,000 kw rooftop systems.

The study concludes that no major coil design changes will be required for conversion of this equipment from R-22 to R-290, R-32 or R-452B. However, the significantly higher design pressure for R-32 and R-452B will require a fatigue based pressure cycle and burst test in place of the current burst test for coil design qualification processes. Minimal new capital equipment will be required to make these changes to the coil manufacturing process. Automated pressure cycling equipment will be required to qualify the coil designs. R-290 has a design pressure that is 10% lower than R-22. Therefore use of R-22 coil design qualification processes will be acceptable for the coils when used with R-290.

All three of the alternative refrigerants in this study are considered flammable. At present the safety codes that govern use of flammable refrigerants in occupied spaces vary wildly by locale. The standard used in North America is currently being reviewed and revised to reflect the low flammability risks associated with use of the A2L refrigerants R-32 and R-452B. These revisions are expected to allow use of A2L refrigerants in occupied spaces with only minimal restriction and, in some cases, ignition mitigation systems. These restrictions are not expected to cause application issues with either mini-splits or rooftops systems.

R-290 is designated as class A3 refrigerant- highly flammable. Current code restrictions prohibit its use in North America to equipment rooms and restrict the charge to 3 kg. Codes in other areas are not as restrictive but still limit charge. The current charge limits in place for class A3 refrigerants have the effect of limiting the capacity of R-290 systems to about 1 kw with current coil designs. Somewhat larger charges will be allowed if additional active ignition mitigation is used. Thus practical application of direct expansion AC equipment using R-290 will be limited to refrigerators, small coolers and perhaps small mini-split type AC units unless design changes are made to reduce the refrigerant charge.

The refrigerant charge in R-290 systems can be significantly reduced by heat exchanger tube diameter reduction. This allows an increase to the maximum refrigeration capacity for systems using R-290 as a refrigerant. This study also discusses the performance and manufacturing process changes associated with reducing tube diameter in heat exchanger designs.

[1] UNEP/OzL.Pro/ExCom/66/51.

1. Introduction

The Secretariat assists the Executive Committee (ExCom) in managing the operation of the Multilateral Fund. At its 75th meeting in December 2015 the ExCom considered the HPMP for Brazil which included modification of heat exchanger manufacturing lines to produce R-290 based AC units. Due to the complexity of calculating the costs associated with this change, the Secretariat was directed to obtain additional information on the incremental cost for conversion of heat exchanger manufacturing lines to alternative refrigerant technologies and report back to the ExCom at the May 2016 meeting.

The objective of this study is to provide technical information on required modifications to heat exchangers and heat exchanger manufacturing lines for conversion of R-22 based AC to (i) HC-290, (ii) HFC-32, and (iii) DR-55 refrigerants, and to provide an estimation of the incremental capital and operating cost associated with conversion heat exchanger manufacturing lines from R-22 to the three alternative refrigerants. This study is focused on smaller air conditioning (AC) products, specifically 2 to 5 kW residential room AC units (mini-splits) and 30 to 1,000 kW unitary products.

A first version of this study was presented by the Secretariat at the 76th meeting in May 2016. Responses to specific questions from the ExCom on the study have been incorporated in this updated version and also listed separately in Annex I for reference.

R-22 (HCFC) had been the refrigerant of choice for use in residential and small unitary AC systems up until its phase out beginning in 2010. At that time R-410A (HFC) became the accepted replacement refrigerant for small residential split AC systems and medium size unitary (rooftop) products. However, with current focus of using lower GWP for refrigerants, a new set of alternative refrigerants are being considered. This study considered three potential replacements for R-22: R-290, R-32, and R-452B. These refrigerants all have zero ozone depleting potential, low (<750) GWP and physical/ thermodynamic properties suitable for use in small to medium size direct expansion AC systems.

The operating conditions of an AC system using the three alternative refrigerants were compared to the R-22 base. Changes to design pressure, compressor discharge temperature, and refrigerant flow volumes were used to determine if any changes were required to the heat exchanger designs to accommodate the new refrigerants. A “design standard” coil typical for the type and capacity of each product was used as the basis for the design change determination. Capital cost and production volume capabilities of manufacturing equipment modifications required to produce coils with these design changes was estimated based on information from US coil manufacturing equipment suppliers. The production capability of each type of manufacturing equipment and annual production volume requirements were used to determine if any duplicate equipment was required. Where appropriate the conclusions of a similar study completed in October 2011 for conversion from R-22 to R-410A were be used for assessment of costs for conversion from R-22 to the three alternative refrigerants.

2. Baseline coil information[2]

The typical baseline or “design standard” coils and the associated manufacturing processes assumed for this study are described in the following paragraphs. Coil designs in developing countries may or may

[2] The information in this section is based on information in the October 2011 study on heat exchanger conversion from R-22 to R-410A, updated to reflect current coil design standards.

not be similar those described. Older designs may very well be prevalent and upgrade to the design standard is not addressed by this study.

2.1a Residential (2-5 kW)

Equipment of this size and type generally has a single refrigeration circuit driven by one non-unloading rotary compressor. The evaporator coil is contained in cassette or cabinet located in the conditioned space and includes a fan. The condenser coil is located outdoors in an enclosure which also contains a fan and the compressor.

Both the evaporator and condenser use 7 mm outside diameter (OD) internally finned copper round tube coils with configured aluminum plate fins mechanically bonded to tubes. The tube wall thickness is 0.30 mm. Many manufacturers have switched to aluminum tube for the evaporator coil for cost reduction and resistance to formicary corrosion. Condenser and evaporator use one to three row coils with a 1 inch (25.4 mm) triangular tube pitch. Both evaporator and condenser coils use hairpin bends and brazed U bends. In the case of evaporators, a short orifice or capillary tube is used to feed the refrigerant circuits. Headers are made from small diameter copper or aluminum tube. All connections are brazed.

2.1b Unitary (30 – 1,000 kW)

The evaporator and condenser coils are contained within a complete packaged product, generally located on a roof. Equipment of this size and type generally has two or more separate refrigeration circuits each driven by one or more scroll compressors. The coils generally contain both refrigerant circuits in a single coil slab with the circuits intertwined to improve part-load performance. Multiple coil slabs are typically used for the higher capacity equipment.

Evaporator: 3/8 inch (9.5 mm) OD internally finned copper round tube coils with configured aluminum plate fins mechanically bonded to tubes. Some manufacturers have changed from copper to aluminum tubes for material cost reduction and elimination of formicary corrosion in indoor coils. The coils have 2 to 4 rows of tubes typically on a 1.2 inch (30.5 mm) triangular pitch. Coil height, length, rows, and number of refrigerant paths (coil tube circuits) varies by the refrigeration circuit capacity. A combination of hairpin bends and U-bends are used to connect tubes in each coil tube circuit. The U-bend to tube joints are flared and brazed. Tube wall thickness is typically 0.014 inch (0.35 mm). The first tube in each coil tube circuit is fed by a dedicated distributor tube connected to the coil tubes using either a crimped or flared brazed joint design. Each distributor tube is fed from a multiport distributor device. To maximize performance of the heat exchanger and minimize tube wall thickness required, the un-fined length of each tube is kept to a minimum, usually around 0.5 inch (13 mm). Overall refrigerant flow is controlled by a TXV. Outlet headers use pierced or pierced and flared braze joints with mitered or saddle type joints for the gas outlet line.

Condenser: 3/8 inch (9.5 mm) OD internally finned copper round tube coils with configured aluminum plate fins mechanically bonded to tubes. The coils have 1 to 4 rows of tubes on a 1.2 inch (30.5 mm) triangular pitch. Coil height, length, rows, and number of refrigerant paths (coil tube circuits) varies by the refrigeration circuit capacity. A combination of hairpin bends and U-bends were used to connect tubes in each coil tube circuit. The U-bend to tube joints are flared and brazed. Tube wall thickness is typically 0.014 inch (0.35 mm). The first tube in each coil tube circuit is fed from a cylindrical header, and the last tube in each coil circuit is connected to a cylindrical outlet header all made from copper. The diameter of these headers varies by overall refrigeration circuit capacity with the largest outside diameter about 1.625 inch (41mm). To maximize performance of the heat exchanger and minimize tube wall thickness required, the un-fined length of each tube is kept to a minimum, usually around 0.5 inch (13 mm). Both inlet and outlet headers use pierced or pierced and flared braze joints with mitered or saddle type joints for the gas inlet and liquid outlet lines. For non-heat pump applications use of aluminum micro-channel slabs is either an option or standard for many manufacturers. Use of this technology significantly reduces refrigerant charge and heat exchanger cost.

All Coils: Some manufacturers have taken advantage of the improvement opportunities afforded by conversion from R-22 to R-410A and made changes to the baseline coil designs. These changes improve AC system performance, lower refrigerant charge and lower coil material cost. They were made possible by the conversion to R-410A but were not required as part of the conversion process. Examples of these changes include tube diameter reduction, tube pitch changes to take advantage of smaller diameter tubes, and coil re-circuiting.

3. Coil HX design and manufacturing equipment modifications for refrigerant conversions

3.1 Standard manufacturing processes and required tooling/equipment

The standard manufacturing processes generally used for both condenser and evaporator coils in residential and unitary systems are shown in Table 1 below. The amount of automation and use of alternative processes is driven by volume/cost, manufacturing cycle time, manufacturing capacity investment considerations, and manufacturing quality control. Most tooling (fin dies, stackers, tube expanders, support plate stamping dies) used in these processes is specific to tube OD and inside diameter (ID), tube pitch, and fin heat transfer surface design. Therefore they would all need to be replaced if any of these design details are changed. However, the major capital equipment, such as fin presses and tubing benders would not need to change.

Table 1. Standard manufacturing processes used for condenser and evaporator coils in residential and unitary systems

Tooling/equipment	Description
Fins	Punched using a high speed progressive die fin press with automated feedstock and fin stacking. Fins are highly configured with features to improve air side heat transfer. A typical fin press uses a 24 row double progression die with a cycle rate of 300 strokes per minute. Press will be 75% utilized for 200,000 units annual production.
Tube cut off	Automated feed and cut machine, < 50% utilized for 200,000 units annual production
Hairpin bend	Hairpin bender with mandrels and automated feed, <50% utilized for 200,000 units annual production. Assumes 8 tubes per cycle, 6 cycles per minute.
Headers	Punched or drilled with automated or semi-automated machines. T-drill or similar may be used for more robust brazed joint, especially in larger diameter headers. Assuming 30 seconds per header annual production can be produced with a single station running 3 shifts, 5 days a week, (80% utilization.
U-bends	Purchased (brazing rings filler metal may be included)

Tooling/equipment	Description
Coil structural support	Sheet metal is punched using a progressive pierce and bulge dies and press. May be purchased or produced in a single station.
Coil assembly (lacing of tubes)	Manual or semi-automatic. Will probably require more than one station but fixtures are minimal.
Tube expansion	Ball end multi-rod expanders or ball expanders. Requires expander press with tooling specific to the tube ID. The equipment may have tube tensioning features or similar means to prevent buckling and control tube shrinkage during expansion process. Cycle time 30 seconds.
Brazing headers, u-bend, and distributor tubes	Brazed with dry nitrogen purge. U-bends are auto brazed in continuous process. Headers and distributors are hand brazed. One station required for annual volume of 200,000. Tooling/fixtures are minimal. Automatic multi-joint brazing is required for aluminum tube coils.
Pressure and leak test	Air under water immersion in water tank with safety cover. One per coil line.
Final product pressure and leak test	Dry air plus halogen leak detector. One per coil line.
Process fluids	All process fluids used during manufacture and testing of the heat exchangers are selected to be compatible with R-22 (or refrigerant used in the refrigeration system) and the compressor lubricant (currently polyolester (POE) oil for R-410A)

Many low volume equipment manufacturers choose to purchase heat exchanger components rather invest in the manufacturing facility and equipment to build them. This is especially true for aluminum micro-channel air to refrigerant slabs that are often used in cooling only condensers for unitary AC products.

3.2 System design considerations for safety regulation imposed refrigerant volume limits for flammable refrigerants

All three of the alternative refrigerants in this study are considered flammable. Safety regulations that govern use of flammable (A2, A2L and A3) refrigerants in occupied spaces limit the maximum volume of the refrigerant that can be used. For products within the scope of this study using A2 and A2L refrigerants the regulations will have minimal impact to heat exchanger design or product capacity range. For products using A3 refrigerants such as R-290 the maximum refrigerant volume limit will limit product capacity using current heat exchanger coil designs. Details of the regulations and their impact on direct expansion AC systems using the alternative refrigerants are presented in Annex II.

Design changes to the AC equipment such as use of micro-channel heat exchangers, component size reduction, and component elimination can be considered to reduce charge volume in an exercise to determine the maximum capacity of direct expansion systems that could use R-290 without mandatory mitigation systems. Based on compilation of results found in studies presented at recent engineering conferences as much as an additional 50% charge reduction is possible by reducing the coil tube OD from 7mm to 5mm[3]. Therefore the largest R-290 single circuit split system that would comply with

[3] According to UNIDO's experience in developing countries the refrigerant charge reduction resulting from the reduction of the OD of the tubes from 7mm to 5mm in the heat exchanger can be between 15 to 30 per cent.

current safety standards and not require active ignition mitigation systems would be about 6 kW. Larger split system AC equipment would need to have multiple refrigeration compressors and circuits.

Based on this assessment for equipment designed using current heat exchanger coil designs the practical application of R-290 in North America will be limited to refrigerators, small bottle coolers, and mini-split AC products. Countries in South America, Africa and the East Asia will likely have somewhat higher charge limits for R-290 refrigerant in direct expansion AC systems. Additional detail is provided in Annex III.

3.3 Heat Exchanger Coil Design Modifications Related to Design Pressure and Refrigerant

The design pressure for R-290 is about 10% less than the design pressure for R-22 therefore pressure related heat exchanger design/or manufacturing changes will not be necessary. Current coil designs and coil qualification tests that are used for R-22 designs will be acceptable for R-290. Typically this consists of a burst test on several representative samples for each coil design. The minimum burst test pressure required is 5 times the design working pressure. For each production coil a pressure proof test is performed at 1.5 times design pressure.[4] The lower design pressure will allow a reduction to these test pressures. This will have no impact on product or manufacturing cost. The lower design pressure may also allow a small reduction to tube wall thickness, on the order of 0.001”. This change could result in a tube cost reduction of about 5%. Heat transfer improvement from a wall thickness reduction would be small and the impact on overall system performance would be negligible.

The only coil design changes that result in manufacturing tooling cost are design changes to minimize the frequency and severity of leaks. Joint count reduction (if possible) and improvement in header to coil tube, and distributor tube to coil tube joint designs will likely be made to mitigate risks due to the high flammability of the refrigerant. These design changes will carry a minimal capital cost for new header T-drill (US $6,000 each) and distributor tube crimp tooling (US $12,000 each). Production cost to add these processes to the build have not been determined but are expected to be small.

In addition, R-290 with its class A3 rating may in many cases require some system level safety related features. These will be part of the refrigerant system and will not impact the design or manufacture of the heat exchanger coils.

The design pressure for R-32, R-452B and R-410A are 50% to 70% higher compared to R-22. The coil design changes that are necessary to withstand these pressures are driven by governing codes and standards. Applying these codes and standards with the current coil design and manufacturing methods results in an increase to the tube wall thickness and tube material cost. An alternative method for pressure design qualification is available in the standards. This method allows use of current design tube wall thickness for the increased design pressures. Details of this coil design qualification method and impact on the coil manufacturing equipment are included in Annex IV.

In addition, R-32 and R-452B have an A2L flammability rating and may require some system level safety related features. These will be part of the refrigerant system and will not impact the design or manufacture of the heat exchanger coils.

[4] The coil design qualification process noted here is typical for products designed and manufactured in compliance with UL 1995.

Process fluids and mineral oils are carefully chosen and qualified to insure compatibility with R-22. Systems using R-452B and R-32 both use synthetic POE oils for compressor lubrication. Systems using R-290 will use mineral oil lubricant similar to the oil used in R-22 systems. Process fluids used in the manufacture of the heat exchanger coils must be chosen carefully to confirm that they are compatible with the compressor lubricant used in the system and do not cause lubricant breakdown. Lubricant breakdown will ultimately lead to premature compressor failures.

4. Refrigerant Volume Reduction and Material Cost Savings Potential for Coil Tube OD Reduction

4.1 Refrigerant Volume Reduction

A typical 2 to 5 kw residential mini-split AC system with R-22 refrigerant uses 7 mm OD tubes for both the evaporator and condenser heat exchanger coils. Changing to R-290 without any design changes to the heat exchangers provides approximately 50% reduction in refrigerant charge. The lower mass flow rate for R-290 provides an opportunity for further refrigerant charge reduction by reducing the tube diameter in the condenser. Condenser tube OD of 5 mm can be used with no performance degradation due to increased frictional pressure drop and provides as much as an additional 50% charge reduction. A similar reduction in evaporator tube OD has no appreciable effect on refrigerant volume and can cause system performance degradation unless changes to the evaporator refrigerant circuiting are made. It would however be beneficial to the overall coil manufacturing process and material cost to also convert the evaporator to 5 mm OD tube. The typical evaporator coil design for R-22 uses partial row circuiting. Calculations using the assumed base-line evaporator coil show that full row circuiting allows 5 mm OD tubes to be used without pressure drop increase.[5] The refrigerant volume reduction estimates for R-290 and smaller diameter tubes are based on compilation of results found in studies presented at various Engineering conferences in recent years. The NIST coil designer software was used to develop pressure drop and performance predictions for the evaporator. Refer to Annex V for details of this analysis.

The refrigerant charge reduction from smaller OD coil tubes allows larger capacity R-290 AC systems to operate within safety code mandated refrigerant charge limits. For North American applications this increase is not likely to expand application of R-290 significantly but in other regions where volume limits for R-290 in occupied spaces are less restrictive, the charge reduction made possible by coil tube diameter will enable application of R-290 to small residential split systems as well as some small unitary AC systems.

4.2 Coil Material Cost Savings

In addition to the refrigerant volume reduction and coil manufacturing process simplification, smaller diameter tubes provide a significant material cost reduction. Calculations show that reducing the tube OD from 7mm to 5 mm in both the condenser and evaporator reduces copper material content by about 30%. For the condenser and evaporator this change can be made without changing the coil tube pitch or fin surface design[6]. As noted in section 4.1, circuiting changes are required in the evaporator to maintain system performance.

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[5] Partial row circuiting means that the number of parallel refrigerant circuits in the evaporator coil is less than the number of tubes in each row of tubes. Full row circuiting is when every tube in the first row of a coil is fed by refrigerant and the number of parallel refrigerant circuits equals the number of tubes in the first row. Changing from part row to full row circuiting shortens the circuit length and reduces the refrigerant flow rate in each circuit and therefore lowers the total coil refrigerant pressure drop for a given refrigeration capacity.

[6] The tube pitch, face area and number of tubes do not require changes when going from 7mm to 5mm OD tubes. This is because the majority of the heat transfer resistance is on the fin side of the coil.

A 30% reduction in copper material results in approximately a 15% cost reduction for the completed coil. Based on an annual production volume of 200,000 units with a capacity of 18,000 Btu/hr annual copper material cost savings is for the evaporator is $383,000. And the savings for the condenser is $767,000. Total annual copper material cost savings is $1,150,000. Refer to Annex V for details of this analysis.

5. Manufacturing Facility Conversion Cost for Alternative Refrigerants

5.1 Incremental capital cost for coil conversions to R-290, R-32 and R-452b (without coil tube diameter change)

Table 2 below is a summary of the manufacturing tooling changes expected for conversion from R-22 to one of the alternative refrigerants covered in this study. These items are applicable to all three refrigerants unless an exception is noted.

Table 2. Manufacturing tooling changes that are expected for conversion to an alternative refrigerant (w/o tube diameter change)

Tooling/equipment	Description
Fins	No required changes – No tooling cost
Tube cut off	No required changes – No tooling cost
Hairpin bender	No required changes – No tooling cost
Coil headers	T-drill or similar must be used. T-drill heads cost approximately US $6,000 per drill head and can be used with the either manual or automated drilling equipment. The basic drilling equipment itself does not change. For an annual production volume of 200,000 units a single drilling station will be sufficient
U-bends	Purchased item – No tooling cost
Coil structural end plates and supports	No required changes – No tooling cost
Coil assembly	No required changes – No tooling cost
Tube expansion	No required changes – No tooling cost
Headers, U-bend or distributor tubes	Crimping of distributor tube to coil tube estimated to cost about US $12,000 per station. For an annual production volume of 200,000 units a single station will be sufficient
Pressure and leak test	No required changes – No tooling cost
Final product pressure and leak test	No required changes – No tooling cost

Heat exchanger coils designed for use with the higher pressure refrigerants (R-32 and R-452) will require a fatigue test to qualify the design and manufacturing processes. The cost for a test facility to perform this test on multiple coils is around US $120,000. This facility of this size is capable of testing 3 coils in parallel with test duration for 250,000 cycles of about 1 week (150 hours). Initial qualification requires 3 coils to be tested as above. Maintaining design/manufacturing process qualification requires test of 1 sample of each design performed at least annually depending on production level. Coils in continuous production require retest every 3 months. In addition there is a cost to certify, operate and maintain the equipment. For these reasons some smaller manufacturers decide to outsource the testing to a lab that provides this type of service. The cost per test at such a facility is about US $5,000. Assuming that some retesting is required an initial qualification cost per coil design of about US $30,000. Annual cost to maintain qualification for a single design is between US $5,000 and US $20,000 depending on production level.

The total manufacturing facility change over cost per coil line for either R-32 or R-452B is approximately US $18,000 for production tooling and US $120,000 for a fatigue test facility. Alternatively, if the fatigue testing is outsourced there would be an additional changeover cost of US $30,000, and an annual coil qualification cost of US $5,000 to US $20,000. These costs are for each discrete coil design produced.

Capital costs required for the manufacturing process changes (primarily the design qualification tests) for the higher pressure refrigerants (R-32 and R452B) are driven by the pressure fatigue tester. It is expected that only one of these units will be required for each coil production facility. Therefore the per unit cost will be vary inversely with product volume. The tooling required for the other manufacturing process changes (T-drill and tube crimper) that are required for all three refrigerants are low cost. The number of these systems required in a facility will be proportional to product volume. Therefore the capital cost per unit will not change significantly with volume. A summary of the incremental capital cost for coil conversions is presented in Table 3. Estimate of the number of stations or tools required is based on an annual production volume of 200,000 units.

Table 3. Incremental capital cost summary for coil conversions (without coil tube diameter change)

Tool/Equipment	**Incremental cost (US$)**		**Comments**
	R-32, R-452B	**R-290**	
Fin press stackers, fixtures & parts	0	0	No tube or fin changes required
Fin dies	0	0	“
Hair pin tube bending tools	0	0	“
Tube end processing tools	0	0	“
Expander dies	0	0	“
Braze ring insertion tooling	0	0	“
T-drill for headers	6,000	6,000	One required
Crimp tool for distributor tubes	12,000	12,000	One required
Automatic pressure cycle test machine	120,000	0	One required. Used only for design qualification and periodic checking. Not production volume dependent
Total	$138,000	$18,000	

5.2 Capital cost summary for coil tube OD change from 7 mm to 5 mm

Tube diameter changes in fin-coil heat exchangers affects nearly every part of a coil and thus the tooling required to manufacture those parts. However, major capital equipment such as fin presses, tubing benders, tube expanders, material handling equipment and plant arrangement should be usable with the smaller OD tubes. Table 4 below lists the basic tube fin coil manufacturing equipment, changes required for 5 mm OD tubes, the estimated cost to make the changes and an estimate of the number of pieces of equipment or tool required to produce 200,000 units annually. These estimates are for production of mini-split units with a capacity of 18,000 Btu/hr and an annual production volume of 200,000 units. The estimates are based on discussions with equipment suppliers and past experience in the field.

Table 4. Basic tube fin coil manufacturing equipment changes required for 5 mm OD tubes

Tooling/equipment	Description
Fin press	The existing fin press used for the production of Aluminum fins for 7 mm tubes can be used for 5 mm tubes. The annual production capacity of this equipment for 5 mm tubes will be the same as for 7 mm tubes
Fin dies or molds	The fin dies will need to be either modified or replaced for 5 mm tubes. Cost of a modified fin die is US $100,000 to US $150,000. Cost of a new die is US $200,000 to US $300,000. The range of cost is due to variation in die complexity, size, and capacity. The annual production capacity of the fin press and die for 5 mm tubes will be the same as for 7 mm tubes. A single fin die will be sufficient for the annual production
Fin press stackers, fixtures and parts	This equipment will need to be modified for 5 mm tubes. Cost estimated at US $50,000 to US $75,000 for each fin press. The annual production capacity of this equipment for 5 mm tubes will be the same as for 7 mm tubes
Hair pin tube bender	No change to bender, use same as for 7 mm tubes. Programming and tube cutoff may need to be adjusted. The annual production capacity of this equipment for 5 mm tubes will be the same as for 7 mm tubes
Hair pin bender tools	New tube mandrels will be required for 5 mm tubes at a cost estimate of US $10,000 for each tube bender. Annual capacity will be unchanged from 7 mm tubes
Tube expander	Rod expander used for 7 mm tubes will work for 5 mm tubes however scrap rate may increase due to reduced strength of smaller diameter tubes. Annual production volume will remain unchanged from 7 mm tubes. One tube expander will be able to handle the annual volume
Tube expander tools	New tools for the expander will be required at a cost of approximately US $5,000 to US $10,000 per set. One set is required for each coil size in production
Optional pneumatic tube expander	An option would be to use a pressure based tube expander and tube end processor that has been developed for smaller OD tubes. Cost is approximately US $500,000 including one set of coil fixtures. At least one additional fixture set would be required. Cost of additional fixture sets is $50,000. This equipment includes tube end processing. The main benefit of this equipment is reduction of scrap rate due to buckled tubes and inconsistent tube shrinkage during expansion. Purchase of this equipment may be justified on this basis. One expander will be able to produce the annual production volume. If “limited – shrink” rod expansion is used the scrap rate will be similar to pneumatic expansion. Therefore both “limited – shrink” rod expansion and pneumatic expansion are viable tube expansion methods.
Tube end processing tools	New tools are required for 5 mm tubes. Cost estimate is US $10,000 per tube expander. The tools and the operation may be part of the tube expander. The annual production volume will be the same as for the 7 mm tube coils. One set of tools will be able to handle the annual production volume
T-drill for headers	New process required to improve the tube to header joint for class 3 flammable refrigerants. Cost estimate is US $6,000 per station, one station required for each header drill station. Annual production volume is the same as for current header drill process. One drill station should handle the annual production volume

Tooling/equipment	Description
Crimp tool for distributor joints	New process required to improve the tube to distributor joint for class 3 flammable refrigerants. Cost estimate is US $12,000 per station, one station required for each distributor braze station. Annual evaporator coil production volume will be 50% of the production rate for 7 mm tubes because the number of circuits is doubled for 5 mm tubes. This increase should still be able to be handled by one crimping station
Tube insertion stations	This is assumed to be a manual process therefore no tooling changes are expected. Production volume is the same as for 7 mm tubes
Braze ring insertion tools	New tools are required for 5 mm tubes. Cost estimate is US $10,000 per coil braze line. The annual production volume will be the same as for the 7 mm tube coils. One station should handle the annual production volume
Automatic braze station	Minimal or no tooling changes. Cost estimate US $5,000. Minor adjustments may be required. Production volume is the same as for 7 mm tubes. One station should handle the annual production volume
Manual braze station for headers and distributors	No tooling changes expected, minimal if any fixture changes. Additional circuits in evaporator will reduce output by about 50%. Therefore will require 50% more manual braze stations to produce the annual volume

Total cost for manufacturing equipment is US $215,000 to US $475,000 assuming the current tube expander is used. If the pneumatic expander is used the cost is US $750,000 to US $975,000. This does not address the development cost associated with these changes. Costs of this nature are proprietary to manufacturers and will vary greatly depending upon how the coil designs are developed or obtained. They are not considered to be within the scope of this study.

6. Heat exchanger efficiency improvement options

The condenser and evaporator heat exchangers that are part of all AC systems are designed using a complex process of system optimization with inputs such as refrigerant thermal and physical properties, heat transfer surface performance, material and component costs, compressor performance, and system operating conditions. The system design optimization process is performed and the output is an AC system with desired power consumption and capacity. Done correctly this process results in the lowest total cost system to achieve a desired set of performance goals.

When a new refrigerant with properties different from the original refrigerant is “dropped” into the system many of the key input parameters for the system design will be changed. The degree to which this effects the performance of the heat exchangers and how close they are to being optimized for the new refrigerant depends on how close the new properties are to the original refrigerant properties.

In general, use of a replacement refrigerant as a “drop-in” will not result in an optimized AC system and there will be opportunity for improving system performance by optimizing the design of all components including the heat exchangers.

There are several heat exchanger coil design parameters and operating conditions that can be changed to improve the performance of air to refrigerant heat exchangers for a given refrigerant. These include:

- Improving internal and/or external surface performance:
    - Improving the outside surface (fin) performance by changing its geometry. (large tooling cost for new fin dies)
    - Improving the inside surface (tube) performance by enhancing the inside surface of the tube. (usually a purchased component)
    - For evaporator coils, improve the refrigerant metering and distribution system to achieve more uniform refrigerant distribution
    - Increased air flow across the coil (higher fan power required)
    - More uniform air flow across the coil (geometry of cabinet or component placement)
- Increasing the internal and/or external surface area:
    - Increasing the outside surface area by increasing the fin count. (minimal tooling change but adds to fan power)
    - Increasing the inside surface area by decreasing the tube pitch and thus increasing the number of tubes. (large tooling cost for new fin dies)
    - Coil surface increase through additional number of rows or total surface area. (small tooling cost but new design required and higher fan power)
- Improved fluid flow within each coil refrigerant circuit by “interweaving” tubes within the circuit to achieve more ideal temperature profiles on the air and refrigerant sides
- Use of micro-channel tube/fin surfaces to greatly increase surface area to volume ratio and reduce refrigerant charge (system redesign, microchannel coil slabs usually purchased)

The heat exchanger coils operate as part of the refrigeration system. Therefor heat exchanger coil design changes should be done as part of the system re-optimization for the new refrigerant. Merely adding surface to a coil is costly and, as approach temperatures become smaller, decreasingly effective.

7. High ambient temperature considerations

For the purposes of heat exchanger coil physical design high ambient temperature operating conditions do not come into play. This is because the heat exchanger coils are designed to a pressure corresponding to the highest temperature that could be encountered during shipment or while sitting idle in a hot equipment room. The saturation pressure corresponding to 160 F (71 C) was used for this study. This is well above any ambient operating condition likely to be encountered.

8. Conclusion

The primary input parameter for structural design of a coil is maximum design pressure. Two of the alternative refrigerants under consideration, namely R-452B, and R-32, have substantially higher design pressures than R-22. The results of this study show that minimal design changes and addition of a pressure cycle test to the coil manufacturing qualification process will allow existing coil designs to be used at the higher design pressures for R-452B and R-32. The third alternative refrigerant being evaluated, (R-290), has a design pressure lower than R-22 and will therefore not require any pressure related design or manufacturing process changes.

Physical and thermodynamic properties of refrigerants determine the operating conditions and performance of the AC system in which they are used. For the three refrigerants considered in this study these parameters result in significant variation between the refrigerants in system efficiency, system capacity, operating temperatures and refrigerant flow rates. These are of great importance in making a refrigerant selection for a particular type of product, and for design optimization of the heat exchanger components. However, they do not by themselves result in any required changes to the heat exchanger physical design or manufacturing processes that would require capital expenditures.

All three refrigerants have properties that allow significant charge and refrigerant flow volume reductions compared to R-22. This allows, but does not require, the opportunity to cost reduce the coil designs by use of smaller diameter tubes. The cost savings available are substantial but come with a significant capital cost for new manufacturing tooling.

All three of the alternative refrigerants in this study are considered flammable. One, R-290, is highly flammable A3 classification. The others, R-32 and R-452B, are classified as A2L, which is defined as having a flame speed less than 10 cm/sec. (This means that these refrigerants may be difficult to ignite and sustain a flame.) At present the safety codes that govern use of flammable refrigerants in occupied spaces vary wildly by locale. These standards are currently being reviewed and revised to reflect the low flammability risks associated with use of the class 2L refrigerants. For class 2L refrigerants the revisions are expected to allow use within concentration limits established for each refrigerant and risk mitigation systems required on equipment. These restrictions are not expected to cause application issues with either mini-splits or rooftop, but could increase the product cost relative to R-410A. Similar changes to ISO and IEC standards are expected to follow. The timing of these changes to the standards will align with the expected market implementation date for AC equipment using the new refrigerants.

Bottom line for R-32 and R-452B refrigerant is that they will be allowed for use in small (2 kW to 5 kW) split systems located in an occupied space with no mitigation in place. Larger unitary equipment (30 kW to 1000 kW rooftop) will be able to use the refrigerants with leak detection and mechanical ventilation mitigation required on most AC equipment in this range. There will likely be maximum charge limits per circuit but they are not expected to be an issue for application of products in this size range. Therefore charge reduction below the current charge levels is not required for AC equipment using R-32 or R-452B. Current code restrictions in North America and Europe for use of class 3 refrigerants in occupied spaces are unlikely to change significantly but rather will become more widely applied. These charge volume limits will restrict the application of R-290 to systems <2 kW capacity without risk mitigation, up to a maximum of about 20 kW with an active risk mitigation system. The cost of the automatic risk mitigation systems have been estimated at up to 30% of the small AC unit cost. This has the effect of limiting the size of equipment using R-290 to refrigerators, small coolers and perhaps small mini-split type AC units.

Countries in regions outside of North America and Europe are in the process of introducing R-290 refrigerant in AC equipment. These countries may allow somewhat larger refrigerant charge limits than the ones noted above.

Refrigerant charge and material cost reduction is obtained when R-22 is substituted for R-290 in a refrigeration system and the heat exchanger coil tube OD is reduced. With minor changes to refrigerant circuiting a coil tube OD of 5 mm can be used for both the evaporator and condenser without performance penalty.

Annex I

Response to the questions posed by the ExCom at its 76th meeting

While the responses to the ExCom questions have been integrated within this revised study and its conclusions, specific answers are listed below for easy reference.

1. When converting heat exchangers for residential air-conditioners (9,000 to 18,000 Btu/h) from HCFC-22 to HC-290; reducing the tube diameter of the condenser from 7 mm to 5 mm has significant impact on the overall HC-290 charge as evidenced in studies carried out in China. However, the reduction has only minor – if any – impact on the efficiency of the condenser. In case of the evaporator; and in particular since capillary tubes are used for expansion; a reduction in tube diameter will result in increased pressure drop over the evaporator; and hence, have negative impact on the energy efficiency of the air conditioner. This with the assumption that residential air conditioners have size limitations; and that increasing the area/size of the heat exchangers is not an option. Kindly confirm this statement;

<u>RESPONSE</u>: My calculations show that the OD of evaporator tubes can also be decreased to 5 mm when converting to R-290 from R-22 without increased pressure drop or system performance degradation. This is assuming that the number of circuits can be roughly doubled from the number used for R-22. This assumes that the original R-22 coil did not use full row circuiting. The capillary tubes or orifices for each circuit will also need to be changed. As noted above this will not reduce refrigerant charge but will provide a material cost reduction for the evaporator similar to that of the condenser. It will also simplify the production process by requiring that coils with only one tube diameter will need to be produced in the facility

2. In case of confirmation of above, kindly clarify whether above reduction of the condenser tubes will result in an additional operation at manufacturer or whether this can be accommodated within existing manufacturing machinery. The associated costs should also be specified for both scenarios. The reference manufacturer should have an annual production capacity of approximately 200,000 units; and should be located in a developing country;

<u>RESPONSE</u>: The manufacturing equipment and processes used for production of 7 mm tube coils can also be used for 5 mm tube coils with some changes and additions as described in detail in the addendum draft report. Capital costs for these changes range from US $215K to US $975K depending on path taken for conversion.

3. What would be the potential reduction of raw material costs when changing condenser tubes from 7 mm to 5 mm? (Current study makes reference from 9.5 mm to 5 mm)?

<u>RESPONSE</u>: Changing either the condenser or evaporator tube diameter from 7 mm to 5 mm results in a copper material reduction of between 30% and 40% depending on what wall thickness is currently used for the 7 mm tubes and what wall thickness is used for the 5 mm tubes. This equates to an overall coil material cost reduction of 20% to 25% for 7 mm to 5 mm coil tube OD.

4. What is the impact on heat exchanger production complexity when reducing tube diameter from 7 mm to 5 mm; e.g. hairpin bending, expansion, welding/brazing?

RESPONSE: Tube expansion is the only manufacturing operation that is expected to result in added complexity or additional problems when converting to 5 mm OD tubes. It is likely to manifest itself in increased coil slab scrap rate. For this reason I have included some information in the draft report on an alternative pneumatic expander for small OD tube coils.

5. A unit cost breakdown that is presented in the paper should be further elaborated to clarify the estimated number of units required from each tool/equipment in order to produce the required capacity (200,000 units). Explanation for the basis of such estimates would also be appreciated.

RESPONSE: Calculations were made based on assumptions for the capabilities of the current production equipment used at the facilities being considered. A more accurate assessment could be made if details of the current coil designs and manufacturing processes are known.

Annex II

Regulatory background and direction for flammable (A2L and A3) refrigerants

All three of the alternative refrigerants in this study are considered flammable. One, R-290, has a (highly flammable) A3 classification. The others, R-32 and R-452B, are classified as A2L, which is defined as having a flame speed less than 10 cm/sec. This means that these refrigerants may be difficult to ignite and sustain a flame. At present the safety codes that govern use of flammable refrigerants in occupied spaces vary wildly by locale.

The two most widely recognized standards that govern the safe application or refrigeration systems are ASHRAE 15, **Safety Standard for Refrigeration Systems**, and ISO 5149, **Refrigerating Systems and Heat Pumps – Safety and Environmental Requirements.** Limiting this discussion to the occupied spaces that are served by residential split systems and rooftops, both ASHRAE 15 and ISO 5149 take an engineering approach to limit the charge and employ ignition mitigation.

At this writing, ASHRAE 15 has not yet published application rules for the use of A2L refrigerants. Additionally, product standards that are published by Underwriters Laboratory (UL) have not been published. Once ASHRAE 15 is changed and UL product standards become available, it takes 1-3 years to be adopted by the model codes that legally govern the installation of refrigeration systems in North America. Full adoption of standards that allow use of A2L is several years away. However, the process to change these standards to allow the use of A2L refrigerants is well along. The current view is that ASHRAE 15 will allow A2L refrigerants to be broadly used in all sizes of direct expansion AC systems with minimal restrictions or cost impact. Charge limits will be based on an engineering calculation that will prevent leaked refrigerant from reaching the lower flammability limit (LFL). Ignition mitigation may also be used under some circumstances. Presently ASHRAE 15 restricts the use of A3 refrigerants to equipment rooms only and with a maximum charge of 3 kg. There are no proposals to change this.

ISO 5149 was published in 2014 and offers a three level LFL based approach for use with class A2L, A2, and A3 refrigerants. In this standard there are three charge levels calculated using LFL for the refrigerant. These charge levels are coupled with the type of ignition mitigation required. The first charge level (m_1) can be used without ignition mitigation in any size room provided the refrigeration system is sealed with no field service ports provided. The second charge level (m_2) is the maximum refrigerant charge allowed for a specified minimum room area without refrigerant detection and mechanical ventilation. If the room area requirement is met, no other ignition mitigation is required; otherwise mechanical ventilation is required in this m_2 range. This range also requires a sealed system without service ports. Above m_2 mandatory detection and mechanical ventilation is required and a maximum charge limit is set at m_3. ISO 5149 treats A2L, A2, and A3 refrigerants the same except that an additional multiplier on LFL is used when calculating the maximum charge levels for A2L refrigerants.

Applying ISO 5149 to R-32 and R-452B gives refrigerant charge limits of 2 kg, 12 kg, and 59 kg for m_1, m_2, and m_3 levels respectively. Using these charges R-32 and R-452B could be applied w/o mitigation to split systems up to about 8 kw, with minimum room size limit up to about 43 kw, and with ignition mitigation up to about 215 kw per refrigeration circuit. These limits will have minimal impact on application or cost of direct expansion AC products within the scope of this study.

Appling the formula in ISO 5149 to R-290 provides the following refrigeration charge limits for each level:

- 150 gm maximum for no ignition mitigation but with a sealed, non-serviceable refrigeration system

- 1 kg maximum for no ignition mitigation, with minimum room size and sealed, non-serviceable refrigeration system

- 1 kg to 5 kg maximum will require leak detection, mechanical ventilation, minimum room size, and a sealed non-serviceable refrigeration circuit

Annex III

Justification for refrigerant volume reduction for products using R-290 refrigerant

Safety regulation imposed refrigerant volume limits for flammable refrigerants have the effect of limiting refrigeration system capacity for equipment using R-290. Using refrigerant charge limits currently in place along with current direct expansion AC system design technology and only reducing the refrigerant charge per unit capacity to account for the thermodynamic properties of R-290 results in the following maximum equipment cooling capacity levels:

- 0.50 kw – no mitigation, no room size limits, sealed system
- 3 kw – no mitigation, minimum room area requirement, sealed system
- Above 3 kw – mitigation, room size restriction, sealed system
- 16 kw - maximum size for R-290 AC equipment operating in an occupied space.

Maximum allowable R-290 system capacity can be increased by refrigerant charge reduction. The refrigerating effect of R-290 is about double that of R-32 and R452B. Therefore less mass flow is required for a given refrigerating effect. This, coupled with decreased liquid and vapor density, allows charge reductions of about 50% vs R-22 in a “drop-in” situation. Lower refrigerant flow rates also allow use of smaller flow passages (smaller tube diameters) in the coils. Tube OD of 5 mm for evaporators and 7 mm for condensers has been tested and found to reduce system charge by up to an additional 50%. This works out to a charge of about 100 gm per kw of cooling for a non-split (window type) AC system. This is still not low enough allow application of R-290 to the entire range of residential AC products without some form of ignition mitigation (assumes the ISO 5149 charge limits for R-290). Small changes in tube OD such as going from 7 mm to 5 mm OD do not require changes to tube pitch or fin surfaces. However, in the case of the evaporator, refrigerant circuit changes will be required to prevent degradation in system performance due to increased refrigerant pressure drop. An additional benefit of tube diameter reduction is substantial coil material cost reduction.

Indirect refrigeration systems, commonly referred to as chillers, are used extensively for large capacity applications. They have the advantage of reduced charge and higher operating efficiency at the unit level but at the expense of system complexity and additional system cost. They also are typically sited outside of the occupied space and therefore may avoid some of the ignition mitigation normally required for equipment using A3 refrigerants. Overall AC system performance usually ends up on a par with a direct expansion system when energy to operate a fluid pump and efficiency degradation caused by an air to fluid heat exchanger are included in the analysis. Refrigerant charge reduction potential for indirect cooling systems is significant and would roughly double the maximum refrigeration capacities at each of the m_i limit points. Thus maximum capacities increase to 1 kw, 6 kw, and 32 kw for m_1, m_2, and m_3 mitigation levels respectively.

Annex IV

An alternative coil pressure design qualification method for high pressure refrigerants (refrigerants with design pressure greater than R-22)

ASHRAE 15, **Safety Standard for Refrigeration Systems**, and its companion UL 1995, **Heating and Cooling Equipment**, govern product safety for end use AC products in North America. The typical design pressure for R-22 coils is 450 psig, corresponding to 160 F (71 C) saturation temperature. The design pressure is not set by the ambient temperature for the application, but rather by consideration of temperatures that may be experienced during shipment and storage. Per UL 1995 (Clause 61) the design is required to pass a burst test with a minimum burst pressure of 2,250 psig (5 times design pressure). For R-22 the coils achieved 2,250 psig using the standard design and standard manufacturing methods. Employing the same test method and strength requirements to heat exchanger coils using R-410A, R-32, or R-452B would result in a minimum burst test pressure of 3,900, 4,050, and 3,750 psig respectively. Designing coils to meet these pressures would not be practical.

An alternative fatigue test method can be used to quality coil designs for AC products and is found in UL 1995 Clause 61A (Fatigue Test Analysis). For this method, 3 test samples of each design are subjected to a 250,000 cycle pressure test between low and high side design pressures for the actual application, followed by a burst test at 3 times the design pressure. This recognizes the actual system operation where pressure changes occur during cooling cycles from shutdown during the night to the hot afternoon, as well as pressure fluctuations induced by the compressor. The manufacturer has some latitude in determine the high and low pressure for the fatigue test. For R410A, after successful fatigue tests, a burst test pressure is required at 1,950 psig (3 times maximum design pressure of 650 psig - well below the 2,250 psig burst test pressure used for R-22 coils). Burst test pressures for R-32 and R-452B would be approximately 2,040 psig and 1,860 psig respectively (note that these pressures are also below the 2,250 psig burst pressure generally used for R-22 coil design qualification). After initial qualification the test must be repeated at least annually (or every 3 months if coils are considered regular production), on one sample of each coil design produced. Manufacturers of R-410A equipment have found that most existing coil heat exchangers designed for R-22 pass this test with minimal design changes, but with some feature changes. The same is expected to be the case for the design and test pressures used for R-32 and R-452B refrigerants.

The method described above is equally applicable to smooth bore tubes and internally finned coils.

Experience of manufacturers using this coil design qualification method has shown that many fatigue test failures are caused by areas of weakness that can be easily resolved either by manufacturing process improvement, design feature changes, or component quality improvement. Areas of particular importance are:

- Coil heat exchanger tubes must be free of defects such as dents and scratches. Damaged tubes will always produce a fatigue failure.
- The length of coil tubes not covered by fins must be kept to a minimum. This is particularly true for the heat affected zone in tubes outside of the coil casing that are brazed to U-bends or header stubs. The fins provide support for the tube and increase the burst strength of the tube. R-22 designs used 0.5” of length. This was reduced for higher pressure designs.

- Header joint designs need to include reinforcement such as saddle type or flared holes that provide sufficient overlap of material for a sound braze joint. This means that a T-drill or similar is necessary. Cost of T-drill tooling is typically less than US $6,000 per drill head.

- U-bends are generally purchased and it may be necessary to increase the wall thickness of these parts since they will thin during U-bend manufacture.

- The crimp joints that are sometimes used for distributor to coil attachment will not always be sufficiently strong. Designs may need to be changed to a flared end distributor or a purchased flared adapter for this joint. Tooling changes for these features are typically less than US $12,000 per station.

- For headers larger than 1.375” diameter “K” wall thicknesses will probably be required. The heavier wall tube should work on up to 1.625” (41.275 mm) diameter headers. This does not eliminate the need for high quality saddle or flared header to tube joints.

- Brazing quality must be carefully controlled. Especially important are standard brazing procedures and qualification of the manufacturing technician, use of a nitrogen purge during brazing and routine inspection to insure quality. Poor brazing is the largest single source of leaks, which is the largest single warranty expense for manufacturers, and is especially problematic with higher pressure refrigerants.

Purchase or lease of fatigue test equipment that can induce rapid pressure cycles using hydraulic fluids will be direct cost associated with the changeover to any of the higher pressure refrigerants. This cost will vary depending on the size and number of testers required to support a particular facility. Cost estimate for this type of equipment is $100,000 to $200,000 depending on cycle time and number of pressure test ports. Alternatively, an agency could purchase and install the necessary facilities for use for a group of manufacturers. In this case the service is provided as an expense, rather than a capital acquisition or lease. A single test at such a facility will be around US $5,000.

Annex V

Performance and cost Analysis for HX coil tube OD reduction from 7mm to 5mm

Assumptions:

Single circuit mini-split AC unit with a capacity of 5kw (18,000 Btu/hr)
Annual production volume of 200,000 units
Evaporator is 8 tubes tall, 3 row, 20" long with 7mm tubes, half row circuiting, and 12 fpi
Evaporator contains 12 hairpin tubes
Condenser is 24 tubes tall, 2 row, 20" long with 7mm tubes, and 15 fpi
Condenser contains 24 hairpin tubes.
Tube wall thickness is 0.30 mm for 7 mm OD tubes and 0.25 mm for 5 mm OD tubes

Note: These physical attributes are from the specifications for a typical mini-split unit. Actual physical attributes for the units being considered for MLF funding may be different.

Performance for 5 mm OD tubes with full row circuiting in R-290 compared to 7 mm OD tubes with half row circuiting in R-22

Note: NIST[7] coil designer software was used for performance calculations

Description	**R-22**	**R-290**
Tube OD (mm)	7	5
Refrigerant circuits	4	8
Face velocity (fpm)	540	540
Total heat load (Btu/hr)	18,777	18,633
Refrigerant outlet temp (F)	53.4	53.9
Refrigerant flow rate (lbm/min)	4.5	2.4
Refrigerant pressure drop (psi)	3.4	1.6
Refrigerant saturation temperature drop (F)	2.16	1.2

Tube cost reduction estimate for 7 mm OD vs. 5 mm OD tubes in evaporator:

Total length of tubes in evaporator (mm)	8 x 3 x 25.4 x 20 = 12,192
Total copper volume in 7 mm OD tubes (mm^3)	7 x0.3 x 3.14 x 12,192 = 80,394
Total copper volume in 5 mm OD tubes (mm^3)	5 x0.25 x 3.14 x 12,192 = 47,854
Difference in volume (mm^3)	80,394 – 47,854 = 32,540
Volume difference in m^3	0.000032540 m^3
Weight difference (kg)	8,940 kg/m^3 x 0.000032540 = 0.29 kg
Cost of tube including fabrication	$3.00/lb
Cost savings for tube OD reduction in evaporator (US$)	0.29 x 2.2 x 3 = US $1.91 (total annual savings: $383,000)

[7] The National Institute of Standards and Technology.

Tube cost reduction estimate for 7 mm OD vs. 5 mm OD tubes in condenser:

Total length of tubes in condenser (mm)	24 x 2 x 25.4 x 20 = 24,384
Total copper volume in 7 mm OD tubes (mm^3)	7 x.3 x 3.14 x 24,384 = 160,788
Total copper volume in 5 mm OD tubes (mm^3)	5 x.25 x 3.14 x 24,384 = 95,707
Difference in volume (mm^3)	160,788 – 95,707 = 65,081
Volume difference in m^3	0.000065081 m3
Weight difference (kg)	8,940 kg/m^3 x 0.000065081 =c0.58 kg
Cost of tube including fabrication	US $3.00 / lb
Cost savings for tube OD reduction in evaporator (US $)	0.58 x 2.2 x 3 = US $3.83 (total annual savings evaporator: US $767,052)
Total cost savings for 5 mm vs 7 mm OD tubes (200,000 units)	US $767,000 + US $383,000 = US $1,150,000