APPLICATION OF VACUUM INSULATION PANELS IN THE COLD CHAIN – ANALYSIS OF VIABILITY

Hammond E. C. [a,*] and Evans, J. A.[a]

[a] London South Bank University, Churchill Building, Langford, North Somerset, BS40 5DU, UK

* Corresponding author: Tel: +44(0)117 928 9392. Fax: +44(0)117 928 9314, Email: e.hammond@lsbu.ac.uk

ABSTRACT

Vacuum Insulation Panels (VIPs) have already found application in some specialist applications where minimal energy consumption is important and space is at a premium. This paper investigates the feasibility of widespread application of VIPs in the cold chain by embedding them into the polyurethane (PU) foamed walls of traditional refrigerator and freezer cabinets.

Thermal modelling of the insulation of a range of typical refrigerator and freezer cabinets as used throughout the cold chain was carried out both with and without VIPs embedded in the insulating walls. The potential energy savings and payback times were then calculated; for refrigerators the average payback was 9.7 years, for freezers it was 4.5 years.

KEYWORDS

Vacuum Insulation, Thermal Modelling, Cold Chain, Refrigerator, Freezer.

1 INTRODUCTION

VIPs consist of an open cell foam slab enclosed in a barrier film (Figure 1) (Brown, Evans, & Swain, 2007). A high vacuum is achieved within the enclosure, maintained by the impermeability of the barrier film and by the presence of a gas absorber (or getter) within the enclosure. The foam slab maintains the physical dimensions of the panel, supporting the barrier film, reduces convection by the remaining gas molecules and the radiant heat transfer across the panel. The getter absorbs water vapour, atmospheric gasses and gasses emitted by the slab during the life of the panel to maintain the vacuum.

VIPs typically have a thermal conductivity of around 3 $m.W^{-1}.K^{-1}$ (measured at the centre of a panel). However, the film material does influence the conductivity of the panel as a whole and 5 $m.W^{-1}.K^{-1}$ would be more typical when considering the complete panel. Figure 2 compares the thermal conductivity of complete VIP panels to a range of conventional insulations, data is taken from multiple sources to demonstrate a typical performance range for each (Kacimi & Labranque, 2011), (Porextherm GmbH, 2009), (ASHRAE, 2001), (VensilResil Ltd., 2003), (TAASI Corporation, 2012), (Manini, EneaRizzi, Pastore, & Gregorio, 2003), (Domínguez-Muñoz, Anderson, Cejudo-López, & Carrillo-Andrés, 2009), (Thermal conductivity of some common materials and gases, 2012), (Bing, 2006), (Nanopore inc., 2008), (Nanopore inc., 2012).

Much of the published data (based on overlapping VIPs used to form an insulated box) showed only 40 – 50% of the expected benefit of the VIPs over PU foam. More recent research (Hammond & Micic, 2013) has shown that VIPs embedded into PU foamed walls will yield 86% of the expected benefit (assuming manufacturers' thermal conductivity data); the remaining 14% being equivalent to ~2 $m.W^{-1}.K^{-1}$ variation in thermal conductivity of the PU and VIP (within claimed manufacturing tolerances).

Where research has shown VIPs to be only 50 to 60% better, the difference was usually associated with "edge effects" or "thermal bridges" (Brown, Evans, & Swain, 2007) but Kacimi and Labranque (2011) claimed that the metallisation layer on the VIPs was too thin for the thermal conductivity of the barrier film to cause any significant edge effects.

Figure 1 here

Figure 2 here

VIPs are currently integrated into production refrigerating (and other low temperature) appliances, where space and energy efficiency are both of high importance. Typically solid panels of VIP are secured against the outer wall of the insulation cavity before being foamed into position. This reduces heat gain without reducing internal volume. These panels would typically cover around 80 % of the surface area of a cabinet wall but would not extend to the edges or corners of any individual wall. The VIPs are secured to the outer skin using adhesive tape prior to filling the insulation cavity with PU foam. The PU foaming process is unaltered as a result the addition of VIPs except to reduce the volume of foam to account for the VIPs). Figure 3 shows a diagram of the construction of a PU foamed cabinet with embedded VIPs.

Figure 3 here

1.1 Barriers to application

1.1.1 Cost of VIP

The present cost of VIPs is more than that for PU foam, but the energy savings achievable can still make them an economic option. Any increase in the production cost of appliances represents a hurdle or barrier to the take up of the technology and the only way of overcoming the hurdle is to demonstrate that the value of the benefit outweighs the extra cost.

The main benefits of VIPs are the reduced thermal transmission for the same thickness, the reduced space taken by the insulation for the same thermal transmission or a combination of these two, especially where energy indexes are calculated based on internal volume and external dimensions are constrained. It is possible that weight of the materials used may also be important to some users, especially in transport applications.

1.1.2 Application fabrication cost

Since the VIPs considered in this paper were smaller than the total wall area and were inserted into the existing foamed cavity (by taping to the outer steel skin), displacing only a small part of the polyurethane foam, no additional structural changes were required to the cabinets and additional fabrication costs were small.

In applications where existing insulation panels can be directly replaced by VIPs there will be no significant change in fabrication cost. However, if in the application the existing foam insulation is an integral part of the structure, the insertion of VIP's is an extra step in the production process and may require strengthening of the structure in other ways; this will add to the product cost. In appliances where the VIP's barrier film could be vulnerable to damage there may be an additional cost for protection.

1.1.3 Vulnerability of the barrier film

VIPs are vulnerable during manufacture, transport, and fabrication, during applications and in use since the barrier film is very easily punctured. A simple puncture will immediately reduce the insulation value of the panel to no better than a PU foam. The method of installation studied in this paper leaves the VIP extremely well protected but care must be taken during manufacture.

2 METHOD

2.1 Modelling

The FEA modelling was carried out using SolidWorks Flow Simulation software (Dassault Systems, 2012) to quantify the heat reductions which may be achieved by insetting VIPs into the insulation of:

- a domestic refrigerator-freezer,
- a professional service refrigerator,
- a professional service freezer,
- a retail display chest freezer,

Multi-deck retail display cabinets were not considered as the insulation panels could easily be made from thicker PU foam without significant compromise and as larger VIP panels are more complex (and expensive) to manufacture; the payback would be prohibitively long. Similarly, the use of VIPs in transport refrigeration was not considered because large panels which are sufficiently robust are not yet available.

Economic feasibility for each appliance was determined based on current VIP prices. Energy saving and financial payback calculations assumed that the efficiency of heat removal by each appliance was not affected by the reduced heat load.

2.2 VIP costing

20 mm thick VIP panels were considered throughout this study. The cost of each VIP was calculated based on a cost of £38 m^{-2} (€ 46.11 m^{-2}). This was based on an average cost per square meter calculated from a recent quotation for 1000 off quantities of a range of panels up to 600 x 500 x 20 mm provided to the authors at the time of writing. As a comparison, less recent figures were published by Brown, Evans and Swain (2007) implied that costs of VIPs were typically € 29.45 to € 38.52 in production volumes.

2.3 Co-efficient of performance of the refrigeration systems

In order to convert the reduced heat gain to an energy saving, the efficiency of heat removal was required; the COSP (Co-efficient of System Performance) was estimated based on known energy consumption figures for representative cabinets for the professional service refrigerator and professional service freezer models evaluated. Energy consumption data published by (Pederson, Soe, & Jensen, 2004), two cases were presented for each model, one for existing technology and one for “new generation”. The new generation was considered to be representative of current best practice and was used as the basis for the COSP calculation but assumptions for heat gain across the door seal and defrost heat gain (on the freezer only) had to be estimated. Table 12 sets out the figures used. Heat gain through the cabinet walls was taken from the modelling work above. Fan power and total energy consumed (measured in EN441 test room at 25°C, 60% RH with no cabinet door openings) were taken from (Pederson, Soe, & Jensen, 2004). Heat gain across the door seal was estimated by the author and the defrost was based on the heater power stated by Pederson *et al.* with an assumed run-time of 15 minutes every 8 hours.

The COSP for the domestic refrigerator freezer was based on energy label information since the published energy figure for domestic refrigerators is carried out at 25°C and without door openings. The figures used were for a Hotpoint RFAA52P, an A+ rated appliance closely matching the dimensions of the typical appliance modelled. Published energy consumption was 268 kWh per annum and the resulting COSP was 1.4.

2.4 Thermal properties

Thermal conductivity of PU foam and VIPs were taken as 25×10^{-3} and 4×10^{-3} $mW.m^{-1}.K^{-1}$ respectively. These values are taken as the average or "typical" values from the manufacturers' data sheets. Degradation of the insulation over time was not directly considered in the model. In the case of PU foam, degradation can be very rapid unless it is encased in a high integrity enclosure but the VIPs already have a high integrity (although vulnerable to mechanical damage) wrapper. The manufacturer's datasheet claims a vacuum ≤5 mBar on delivery and a rise of <0.5 mBar per year; worst case this equates to a pressure of 10 mBar after 10 years and according to the manufacturer's data sheet the VIP could be expected to degrade from 3.63×10^{-3} $mW.m^{-1}.K^{-1}$ to 4.25×10^{-3} $mW.m^{-1}.K^{-1}$ over 10 years.

The heat transfer coefficients used in the boundary conditions do not have a significant influence on the end result. 10 $W.m^{-2}.K^{-1}$ was used where there was forced air and 5 $W.m^{-2}.K^{-1}$ where there was only natural convection. The reference temperatures are the dominant variable for the models.

Only conduction through the walls was considered. Therefore only the insulation components (varied in the model) needed to be considered. By calculating the heat gain savings at each insulated wall in absolute values (as a reduction in the thermal load through the insulation) it was possible to ignore the other components such as the seal for which thermal data was not known for all cases. Although heat gains through the seal may be significant they were not affected by the introduction of VIPs. Any metal or plastic skins adjoining the insulation were similarly ignored due to their negligible impact on the savings which were being calculated.

2.5 Model setup

The energy savings resulting from the installation of VIPs only comes from reduced heat conduction through the insulating walls which, for a loaded cabinet, will remain approximately constant throughout any cycles due to thermostats or defrosts and any door openings of a short duration. Transient operation of each appliance was therefore ignored as was any heat gain resulting from door openings.

The model only considered steady state heat transfer through solid insulation, air volumes being eliminated from the model by applying heat transfer coefficients to the walls.

2.5.1 Geometries

The geometries were constructed in SolidWorks as Assemblies of Solid Parts. The PU foam of each cabinet was modelled as a single part as were doors. VIP panels were each modelled as separate parts. Dimensions for the geometries were based on measurements of representative production cabinets.

2.5.2 Meshing

The meshing of the geometry was performed using the Flow Simulation automatic mesh function set to its finest resolution (level 8); the resulting mesh sizes can be found in Table 1. The mesh typically resulted in a mesh cell size of around 10 mm with refinement at edges and corners of geometries.

Table 1 here

A sensitivity study was performed (see Table 2) on the retail chest freezer model. Initially a very coarse mesh was used; the resolution was gradually increased through the automatic mesh scale from 3 to 8 until finally an extreme manual mesh based on no single cell being larger than 5 mm in any dimension was run. Variation in the results from the 52 k cell model to the 12.5 million cell model was negligible; since model run times were still acceptably short at the level 8 mesh resolution the data presented in this report was calculated using that resolution.

A level 8 mesh starts with the same initial mesh as levels 5 to 7 but benefits from additional refinements during the calculation to improve resolution in areas with large differentials between adjacent cells. It was assumed that the combination of fine initial mesh and the automatic refinement at level 8 would also be adequate for the other models; the mesh sensitivity study was not repeated for the other models.

Table 2 here

2.5.3 Domestic Refrigerator-Freezer

The system modelled was an upright domestic refrigerator-freezer with evaporator on the back wall of the fridge and tube and wire (shelves) evaporator in the freezer. Figure 4 shows the model geometry and the locations of the VIPs. The boundary conditions used are detailed in Table 3.

The refrigerator walls were nominally 30 mm thick, the freezer walls, including around the compressor step nominally 50 mm thick. The overall dimensions of the cabinet were (WxHxD) 540 x 1715 x 522 mm.

Figure 4 here

Table 3 here

2.5.4 Professional Service Refrigerator

The system modelled was an upright professional service refrigerator with a forced air evaporator in a top mounted cassette. Figure 5 shows the model geometry and the locations of the VIPs. The boundary conditions used are detailed in Table 4.

The walls were 40, 40, 60 and 75 mm thick on the door, cassette, base and sides respectively (based on dimensions measured on typical units). The overall dimensions of the cabinet were (WxHxD) 740 x 1830 x 815 mm.

Figure 5 here

Table 4 here

2.5.5 Professional Service Freezer

The system modelled was an upright professional service freezer with a forced air evaporator in a top mounted cassette. Figure 6 shows the model geometry and the locations of the VIPs. The boundary conditions used are detailed in Table 5.

The walls were 40, 40, 60 and 75mm thick on the door, cassette, base and sides respectively (based on dimensions measured on typical units). The overall dimensions of the cabinet were (WxHxD) 740 x 1830 x 815 mm.

Figure 6 here

Table 5 here

2.5.6 Retail Display Freezer

The system modelled was a glass top retail display chest freezer. Figure 7 shows the model geometry and the locations of the VIPs. The boundary conditions used are detailed in Table 6.

The walls were all 65 mm thick. The overall dimensions of the cabinet were (WxHxD) 2100 x 780 x 850 mm.

The evaporator is a tube wrapped around the walls but this was not modelled. Since only the conduction through the foam was of interest, the glass top was also ignored.

Figure 7 here

Table 6 here

3 RESULTS

The results of each model are presented in Tables 7 to 10. The output from the FEA models was the heat conduction through the walls in Watts and wall descriptions match those shown on the diagrams in Figures 1 to 4.

Further calculations based on the heat conduction data were based on the energy and VIP costs available as described in the method section of this report. Assumptions used for the payback calculations are summarised in Table 11.

Payback times of less than 3 years have been highlighted in the tables in green.

Table 7 here

Table 8 here

Table 9 here

Table 10 here

Table 11 here

Table 12 here

4 DISCUSSION

Only the thinnest panels of the professional service freezer offered payback durations of less than three years. This indicates that the insulation of typical refrigeration appliances is generally well optimised with PU foam insulation.

Where low temperature (freezer) cabinets have thinner than desirable insulation VIPs offer a benefit. Walls of at least 60 mm resulted in payback >3 years whereas walls of 40 mm resulted in payback <3 years. However, of the appliances evaluated there was no obvious reason why a thicker PU foam insulated wall could not have been used in place of the current 40 mm wall given that thicker walls had been implemented in other parts of the cabinet.

In some situations (mainly domestic) the cure time of the insulation can limit the maximum thickness of the walls in order to satisfy production times and the use of internal volume in the energy labelling and sales literature of the cabinet can also encourage the use of thinner walls in order to maximise the internal volume; motivation for VIPs may not therefore be driven by energy payback.

The ambient temperature of 25°C is likely to be warmer than most retail or domestic locations but is possibly low for professional service cabinets in professional kitchens. If cooler ambient temperatures are used then pay back periods will be even longer.

The findings of Hammond and Micic (2013) concluded that only 86% of the benefit predicted may be realised in practice. The results in this report do not account for the reduced performance which would extend payback durations by 14% although validation studies would be required in order to confirm this figure for each appliance type

5 CONCLUSIONS

VIPs are only likely to become more widely used in situations where drivers other than energy pay back dominate; any product incorporating a VIP will need to command a premium price. In situations where space is limited, and particularly where energy labelling indices are based on internal volume, VIPs may become attractive.

The production costs of VIPs needs to fall below €25 per square meter (based on the 20 mm thick panel studied in this report) before they become universally economical for freezer applications and payback periods start to become attractive when implemented with typical PU insulation thicknesses. For refrigerators, even at €25 per square meter the payback is only starting to become interesting on the walls adjoining heat sources, such as behind the condenser or compressor.

6 ACKNOWLEDGEMENTS

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Figure

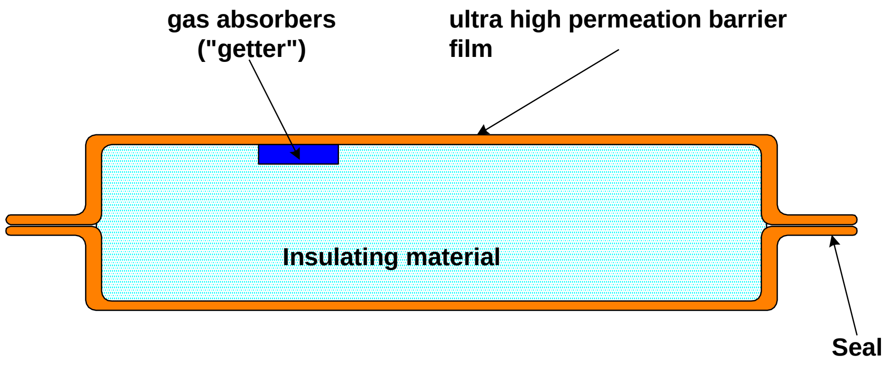

Figure 1. Schematic of a typical Vacuum Insulating Panel (VIP) (Swain & Brown, 2004).

Figure

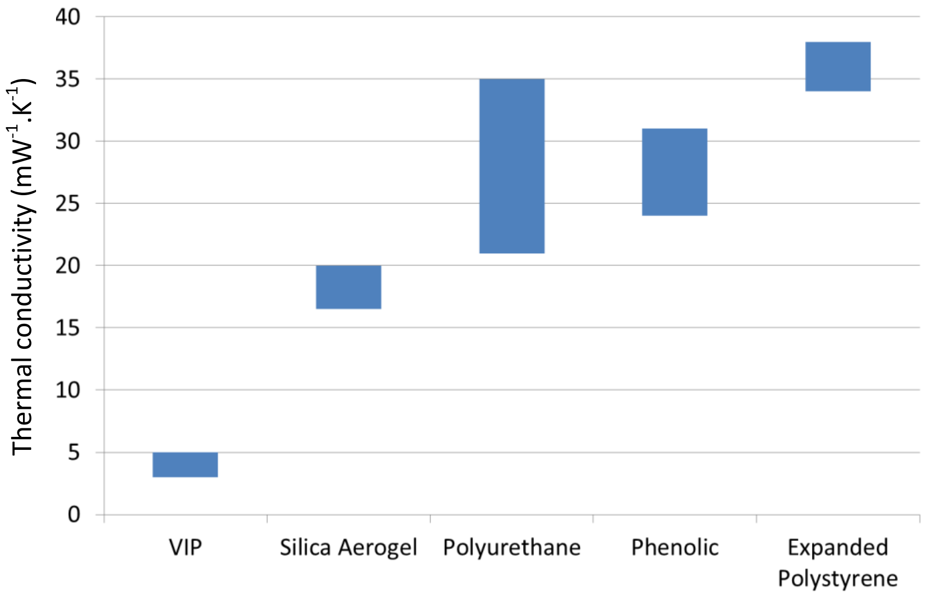

Figure 2. Ranges of typical thermal conductivities of conventional insulation materials and VIPs. Values based on material manufacturers' datasheets and other technical manuals.

Figure

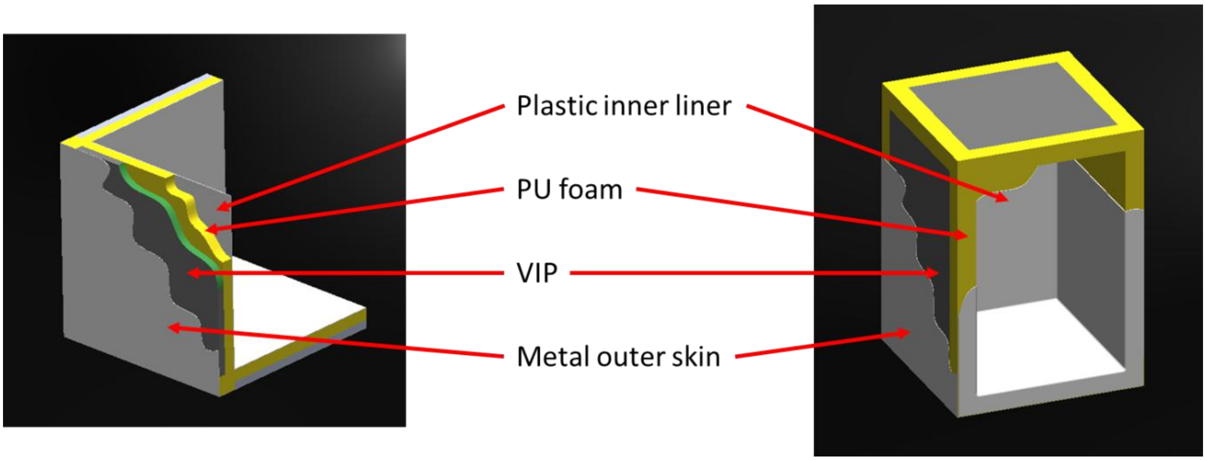

Figure 3. Construction of appliance with VIPs embedded into the PU foamed walls.

Figure

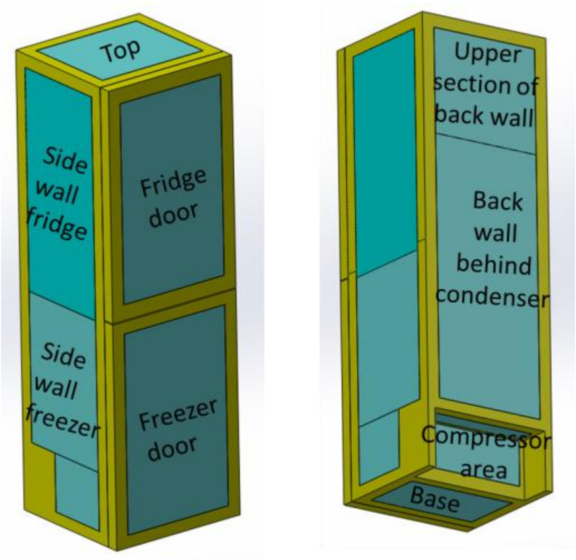

Figure 4. Domestic refrigerator-freezer model layout. PU foam shown in yellow, VIP shown in green.

Figure

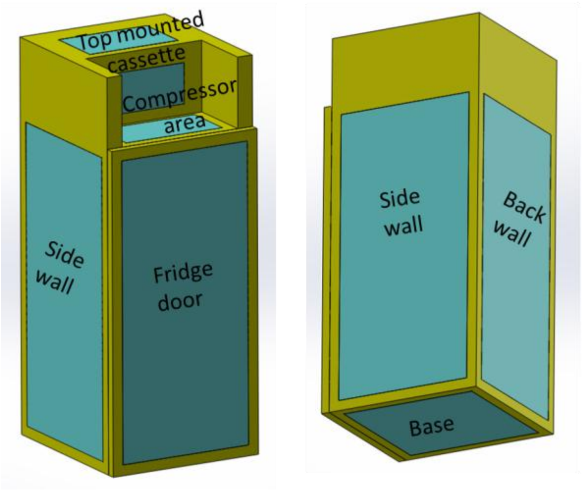

Figure 5. Professional service refrigerator model layout. PU foam shown in yellow, VIP shown in green.

Figure

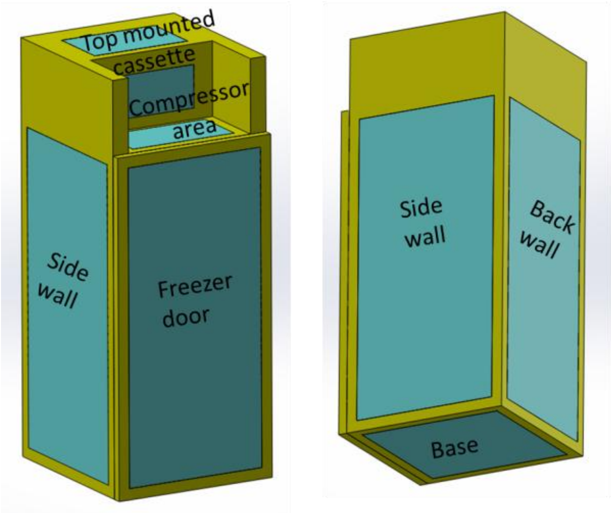

Figure 6. Professional service freezer model layout. PU foam shown in yellow, VIP shown in green.

Figure

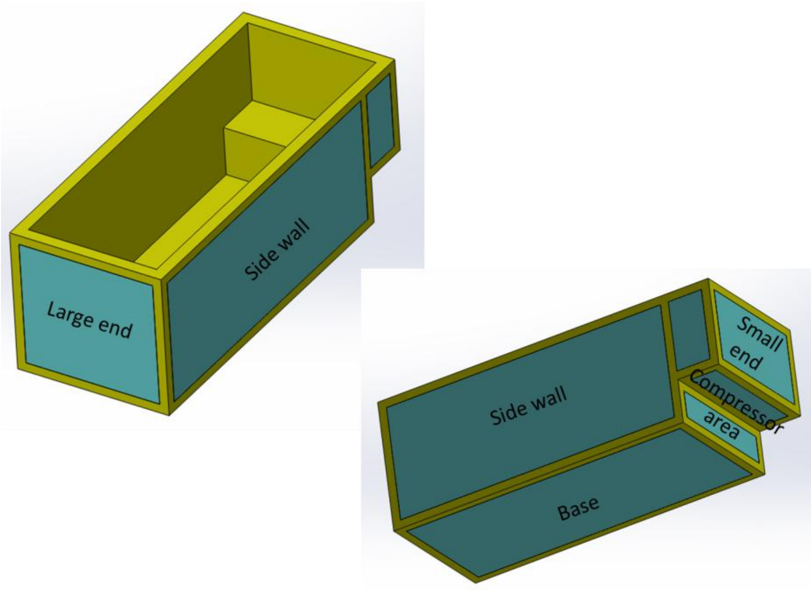

Figure 7. Retail display freezer model layout. PU foam shown in yellow, VIP shown in green.

Figure

Table 1. Mesh size for each model.

Model	No. Mesh Cells (PU model)	No. Mesh Cells (VIP model)
Domestic Refrigerator-Freezer	259,469	1,491,070
Professional Service Refrigerator	1,647,515	1,827,528
Professional Service Freezer	1,634,775	1,813,780
Retail Chest Freezer	337,925	1,022,539

Figure

Table 2. Sensitivity study results.

Mesh level	Auto level 3	Auto level 6	Auto level 8	Manual (5mm maximum cell dimension)
No. mesh cells	52,538	348,961	1,022,539	12,503,216
Calculated heat flows:				
Large side wall	12.3	12.3	12.4	12.3
Large end wall	5.0	5.0	5.0	5.0
Small end wall	3.2	3.3	3.3	3.2
Base	10.6	10.6	10.7	10.6
Condenser wall	5.0	5.1	5.1	5.1
Total heat on cold side	48.3	48.6	48.7	48.6

Table

Table 3. Boundary conditions used for domestic refrigerator-freezer study.

Boundary	Reference Temperature (°C)	Heat Transfer Coefficient ($W.m^{-2}.K^{-1}$)
External walls (to room air)	25	5
External wall (behind condenser)	30	5
External wall (behind and above compressor)	30	5
Fridge walls (non-evaporator)	5	5
Fridge evaporator wall	0	Const. Temp.
Freezer walls	-20	5

Table 4. Boundary conditions used for professional service refrigerator study.

Boundary	Reference Temperature (°C)	Heat Transfer Coefficient ($W.m^{-2}.K^{-1}$)
External walls (to room air)	25	5
Internal cold walls	5	10
Evaporator walls (on cassette)	0	20
Condenser /compressor walls	30	10

Table

Table 5. Boundary conditions used for professional service freezer study.

Boundary	Reference Temperature (°C)	Heat Transfer Coefficient ($W.m^{-2}.K^{-1}$)
External walls (to room air)	25	5
Internal cold walls	-20	10
Evaporator walls (on cassette)	-30	20
Condenser /compressor walls	30	10

Table

Table 6. Boundary conditions used for retail display chest freezer study.

Boundary	**Reference Temperature (°C)**	**Heat Transfer Coefficient ($W.m^{-2}.K^{-1}$)**
External walls (to room air)	25	5
Internal cold walls	-25	Const. Temp.
Condenser /compressor walls	30	10

Table

Table 7. Summary of heat gains and pay back periods for each possible VIP proposed in a domestic refrigerator.

Domestic Refrigerator-Freezer	Heat conduction through walls		Reduction in heat gain through use of VIP		Electrical Energy Saving (COSP = 1.4)			Dimensions of VIP			
	PU only (W)	With VIP (W)	(W)	(kWh.yr^{-1})	(kWh.yr^{-1})	(€.yr^{-1})	Cost of VIP (€)	Length (mm)	Width (mm)	Thickness (mm)	Payback (years)
Total heat through cold walls	43.3	22.2	21.0	184.1	131.5	€ 18.94	€ 128.44				6.8
Side wall -fridge	3.7	1.8	2.0	17.4	12.4	€ 1.79	€ 14.28	835	375	20	8.0
Side wall -freezer (2 panels per wall)	4.3	2.3	2.0	17.9	12.8	€ 1.84	€ 12.12	565 245	375 220	20 20	6.6
Fridge door	4.2	2.4	1.8	15.4	11.0	€ 1.58	€ 15.05	750	440	20	9.5
Freezer Door	5.4	3.3	2.1	18.0	12.9	€ 1.86	€ 15.05	750	440	20	8.1
Back wall above condenser	3.1	1.2	1.9	16.7	11.9	€ 1.72	€ 8.03	440	400	20	4.7
Back wall behind condenser	9.0	3.6	5.5	48.0	34.3	€ 4.93	€ 20.06	1000	440	20	4.1
Above compressor	1.0	0.8	0.1	1.2	0.9	€ 0.13	€ 1.50	440	75	20	11.8
Behind compressor	1.8	1.2	0.6	5.3	3.8	€ 0.55	€ 3.41	440	170	20	6.2
Top	2.3	1.1	1.2	10.3	7.3	€ 1.06	€ 7.52	440	375	20	7.1
Base	1.8	1.0	0.8	7.1	5.0	€ 0.73	€ 5.02	440	250	20	6.9

Table

Table 8. Summary of heat gains and pay back periods for each possible VIP proposed in a professional service refrigerator.

Professional Service Refrigerator	Heat conduction through walls		Reduction in heat gain through use of VIP		Electrical Energy Saving (COSP = 1.5)			Dimensions of VIP			
	PU only (W)	With VIP (W)	(W)	($kWh.yr^{-1}$)	($kWh.yr^{-1}$)	(€.yr^{-1})	Cost of VIP (€)	Length (mm)	Width (mm)	Thickness (mm)	Payback (years)
Total heat through cold walls	40.5	21.6	18.9	165.6	110.4	€ 15.89	€ 202.83				12.8
Side wall	6.5	3.5	2.9	25.7	17.1	€ 2.47	€ 43.09	1400	675	20	17.5
Door	9.8	3.9	6.0	52.1	34.8	€ 5.01	€ 40.86	1400	640	20	8.2
Back panel	6.3	3.7	2.7	23.3	15.5	€ 2.23	€ 40.86	1400	640	20	18.3
Base	3.3	1.7	1.7	14.5	9.6	€ 1.39	€ 19.70	675	640	20	14.2
Under compressor	2.0	1.2	0.7	6.5	4.3	€ 0.62	€ 5.59	490	250	20	9.0
Compressor-evaporator wall	3.1	2.1	1.0	8.6	5.7	€ 0.82	€ 3.90	450	190	20	4.7
Above evaporator	2.5	1.4	1.1	9.9	6.6	€ 0.95	€ 5.75	450	280	20	6.0

Table

Table 9. Summary of heat gains and pay back periods for each possible VIP proposed in a professional service freezer.

Professional Service Freezer	Heat conduction through walls		Reduction in heat gain through use of VIP		Electrical Energy Saving (COSP = 0.9)			Dimensions of VIP			
	PU only (W)	With VIP (W)	(W)	(kWh.yr^{-1})	(kWh.yr^{-1})	(€.yr^{-1})	Cost of VIP (€)	Length (mm)	Width (mm)	Thickness (mm)	Payback (years)
Total heat through cold walls	89.7	47.5	42.2	369.7	410.7	€ 59.15	€ 202.83				3.4
Side wall	14.6	8.1	6.5	56.9	63.2	€ 9.10	€ 43.09	1400	675	20	4.7
Door	22.2	8.9	13.4	117.1	130.1	€ 18.74	€ 40.86	1400	640	20	2.2
Back panel	14.2	8.2	6.0	52.3	58.1	€ 8.37	€ 40.86	1400	640	20	4.9
Base	7.6	3.8	3.8	33.2	36.9	€ 5.31	€ 19.70	675	640	20	3.7
Under compressor	3.9	2.4	1.5	12.9	14.3	€ 2.06	€ 5.59	490	250	20	2.7
Compressor-evaporator wall	6.0	4.1	2.0	17.2	19.1	€ 2.75	€ 3.90	450	190	20	1.4
Above evaporator	5.8	3.3	2.5	21.9	24.3	€ 3.50	€ 5.75	450	280	20	1.6

Table

Table 10. Summary of heat gains and pay back periods for each possible VIP proposed in a retail chest freezer.

Retail Chest Freezer	Heat conduction through walls		Reduction in heat gain through use of VIP		Electrical Energy Saving (COSP = 0.9)			Dimensions of VIP			
	PU only (W)	With VIP (W)	(W)	(kWh.yr^{-1})	(kWh.yr^{-1})	(€.yr^{-1})	Cost of VIP (€)	Length (mm)	Width (mm)	Thickness (mm)	Payback (years)
Total heat through cold walls	96.7	48.7	48.0	420.3	467.0	€ 67.25	€ 234.97				3.5
Large side walls (each) (2 panels)	25.0	12.4	12.6	110.8	123.1	€ 17.72	€ 57.98	1740 420	680 210	20 20	3.3
Large end wall	9.9	5.0	4.9	43.1	47.8	€ 6.89	€ 23.26	750	680	20	3.4
Small end wall	6.2	3.3	2.9	25.4	28.3	€ 4.07	€ 14.36	750	420	20	3.5
Base	23.5	10.7	12.8	112.5	125.1	€ 18.01	€ 59.51	1740	750	20	3.3
Condenser-Compressor walls (2 panels)	7.1	5.1	2.0	17.8	19.7	€ 2.84	€ 10.94	750	160	20	3.8

Table

Table 11. Costing and efficiency assumptions used to calculate results presented in tables 5 to 8.

VIP cost	€ 45.60	€.m^{-2} @ 20 mm thick
COSP	0.9, 1.5 1.4	Freezer, Refrigerator Domestic refrigerator/freezer
Energy Cost	€ 0.12	€.kWh^{-1} (Europe's Energy Portal)

Table 12. Figures used in COSP calculation for professional service and retail.

	Fridge	**Freezer**	
Heat loads	***51.2***	***118.7***	W
Cabinet walls	38.2	84.7	W
Fan	10.0	20.0	W
Door seal	*3.0*	*3.0*	W
Defrost		*11.0*	W
Energy consumed	0.8	3.3	$kWh.24h^{-1}$
Averaged power	33.3	137.5	W
COSP	**1.5**	**0.9**	